Madrid, Spain
La Sede Cultural Centre
23–24 September 2015

36th AIVC Conference
5th TightVent Conference
3rd venticool Conference

Effective ventilation in high performance buildings

PROCEEDINGS

In cooperation with:

Acknowledgments

The conference organisers gratefully acknowledge the support from:

AIVC with its member countries : Belgium, Czech Republic, Denmark, Finland, France, Germany, Greece, Italy, Japan, the Netherlands, New Zealand, Norway, Poland, Republic of Korea, Sweden, UK and USA.

Since 1980, the annual AIVC conferences have been the meeting point for presenting and discussing major developments and results regarding infiltration and ventilation in buildings. AIVC contributes to the programme development, selection of speakers and dissemination of the results. Pdf files of the papers of older conferences can be found in AIRBASE. See www.aivc.org.

TightVent Europe

The TightVent Europe 'Building and Ductwork Airtightness Platform' was launched in January 2011.
It aims at facilitating exchanges and progress on building and ductwork airtightness issues, including the production and dissemination of policy oriented reference documents and the organization of conferences, workshops, webinars, etc. The platform has been initiated by INIVE EEIG (International Network for Information on Ventilation and Energy Performance) and receives active support from the following organisations: Buildings Performance Institute Europe, BlowerDoor GmbH, Eurima, Lindab, Retrotec, Soudal, and Wienerberger. More information can be found on www.tightvent.eu.

venticool

The international ventilative cooling platform, venticool (venticool.eu) was launched in October 2012 to accelerate the uptake of ventilative cooling by raising awareness, sharing experience and steering research and development efforts in the field of ventilative cooling. The platform supports better guidance for the appropriate implementation of ventilative cooling strategies as well as adequate credit for such strategies in building regulations. The platform philosophy is pull resources together and to avoid duplicating efforts to maximize the impact of existing and new initiatives. venticool will join forces with organizations with significant experience and/or well identified in the field of ventilation and thermal comfort like AIVC (www.aivc.org) and REHVA (www.rehva.eu). Venticool has been initiated by INIVE EEIG (International Network for Information on

Ventilation and Energy Performance) with the financial and/or technical support of the following partners: Agoria-NAVENTA, ES-SO, Velux, Wienerberger and WindowMaster

Eduardo Torroja Institute for Construction Science - IETcc-CSIC

The Eduardo Torroja Institute for construction science- IETcc (http://www.ietcc.csic.es/index.php/en/) belongs to the Spanish Scientific Research High Council (http://www.csic.es/web/guest/home). Since its foundation in 1934 its main goal has been to carry out scientific research and technology development on the construction field. This goal is achieved through the development of R+D projects with both public and private funding. The IETcc´s motto is “Tecnica plures opera unica”.

The IETcc also provides scientific and technical support to the construction sector and to the Ministry of Development writing and maintaining the Spanish building regulations, certifies and asses materials, products and building systems and transfers knowledge through patents, publications, consulting, conferences, courses and seminars. Within the context of its participation in the building regulations, the IETcc is involved in the development of the IAQ and energy saving regulations.

The IEtcc is the representative of a Spanish platform that has been recently created with the aim of sharing experience and steering research and development efforts in the IAQ field. The platform comprises research organizations across academia and industry: Sevilla University, The Building Energy and Environment Group of CIMNE-UPC, Alder Venticontrol SA and Siber Zone SL.

INIVE EEIG (International Network for Information on Ventilation and Energy performance)

INIVE was founded in 2001.

INIVE is a registered European Economic Interest Grouping (EEIG), whereby from a legal viewpoint its full members act together as a single organisation and bring together the best available knowledge from its member organisations. The present full members are all leading organisations in the building sector, with expertise in building technology, human sciences and dissemination/publishing of information. They also actively conduct research in this field - the development of new knowledge will always be important for INIVE members.

INIVE has multiple aims, including the collection and efficient storage of relevant information, providing guidance and identifying major trends, developing intelligent systems to provide the world of construction with useful knowledge in the area of energy efficiency, indoor climate and ventilation. Building energy-performance regulations are another major area of interest for the INIVE members, especially the implementation of the European Energy Performance of Buildings Directive.

With respect to the dissemination of information, INIVE EEIG aims for the widest possible distribution of information.

The following organisations are members of INIVE EEIG (www.inive.org):

BBRI - Belgian Building Research Institute - Belgium
CETIAT - Centre Technique des Industries Aérauliques et Thermiques - France
CSTB - Centre Scientifique et Technique du Bâtiment - France
IBP - Fraunhofer Institute for Building Physics - Germany
SINTEF - SINTEF Building and Infrastructure - Norway
NKUA - National & Kapodistrian University of Athens - Greece
TNO - TNO Built Environment and Geosciences, business unit Building and Construction - Netherlands

The following organisations are associated members.

eERG - End-use Efficiency Research Group, Politecnico di Milano, Italy
ENTPE - Ecole Nationale des Travaux Publics de l'Etat, Vaulx en Velin, France
TMT US - Grupo Termotecnia, Universidad de Sevilla, Spain

INIVE

Table of Contents

INDOOR AIR QUALITY REGULATIONS. THE SPANISH CASE.

José A. Tenorio [*1], Cristina Jiménez [1], Pilar Linares[1], Sonia García [1]

1 Eduardo Torroja Institute for construction sciences-CSIC
4, Serrano Galvache St.
Madrid, Spain

**Corresponding author: tenorio@ietcc.csic.es*

EXTENDED SUMMARY

The current regulation in Spain regarding Indoor Air Quality (IAQ) provisions is Building Code (Código Técnico de la Edificación) for dwellings and Regulations on Building Heating Installations (Reglamento de las Instalaciones Térmicas en los Edificios. RITE) for other kind of buildings.

The Spanish Building Code is the regulation that establishes the Safety and Habitability requirements in buildings. It was developed by the Ministry of Housing, in collaboration with the Building Quality Department of Eduardo Torroja Institute for construction Sciences. CSIC. Building Code (BC) came into force in 2006 being in general performance-oriented and is composed by several basic documents. BC was the first regulatory provision in housing containing an IAQ however in a prescriptive approach. BC establish a minimum ventilation rates in habitable rooms. Before BC requirements in general was assumed that traditional ventilation and air from infiltration was enough to maintain IAQ.

In the other hand, the story of RITE goes back to the 1980s to the fulfilment of a number of obligations related to heating, climate control and hot water facilities (RD 1618/1980), this regulation was revised several times 2946/1982, RD 1751/1998, RD 1027/2007, RD 238/2013, in order to adapt requirements to European directives and the TBC. This Regulation lays down the conditions of the facilities to meet the demand of thermal welfare and hygiene through heating, cooling and hot water for a rational use of energy.

However, the experience since the approval of the Code has demonstrated that these requirements must be reviewed to make easier the application of other, especially those referred to energy economy and protection against noise. It has been proposed that the IAQ requirement can be satisfied with the supply of a flow rate which is able to keep the CO_2 concentration under some established limits (an average concentration of 900 ppm and an accumulated value of 500.000 ppm*h yearly accumulated above 1600 ppm). New

flow rate values are given to ensure this fact, lower than those include in the current Building Code.

The Ministry of Developing (Housing) is member of the Inter-Jurisdictional Regulatory Collaboration Committee (IRCC), so it was decided to use a survey with a questionnaire to the rest of the members of this organization, to collect the information. It has been recognised the good practice of show this new proposal to other countries and know the current other counties regulations..
IRCC is an organization whose purpose is “to advance at an international level, framework, guidance, and support documents on construction-related regulatory environment issues relative to the development, implementation, and support of performance-based regulatory systems. The intent is to advance a common understanding of the international regulatory environment, to promote the exchange of information, and to facilitate a more open environment of inter-jurisdictional commerce in the areas of building design and construction.” (http://www.ircc.info//)

Eduardo Torroja Institute for construction Sciences-CSIC developed the questionnaire. The Institute supports the Ministry of Development on the continuous development and update of the Spanish Building Code, which includes regulations on IAQ that are currently under revision. As it was said before this revision responds to the goal of reducing energy demand due to ventilation systems.

The institutions members of the IRCC were invited to fulfil a questionnaire that was focused on the most relevant matters related to IAQ. The questionnaires were sent on February 2015. Information from Australia, Austria, Canada, China, Japan, Netherland, New Zealand, Norway, Scotland, Singapore, Spain, Sweden and United States of America was collected.

The main objective was to know the current state of IAQ regulations in other countries and their proposals of improvement for future versions, and particularly in learning about how IAQ performance in dwellings and car parks is expressed.

The questionnaire was organized in 16 questions in several parts. The first question asked about the area of application. From the second to the thirteenth, it was asked about regulations for dwellings; the fourteenth, about regulations for car park; and the fifteenth about regulations for other buildings. The last one was to collect any comments.

The objective of this communication is to show the Spanish regulation and the new proposal and a comparative approach with the other countries involved in the survey.

ENERGY EFFICIENT VENTILATION FOR NZEB IN MEDITERRANEAN COUNTRIES.

Servando Álvarez, José Manuel Salmerón

Group of Termotecnia, Energy Engineering Department, Engineering School University of Seville

Camino de los descubrimientos s/n 41092 Seville SPAIN

SUMMARY

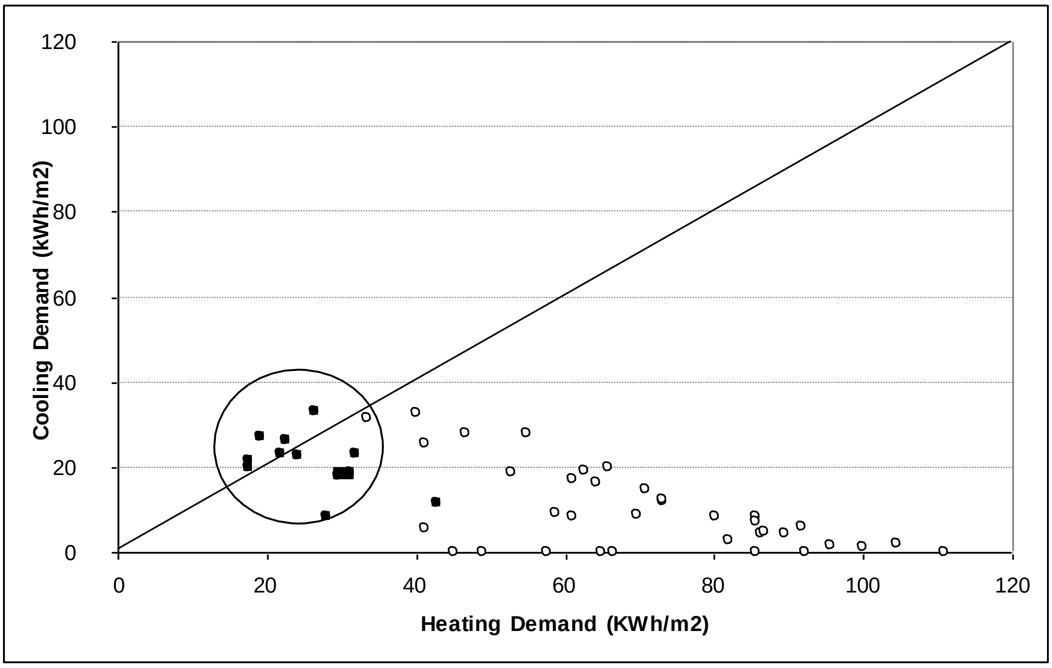
In the definition of the nZEB, the use of only one requirement is misleading. Different requirements are combined to a coherent assessment of an nZEB and to fit the definition given by the EPBD (2010/31 /EU) in article 2. In this presentation, we focus on the first requirement reflecting the performance of the building envelope characterised by the energy needs for heating and cooling.

Reaching the requirements on energy needs for NZEB is a gradual process that is obtained by a roadmap that includes the technologies to be used (the qualitative catalogue) and the variants to take into account for every technology (the quantification of each catalog item).

Usually the most representative and mature technologies in a given country would be listed first. These technologies are typically insulation of opaque components and windows (for heating purposes) and solar control (for cooling purposes). The variant to choose for each technology (or the emphasis to put on it) is a cost-benefit problem whose solution is obtained by the concept of cost optimality (minimal life cycle cost over the lifetime).

The initial cost of the different technologies is not very different from country to country but the associated energy savings are very climatically dependent. Consequently, the energy needs requirement of nZEB in different climates will not always be based on the same technologies and undoubtedly will never be based on the same variants for those technologies who have in common.

Figure shows the typical values of the energy needs for heating and cooling of new single- family dwellings built according to the 2006 Spanish regulation. The results are for the capitals of the 50 provinces. In black appear the Mediterranean locations (in the coast or less than 100km inland).



It can be seen that the heating and cooling needs of buildings in locations of Mediterranean climate exhibit a common pattern characterised by low heating needs (compare to other Spanish locations) and relatively high cooling needs often with a ratio cooling/heating close to the unity.

In the presentation we will examine for these climates the opportunity of energy efficient ventilation technologies covering air-tightness, demand control ventilation, balanced ventilation with heat recovery and night ventilation (for cooling purposes).

IS VENTILATION NECESSARY AND SUFFICIENT FOR ACCEPTABLE INDOOR AIR QUALITY?

Kwok Wai Tham

Department of Building, School of Design and Environment, *National University of Singapore*
SDE 1, 4 Architecture Drive, Singapore 117566, Singapore
Corresponding author: bdgtkw@nus.edu.sg

ABSTRACT

The role of ventilation in achieving acceptable indoor air quality is examined in the light of emerging challenges, alternative mitigation strategies and performance indices within the spatial and time matrix of the indoor environment. By considering the source of contaminants, their nature, transportation mechanism and participation in source-sink relationships, several studies have shown that it may not be feasible nor adequate to rely on ventilation alone to attain the desired level of exposure, especially with respect to airborne aerosolised droplets with infectious potential. Nonetheless, until a full characterisation of the contaminants and appropriate mitigation strategies are developed and effectively implemented, ventilation provides the generic dilution that is necessary to achieve exposure levels that are within desirable levels.

KEYWORDS

Ventilation, acceptable indoor air quality, exposure, contaminant, air flow patterns, airborne infection

1 INTRODUCTION

Ventilation is the intentional movement of outdoor air into the indoors of a building, and performs the primary function of achieving acceptable indoor air quality (ASHRAE, 2013; CEN, 1998). Outdoor air is subjected to requisite cleaning (including filtration and scrubbing) when it does not meet stipulated standards.

Indoor air quality (IAQ) is influenced by indoor sources (including human bio-effluents), indoor reactions (notably indoor chemistry), source-sink relationships and contaminant removal mechanisms.

The evolution of indoor environments in response to sustainability and economic challenges have accentuated IAQ issues associated with emissions from materials and processes, increased occupant density and the cost of air cleaning and air-conditioning.

Whereas guidance on ventilation has been predicated on macro consideration of uniform (well-mixed) dilution and removal of contaminants, recent findings implicate airflow patterns, source location and even respiratory activity as having strong influence on contaminant (and infection) exposure of the occupant.

The necessity and sufficiency of ventilation for achieving acceptable IAQ is discussed from the perspective of the nature, behaviour and distribution of contaminants, consequential human exposure, and the alternatives for mitigating exposure.

2 METRICS OF INDOOR AIR QUALITY

Conventional metrics of IAQ are health and satisfaction as described by acceptable indoor air quality being "air in which there are no contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction" (ASHRAE 2013). This is evaluated by: (i) measuring exposure, typically through IAQ audits comparing measured selected contaminant values in the occupant zone (or breathing zone) against thresholds, and (ii) ascertaining occupant satisfaction through surveys or analysis of complaints.

Based on population epidemiology and dose-response studies, threshold exposure values (STELs or 8 hour averages) have been specified, and most indoor environmental control strategies are structured on achieving these requirements in the most resource efficient manner. Perception of indoor air quality is dominated by sensory (predominantly olfactory) sense; it is arises from a holistic summation across the entire spectrum of inhaled constituents (chemical, biological and physical) that far exceed the number of targeted contaminants measured in an IAQ audit (Cain, 1979). The *olf* unit of odour perception (Fanger, 1988), and more pragmatically in field studies, the use of acceptability of perceived air quality (PAQ) has gained wide implementation (Zagreus et al, 2004; Wargocki et al, 2002; Tham and Willem, 2010).

Airborne infection has recently emerged as an essential consideration and several performance indices related to exposure have been defined including inhalation fraction (Nazaroff) and Personal Exposure Effectiveness (PEE) (Melikov, Cermak and Majer, 2002; Pantelic, Tham and Licina, 2015).

While IAQ standards and regulations have yet to include infection considerations, the evidence for airborne infection and its occurrence across several indoor environments (offices, airplanes, hospitals, transportation environments), and the upsurge in infectious agents (Li et al 2007, Mangili and Gendreau, 2005) suggests that these metrics are needed to complement the existing ones.

3 NATURE OF CONTAMINANTS

Understanding the nature and behaviour of indoor contaminants would help us evaluate the effectiveness of ventilation in achieving acceptable indoor air quality.

Contaminants of outdoor origin, such as particulate matter and environmental bioflora, vehicular and industry related gaseous chemical pollutants opportunistically utilize ventilation to penetrate indoors. When ventilation is not operational, such as after normal operating hours, they infiltrate depending on airtightness and external wind conditions.

Indoor contaminants comprise a multiplicity of sources:

- Building fabric and furniture – gaseous and particulate emissions, thermal and moisture issues; have substantial source-sink capacity
- Ventilation system : coils (especially cooling and dehumidification), filters, ductwork; repositories of particulate and biological contaminants
- Equipment and processes : printers, materials, stationery
- Humans : bio-effluents, infectious particles
- Inter-zonal cross contamination : airborne contamination; pressurization balancing and thermal or mechanically (movement of elevators in lift lobbies) induced

Indoor chemistry, especially associated with highly oxidizing agents such as ozone (introduced via ventilation) may transform primary to secondary contaminants, some of which are more toxic, and often of ultrafine particle size in the form of secondary organic aerosols (Weschler 2000, Fadeyi et al, 2009).

Bacteria and viruses, particularly those that are infectious attach to particulate or moisture nuclei that readily follow micro airflow patterns (Szeto et al., 2009; Pantelic and Tham, 2013).

Source-sink effects are complex. Apart from physical drivers of gravitational settling for larger sized particles and Brownian diffusion for smaller sized ones, chemically gaseous diffusion are motivated by concentration gradients and dispersion by air patterns. Surface and bulk material characteristics, temperature as well as air velocity across surfaces affect the deposition, sorption and subsequent emissions.

4 EXPOSURE

What determines exposure? This relates to spatial and time relationship between contaminants from their release at source and humans, and their transformation during that process. This is contrasted against the more simplistic model of uniform mixing and removal via ventilation alone.

For contaminants that are chemically and biologically inactive, occupant exposure levels are determined by the strength/rate of their emission and their ability to rapidly diffuse across the main airflow streams that transport them within the indoor environment. Gases tend to diffuse well, but the smaller size particulate matter and bacteria and viruses that attach to ultrafine particulate matter tend to follow air streams. Ventilation is effective insofar as their ability to dilute the contaminant concentrations in these two components, and therefore largely depends on how ventilation is being introduced.

Location of sources matter. Melikov (2015) demonstrated that freestream gaseous exposure, using SF6 as a marker, is affected by relative position, whilst Dusan et al (2015a, 2015b) showed the influence of the convective boundary layer in the transport of contaminants from near body sources.

More recent concerns about the movement of aerosolised airborne infectious particles motivated studies which utilized simulated saliva and clearly demonstrated that exposure risk is influenced by both distance and orientation between the infector and susceptible persons. (Pantelic et al., 2015)

The modifying role of air streams in significantly affecting the exposure were established through studies using gaseous surrogates (SF6) as well as particles and simulated saliva. At the zonal level, Pantelic and Tham (2013) showed that downdrafts from air supply diffusers and circulatory vortices generated by the interaction of the airstreams altered the airborne particle movement in the nearfield of an occupant. Dusan et al (2015a) demonstrated the effects of traverse, opposing and assisting flow patterns have varying degrees of influence on particle movement in the breathing zone. At a more individually based scale, Pantelic et al. (2009) demonstrated the advantageous mitigating effect of personalised ventilation against an intruding cough release at various distances and orientations. Personalised ventilation when used alone (Pantelic et al., 2009; Melikov, 2015) or when augmented by personalised exhaust (Yang et al, 2015) have been demonstrated to be effective in reducing personalised exposure in several scenarios.

These series of studies established that a macro evaluation based on ventilation rate for uniform dilution and removal of contaminants is inadequate to ascertain exposure.

5 ACHIEVING ACCEPTABLE IAQ

The efficacy of achieving acceptable IAQ follows a well-established strategy that prioritize source control (elimination, substitution, encapsulation) over air cleaning, and then ventilation (dilution). The cost associated with ventilation is increasing due to the densification and concomitant degradation of ambient air quality of cities, which may necessitate adequate treatment (filtration and contaminant removal) before introduction into the indoors.

A responsible design and procurement that addresses contaminant sources would alleviate the burden of achieving acceptable IAQ both in terms of capital investment and operational costs for air treatment. Facilities management with a clear commitment to sustaining acceptable IAQ is essential; however these are usually non-tenant related responsibilities involving air intake, treatment and air distribution such that at least the ventilation portion is of acceptable quality.

Indoors, the contamination is determined by tenants – choice of materials, equipment and processes and occupant related issues. These translate to source location and concentration characteristics, with air distribution patterns conspiring to mix and transport contaminants resulting in exposure characteristics in a space-time matrix.

Achieving acceptable IAQ is therefore necessitate an understanding of these complex interactions and to address them through a combination of strategies that should include:

- Source treatment of airborne contaminants(both ventilation and recirculated air);
- Source removal or reduction through responsible use of materials and processes, or even provision of isolation and dedicated exhaust as justifiable;
- Supplementary air cleaning where pragmatic constraints and economic considerations favour these over ventilation
- Localised technologies of air cleaning or ventilation judiciously deployed in relation to disrupt or mitigate contaminant pathways towards occupants (here personalized ventilation has been actively developed for targeted environments particularly those associated with healthcare environments).

Given the advancements in these areas, coupled with sensor and control technologies which could embrace occupant feedback in a more engaged manner, the indoor environment is likely set to be transformed to one whereby acceptable IAQ need not be a macro indicator, but one which is much more individual-oriented. Societal evolution towards dynamic workforce and variation in office occupancy, and the diversity of occupant preference of micro-environmental conditions provide impetus in this direction.

6 NECESSITY FOR VENTILATION

Ventilation nonetheless retains a primary role in achieving acceptable IAQ. Its necessity is argued from the following considerations.

Firstly, the indoor environment is an extremely complex mix of chemical and biological contaminants. These may act alone or may react / interact to produce adverse consequences. Recent understanding of ozone-initiate chemistry is a case in point. Ventilation provides the generic dilution, and if judiciously distributed, would significantly enhance IAQ by reducing exposure: generically in the overall room concentration, or at the individual level through personalised ventilation.

Secondly, there are constraints in the efficacy and upper limits of economically feasible air cleaning. Air cleaning works well if the targeted volume of air is effectively channelled through the air cleaning device – this may not be easily achieved. The combination of air cleaning and ventilation may be a considered strategy.

An important consideration is that of exposure being experienced at different locations. A matrix for characterizing exposure would include the following considerations:

- Spatial granularity (floor, zonal, individual)
- Sources and nature (passive sources, active source, human related eg respiratory, speaking, coughing etc …)
- measures of performance against such exposure (eg ACH, PEE, Exposure index, etc…)

Spatial granularity is determined by the envisaged occupancy, usage and contaminant characteristics and the desired response achieved by ventilation and air distribution. It is tightly coupled with the thermal loads and requires suitable sensing and control, ventilation and air distribution design. Sources, processes, activities and nature of contaminants affect the suitability and effectiveness of single, or combination of strategies to achieve a dynamic response commensurate with the desired exposure level. These are defined by the measures that reflect the specificity of performance ranging from comfort through satisfaction to exposure indices.

7 CONCLUSION

Ventilation has a definite role in the overall strategy to achieve acceptable indoor air quality. The multiplicity of contaminant sources, transportation and sinks, further compounded by their interactions bestow upon ventilation a general dilution capability that complements targeted mitigation air cleaning technologies and air distribution strategies. Whilst necessary, it may not be sufficient, nor economically feasible, to be relied upon solely for achieving the desired indoor air quality and exposure levels. Its sufficiency needs to be considered in a matrix dimensioned along spatial granularity, sources and nature of contaminants, and the performance measures. At this present knowledge and technology, it is necessary but its sufficiency depends on the location in the matrix.

8 REFERENCES

ASHRAE. (2013). ANSI/ASHRAE Standard 62.1-2013. Ventilation for Acceptable Indoor Air Quality. ASHRAE, Atlanta, USA.

Cain, WS. (1979). To know with the nose: keys to odour identification. Science 203 (4379), 467-470.

CEN (1998). CR 1752: Ventilation for Buildings: Design Criteria for the Indoor Environment.

Licina D, Melikov A, Sekhar C and Tham KW. (2015a). Human convection flow in spaces with and without ventilation: personal exposure to floor-released particles and cough-released droplets, Indoor Air, DOI: 10.1111/ina.12177.

Licina D, Melikov A, Sekhar C and Tham KW. (2015b). Transport of Gaseous Pollutants by Convective Boundary Layer around a Human Body, Science and Technology for the Built Environment, DOI:10.1080/23744731.2015.1060111.

Fadeyi MO, Weschler CJ and Tham KW. (2009). The impact of recirculation, ventilation and filters on secondary organic aerosols generated by indoor chemistry, Atmospheric Environment 43(22):3538-3547.

Fanger, PO. (1988). Introduction of the olf and the decipol units to quantify air pollution perceived by humans indoors and outdoors, Energy and Buildings, Volume 12, Issue 1, 24 April 1988, Pages 1–6.

Li Y, Leung GM, Tang JW, Yang X, Chao CYH, Lin LZ, Lu JW, Nielsen PV, Niu J, Qian H, Sleigh AC, Su H-JJ, Sundell J, Wong TW and Yuen PL. (2007). Role of ventilation in airborne transmission of infectious agents in the built environment – a multidisciplinary systematic review, Indoor Air 17(1):2-18.

Mangili A and Gendreau MA. (2005). Transmission of infectious diseases during commercial air travel, Lancet 365 (9463):12-18.

Melikov A. (2015). Advanced air distribution: improving health and comfort while reducing energy use, Indoor Air, doi:10.1111/ina.12206.

Melikov AK, Cermak R, Majer M. (2002). Personalized ventilation: evaluation of different air terminal devices, Energy and buildings, Volume 34(8): 829–836.

Nazaroff WW. (2006). Inhalation intake fraction of pollutants from episodic indoor emissions, Building and Environment, 43(3): 269–277.

Pantelic J, Sze-To GN, Tham KW, Chao CYC and Khoo YCM. (2009). Personalized ventilation as a control measure for airborne transmissible disease spread, Journal of Royal Society Interface : S715 – S726.

Pantelic J and Tham KW. (2013). Adequacy of air change rate as the sole indicator of an air distribution system's effectiveness to mitigate airborne infectious disease transmission caused by a cough release in the room with overhead mixing ventilation: A case study. HVAC&R Research 19(8):947-961.

Pantelic J, Tham KW and Licina D. (2015). Effectiveness of a personalized ventilation system in reducing personal exposure against directly released simulated cough droplets, Indoor Air, DOI: 10.1111/ina.12187.

Sze-To, G.N.,M.P.Wan, C.Y.H.Chao, L.Fang, and A. Melikov. 2009. Experimental study of dispersion and deposition of expiratory aerosols in aircraft cabins and impact on infectious disease transmission. Aerosol Science and Technology 43:466–85.

Tham KW and Willem HC. (2010). Room air temperature affects occupants' physiology, perceptions and mental alertness, Building and Environment 45 (1):40-44.

Wargocki P, Lagercrantz L, Witterseh T, Sundell J, Wyon DP and Fanger PO. (2002). Subjective perceptions, symptom intensity and performance: a comparison of two independent studies, both changing similarly the pollution load in an office, Indoor Air, 12 (2): 74–80.

Weschler CJ. (2000). Ozone in indoor environments:concentration and chemistry, Indoor Air 10(4):269-288.

Yang J, Sekhar SC, Cheong KWD and Raphael B. (2015). Performance evaluation of a novel personalized ventilation–personalized exhaust system for airborne infection control, Indoor Air 25(2):176-187.

Zagreus L, Huizenga, Arens E and Lehrer D. (2004).Listening to the occupants: a Web-based indoor environmental quality survey, Indoor Air 14 (s8): 65–74.

VENTILATION PROJECTS WITHIN THE INTERNATIONAL ENERGY AGENCY: OBJECTIVES, APPROACHES AND EXPECTED RESULTS

Per Heiselberg

Aalborg University
Sofiendalsvej 11
9200 Aalborg SV, Denmark
E-mail: ph@civil.aau.dk

ABSTRACT

Ventilation and air infiltration in buildings represents a large share of the building energy demand and account for approximately 25-50% of the energy use for heating and cooling. Therefore, research in energy efficient ventilation systems and improved building air tightness has been an important topic in the EBC programme since its beginning in 1977.

During the years many annexes has dealt with different aspects of ventilation and air infiltration – in fact 22% (15 out of 69 annexes) of the annexes have had it as its main focus and several others has partly dealt with the issue. Annex 5 “Air infiltration and Ventilation Centre” was established as early as in 1979 and is still running. The purpose of AIVC is to promote understanding of the complex behaviour of air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock. Its main role is to disseminate research results presented in accessible and informative publications and software.

In the recent few years the number of annexes running in parallel has increased and several of them deal with ventilation and air infiltration related aspects, se figure 1.

No.	Annex title	Ventilation Strategies	IAQ	Ventilative Cooling	Airtightness	Energy Legislation	Peak Power Demand	Occupancy
69	Strategy and Practice of Adaptive Thermal Comfort in Low Energy Buildings			X		X	X	X
68	Design and Operational Strategies for High IAQ in Low Energy Buildings	X	X			X	X	X
67	Energy Flexible Buildings		X	X		X	X	X
66	Definition and Simulation of Occupant Behavior in Buildings	X		X		X		X
62	Ventilative Cooling	X		X		X	X	X
61	Business and Technical Concepts for Deep Energy Retrofit of Public Buildings	X		X	X			X
60	New Generation Computational Tools for Building & Community Energy Systems	X	X	X	X	X	X	X
59	High Temperature Cooling & Low Temperature Heating in Buildings			X				
5	AIVC	X	X	X	X	X	X	X

Figure 1. On-going EBC annexes and their relation to ventilation related topics.

This has lead to an increasing need for coordination of research objectives, mutual uptake of findings and strengthening of cross-annex outcomes. It is essential for the EBC programme to achieve sufficient interaction and collaboration between the different annexes and ways to ensure this is currently under development and discussion. This also includes improved cooperation with Annex 5 on the dissemination of research results.

The key note will present the objectives, approaches and expected results of the on-going annexes relevant for ventilation and air infiltration in buildings and illustrate the importance of and expectations to an increased cross-annex coordination and cooperation.

BIM, INFORMATION ON SUSTAINABLE BUILDING AND PERFORMANCES

Frank HOVORKA

SBA
4, avenue du recteur Poincaré
Paris
Frank.hovorka@caissedesdepots.fr

REHVA
40 rue de Washington
Bruxelles
Frank.hovorka@caissedesdepots.fr

ABSTRACT

Sustainability thresholds generating Value
There is an expanding literature on the value of sustainability features in buildings (European Commission, 2013; World GBC, 2013; World GBC, 2014). While several publications focus on the price differentiation between buildings with sustainability credentials and buildings with no sustainability credentials (Eichholtz *et al.*, 2010; etc.), others examine the costs and benefits of sustainability features individually referring to both monetary and intangible values (Heerwagen, 2000).
These various studies usually refer to different concepts of value and encompass benefits that may not always directly profit the investors themselves. The value of sustainability can thus be interpreted differently according to the stakeholders and scope considered. On the one hand, market value refers to the price at which the good would be traded in a perfect open market. On the other hand, a concept of total value could be defined to encompass all the benefits associated with sustainability features for the various stakeholders (investors as well as users, local authorities, citizens, etc.). This notion would include a wider range of benefits which may not all be priced by the market. However, examining the mechanisms through which benefits for the different stakeholders could impact investors helps better understand the financial gains investors can expect from sustainability.
.

KEYWORDS

Sustainable investment, risk analysis, green value

1 INTRODUCTION

ESG and climate risks should be managed by asset owners and trustees through their determined investment framework and approach, and importantly through their selection of managers and advisers as integral part of their investment strategies. The result is that all asset specific and selection dialogues contain an ESG and climate risk component embedded and as determined through the strategic review process

- **Holistic description of the building performance**

As a starting point, sustainability-related features should be treated as constituents of an "extended" approach of describing building quality. A separate consideration of overall building quality and sustainability-related features does no longer make sense.
Data collection should be integrated in the design and building management processes. The cost of systematic data collection and storage would probably be lower than the costs of one-off due diligences required to start each time from scratch. Information should be collected directly from the people who create the data or who have access to the data, namely designers and contractors for new buildings, and facility managers for existing buildings. The verification of this information by third parties would increase the value of the data.

- **Integration of sustainability-related information into market value and investment worth**

Current valuation and investment decision practices partially incorporate sustainability-related features. The presence of a sustainability certification (BREEAM, LEED, HQE, BNB/DGNB or equivalent) sometimes appears "translated" into an additional rental value in markets where certification schemes have not yet widely spread. Conversely, its absence is increasingly often translated into a discount in markets where certification credentials have become standards. Capital expenses increasingly include costs for sustainability retrofit in particular when investors aim to achieve a given sustainability credential. Adjustments are also sometimes made to operation expenses, letting periods and yields. However, the type and scale of the adjustments performed are still heterogeneous.
Improvement should be made to better account for technical data and individual features and propose a more standardised integration of sustainability-related features. Discounted cash flows (DCF) approach appears particularly appropriate to better integrate sustainability-related information into valuation exercises. An assessment of the sustainability performance and an examination on how the market context respond to sustainability performance can be used to properly adjust in a transparent manner the various input parameters (rents, rental growth, operation and capital expenses, durations to let, yields and risk premium) of the DCF calculation.
The adjustments completed should be documented and the DCF parameters should be presented in a standardised format allowing for transparency on how sustainability features are integrated into the valuation. A generic format for this transparent and comprehensible integration is suggested in this report.

In addition, uncertainty associated with the reliability of the information used to assess sustainability performance and the potential impact of said performance on market value drivers should be accounted for. Accounting for uncertainty in the input variables is paramount in order to avoid the impression of unrealistic levels of precision. Monte-Carlo simulation is a method of choice to account for the impact of uncertainties on the valuation output and present a sensitivity analysis. The simulation outcome is a value distribution.

- **Accounting for flexibility and adaptability into building investment worth appraisals and decision-making**

Integration of flexibility into investment worth appraisal and investment decision-making is important as flexibility can have a considerable impact on building occupancy rates, retrofit costs and environmental performance through lower consumption of building materials. Two

key types flexibility should be distinguished: service flexibility and adaptability (Kendall *et al*., 2013). Currently there is no common explicit technical requirements, metrics for assessment or benchmarks to investigate these two types of flexibility. In addition, although building owners, valuers and analysts increasingly acknowledge the impact of service flexibility and adaptability on market value, there is no standardised framework to integrate the flexibility information collected into investment decisions and valuation exercises.
Moreover, valuing flexibility in buildings is not straightforward due to the uncertainty on the potential future cash flows. Pricing flexibility is difficult because its value depends particularly on the uncertainty of the future organizational activities, i.e. whether the flexibility will be actually put into use (e.g., Vimpari *et al*., 2014). Since the value of flexibility is a contingent claim into the future, real option analysis (ROA) could be used to account for service flexibility and adaptability. Based on expert interviews and a review of state-of-the arts projects and studies on valuing flexibility, this report recommends valuers and building owners to apply ROA as a supplement to the DCF valuation. A simple calculation using payoff methods is presented. This method is practical and straightforward, as only three payoff scenarios are needed for valuation.

- **Accounting for risks and resilience against future changes**

Current valuation methods do not very well account for risks associated to shift in the future market context. However, future changes in the users' expectations or in the regulatory framework would impact property value since retrofit works would be required to maintain the building attractiveness. A flexible design enabling the owner to adapt his building to the evolutions of the context would offer a protection against these risks and would thus improve the value of the building. For example, better fits between users' needs and buildings space can improve occupancy rate, reduce capital costs required to update the building. In special cases –in particular for owner occupant willing to decide between different retrofit solutions- a simplified real options method may allow to calculate an option value of a sustainable flexible design. The output may be added to the DCF results.

2 CONCLUSIONS

Current valuation methods do not very well account for risks associated to shift in the future market context. However, future changes in the users' expectations or in the regulatory framework would impact property value since retrofit works would be required to maintain the building attractiveness. A flexible design enabling the owner to adapt his building to the evolutions of the context would offer a protection against these risks and would thus improve the value of the building. For example, better fits between users' needs and buildings space can improve occupancy rate, reduce capital costs required to update the building. In special cases –in particular for owner occupant willing to decide between different retrofit solutions- a simplified real options method may allow to calculate an option value of a sustainable flexible design. The output may be added to the DCF results

3 ACKNOWLEDGEMENTS

Yona Kamelgarn
Thomas Lutzkendorf
David Lorenz

4 REFERENCES

Adair A., Hutchison, N. (2005) The reporting of risk in real estate appraisal property risk scoring. *Journal of Property Investment & Finance*, Vol. 23, n° 3, pp.254 – 268.

Bozorgi, A. (2013) Integrating value and uncertainty in the energy retrofit analysis in real estate investment. Doctoral thesis.

Chegut, A., Eichholtz, P., Kok, N. (2014) Supply, Demand and the Value of Green Buildings. *Urban Studies*,Vol. 51, n° 1, pp. 22-43.

De Neufville, R. (2002) Architecting/designing engineering systems using real options, ESD-WP-2003-01.09, Engineering Systems Division, Massachusetts Institute of Technology, Cambridge, MA, 29-30 May.

Eichholtz, P., Kok, N., Quigley, J. M. (2010) Doing Well by Doing Good: Green Office Buildings. *American Economic Review*, Vol. 100, n° 5, pp. 2494–511.

Eichholtz, P., Kok, N., Quigley, J. M. (2013) The economics of green building. *Review of Economics and Statistics*, Vol. 95, n° 1, pp. 50-63.

European commission (2013) *Energy performance certificates in buildings and their impact on transaction prices and rents in selected EU countries*. Final Report. DG Energy.

Fuerst, F., McAllister P. (2011) Green Noise or Green Value? Measuring the Effects of Environmental Certification on Office Values. *Journal of Real Estate Economic,* Vol.39, n° 1, pp. 45–69.

Kajander, J.-K., Sivunen, M., Junnila, S. (2014).Valuing Indoor Air Quality Benefits in a Healthcare Construction Project with Real Option Analysis. *Buildings, Vol. 4.*

Kamelgarn, Y., Hovorka, F. (2013) Energy efficiency strategy at the portfolio scale of a property owner. *Rehva Journal*, Vol. 50, n°1, p. 41 -46.

Lützkendorf, T., Lorenz, D. (2007) Integrating sustainability into property risk assessments for market transformation. *Building research & Information*, Vol. 35, n°6, pp 644-661.

Nappi-Choulet, I., Décamps, A. (2013). Capitalization of energy efficiency on corporate real estate portfolio value. *Journal of Corporate Real Estate*, Vol. 15, n° 1, pp. 35-52.

RICS (2009) *Sustainability and Commercial Property Valuation*. RICS Information Valuation Paper n°13.

RICS (2011) *Best practice & Guidance note for technical due diligence of commercial, industrial & residential property in continental Europe*. RICS Europe 2011.

RICS (2013) *RICS Sustainability and Commercial Property Valuation*. Global Guidance Note 2013. Available online at: http://www.rics.org/fr/knowledge/professional-guidance/guidance-notes/

RICS (2014) *RICS Red Book 2014*. Available online at:http://www.rics.org/fr/knowledge/professional-guidance/le-red-book/red-book-2014-in-full/

Sayce, S., Lorenz, D., Michl, P., Quinn, F., Lützkendorf, T. (2013) *RICS members survey on the uptake of VIP 13*. Work in progress.

Sivunen, M., Kajander, J-K., Toivo, J.; Kiiras, J. (2014) Managing risks related to functional changes by Design Alliance. *Procedia Engineering.*

UNEP FI (2014) *Sustainability Metrics : Translation and impact on property investment and management.* Available online at: http://www.unepfi.org/publications/property/

ANALYSIS OF RESULTS FROM ATTMA LODGEMENT – WHAT ARE THE REALISTIC AIT PERMEABILITY CHARACTERISTICS OF UK HOUSING

Barry Cope – Scheme Manager
ATTMA

Airport House, Purley Way, Croydon, London, CR0 0XZ

ABSTRACT

ATTMA, the Air Tightness Testing & Measurement Association has introduced mandatory lodgement for all members, representing over 140 companies and over 350 test engineers across the UK. This presentation will give an oversight as to how we have made lodgement mandatory, the software we are using and finally some statistical analyses showing where the UK is at the moment with average Air Permeability (AP_{50}) results.

1 INTRODUCTION

ATTMA were recently (1st January 2015) appointed to operate as a Competent Persons Scheme in the UK air testing industry following the British Institute of Non-Destructive Testing's request to no longer run the Competent Persons Scheme. From this, we have been able to introduce mandatory lodgement of testing for a multitude of reasons.

2 LODGEMENT

Lodgement has been widely accepted by the majority of the industry who see lodgement as a way of protecting their own business. Below are the steps we have taken in order to introduce lodgement

2.1 Proprietary Software

We have worked alongside the main fan manufacturers used in the UK to support the development of their test software to be able to lodge tests from within the software. The tester's credentials are stored within the software and automatically used to lodge the test once the user requests. A demonstration of this process will be shown during the presentation.

2.2 ATTMA Software

We have developed our own software which can also be used. By developing our own test software we are able to control the fields and are able to make updates and changes as required. A short demonstration will be shown.

2.3 Lodgement Database

We outsourced the development of a lodgement database which acts as a portal for all testing members. A short demonstration of this will be shown during the presentation.

3 STASTISTICAL ANALYSIS

One of the main reasons for creating and using lodgement is to provide independent statistical analysis of live air testing data that is not only from a single company. The presentation will break down the latest data and show industry trends.

4 CONCLUSIONS

Conclusions will be discussed as part of the presentation.

BUILDING AIRTIGHTNESS IN GERMANY - WHAT ARE THE DRIVING FORCES?

Oliver Solcher[1]

1 Fachverband Luftdichtheit im Bauwesen FLiB e.V.
Kekuléstrasse 2-4
12489 Berlin, Germany

SUMMARY

Building airtightness in Germany is on a good way. The latest survey amongst FLiB members shows the n50-values are much better than the benchmarks given in EnEV 2014 (German EPBD). For airtightness tests in 2014 the average n50-value of single-family houses is 1.1 ACH for new buildings and 1.6 ACH for refurbishments. In multi-family houses the average n50-value for new buildings is 0.9 ACH and 1.5 ACH for refurbishments.

One major driving force seems to be the EnEV in combination with funding programs. EnEV gives certain benchmarks to achieve, in order to have a benefit in energy performance calculation. EnEV prescribes a durable airtight building envelope but the test itself is not mandatory. Funding programs of the KfW (Kreditanstalt für Wiederaufbau) give out subsidies or credits with low interest rates only if an airtightness test is done and the n50-value complies with the benchmarks.

In the survey 2/3 of the airtightness tests in 2014 were done to proove that EnEV benchmarks were achieved.

The test itself does not improve airtightness. To achieve a certain level of air tightness the building envelope must be planned considering an airtightness concept. To match the KfW-requirements an airtightness concept is mandatory as well. In cooperation with KfW FLiB worked out an internet platform where planners, craftsmen and building owners get the information necessary to make an airtightness concept.

6 YEARS OF ENVELOPE AIRTIGHTNESS MEASUREMENTS PERFORMED BY FRENCH CERTIFIED OPERATORS: ANALYSES OF ABOUT 65,000 TESTS

Adeline Bailly*[1], Gaëlle Guyot[1], and Valérie Leprince[2]

1 CEREMA – Direction Territoriale Centre-Est
46 rue Saint Théobald – BP128
38081 L'Isle d'Abeau Cedex, FRANCE
*Corresponding author: adeline.bailly@cerema.fr

2 PLEIAQ
84 c Av. de la République
69330 Meyzieu, FRANCE

ABSTRACT

Since 2000, the French EP-calculations have been considering thermal losses due to building envelope airtightness. The last two regulations (RT2000 and RT2005) had included a default value for airtightness and the possibility to use a better-than-default value with a mandatory justification of this value, especially for voluntary approaches such as the BBC-Effinergie label. In 2013, strengthening the airtightness has become a requirement of the current EP-regulation (RT2012). It has implemented a limit value for airtightness for all new dwellings, and the mandatory justification of the building airtightness level through either an airtightness measurement or the application of a certified quality management approach. Therefore, there are more and more measurements of building envelope airtightness performed in a regulatory context.

In order to assess the quality of those measurements, the French Ministry for Ecology has implemented a process to certify the measurers. The first step of the process involves a qualifying State-approved training, a training exam, and the justification of a sufficient testing experience. Then, every tester is required to fill in a standard form every year. This form describes, for each measured building, various construction characteristics and airtightness measurement results. Since 2009, Cerema has been gathering the forms filled in by all qualified measurers. In 2015, those forms amount to about 65,000 measurements performed on dwellings and non-residential buildings in France.

This paper presents an analysis of this database. First, it proposes an overview of the measured buildings characteristics, including the main structural material and the use of the buildings. The second part of this paper presents first analyses on envelope airtightness. It describes the location of the leaks from criteria established in 2012. It also presents the evolution of the measured airtightness depending on various parameters such as the date of construction and the volume of the buildings. It also provides hints to evaluate the bias induced by thermal conditions through an evaluation of the impact of the season of measurement on the measured airtightness. The last part of this paper gives some feedback about the consequences of requiring a limit value: firstly, on the airtightness of buildings (both those subject to the requirement and those that are not), and secondly, on the practices of measurers.

In conclusion, this paper includes guidelines for using those data for different purposes, such as improving the control of certified measurers and reinforcing airtightness requirements for the next regulation.

KEYWORDS

Envelope airtightness – Measurements – Database

1 GENERAL CHARACTERISTICS OF THE DATABASE: DATA SOURCES

Since 2000, the French EP-regulation takes into account the airtightness of a building envelope. Previous EP-regulations RT2000 and RT2005 used to propose a default value for airtightness, with the possibility to use a better value (if justified). Now, the RT2012 regulation requires a limit airtightness level for residential buildings that has to be justified, but still allows to use default values for non-residential buildings (Charrier, 2015).
Since 2008, an airtightness measurement performed in order to comply with the current EP-regulation, or to justify a better-than default value in the thermal calculation, has to be performed by a third-party tester, certified by Qualibat, the certification body. Moreover, all airtightness tests performed in order to obtain the Effinergie certification also have to be made by a certified tester.
In July, 2015, almost 1,000 testers are certified. The certification includes not only training, examination and testing experience, but also a yearly follow-up. Therefore, all certified testers have to fill in a professional standard form that includes results of all airtightness measurements they have done within the year.
Each year, Cerema gathers those forms and fill in a national database with the airtightness tests results. In July, 2015, the database includes more than 68,000 tests.

The current version of the professional register proposes, for each test, to fill in 39 fields. 29 of them are required to comply with the tester certification. Those fields give information about:

- Building general information: owner, location, use, year of the construction, year of the rehabilitation;
- Special requirements: label, certification;
- Building main characteristics: main material, constructional type, insulation, ventilation system, heating system;
- Measurement protocol: operator, date of measurement, measurement device, time of measurement (building state), method;
- Measurement input data: envelope area (excluding low floor), floor area, volume;
- Measurement results: C_L, n, q_{a4}, n_{50}, uncertainties;
- Leaks: classification of the leaks (46 categories).

The database is composed with this information, after removal of duplications, irrelevant data and incomplete recording.

2 OVERVIEW OF BUILDING MEASURED CHARACTERISTICS

The RT2012 imposes to justify the envelope airtightness only for residential buildings. Nevertheless, some non-residential buildings have been measured, mostly in order to use a better-than-default value in the EP-calculation. Moreover, the Effinergie+ label imposes envelope airtightness measurement for non-residential buildings if the floor area is below 3,000 m². Therefore, the database includes residential and non-residential buildings. The main characteristics of those buildings are presented in the following paragraphs for:

- single-family houses;
- multi-family dwellings;
- non-residential buildings.

The data for about 65,200 buildings have been analysed (about 3,000 entries are incomplete and then have not been taken into account in this study).

2.1 Characteristics of residential buildings

More than half (35,382) of the measured buildings are single-family houses. Most of them are new buildings: 94% have been built since 2010. For others, about 350 (1%), the airtightness measurement has been performed after a rehabilitation.
Concrete block houses and brick houses represent 81 % of the measured houses, and 15% are built in wood. Others material are stone, wood and concrete combined or steel, and a few are built with straw or hemp.

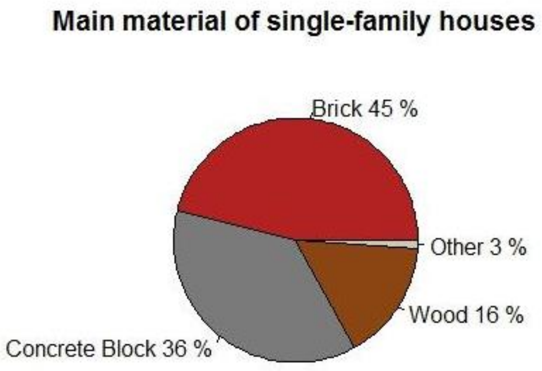


Figure 1: Main material of single-family houses

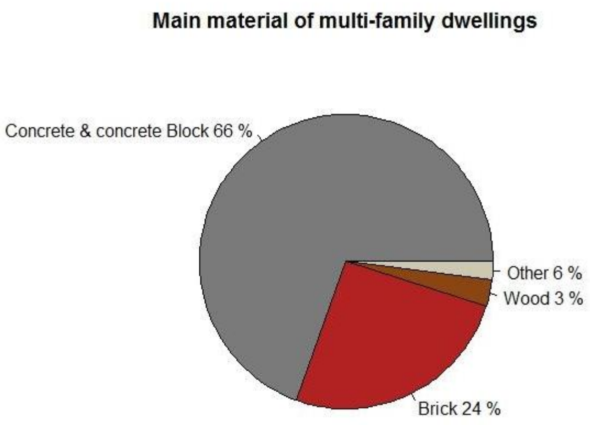
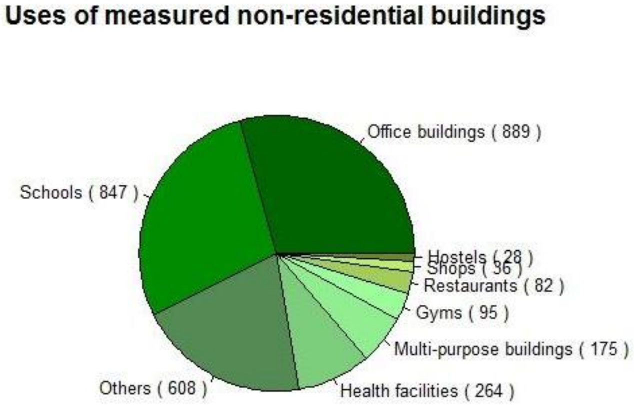


Figure 2: Main material of multi-family dwellings

The second type of buildings are multi-family dwellings: 26,823 buildings have been analysed in this study. They are almost all new buildings: 96% have been built since 2010. For those buildings, concrete and concrete block (66%) and brick (21%) are also the main material used. Wood is only used in few cases (3%).

2.2 Characteristics of non-residential buildings

Even if RT2012 does not impose an airtightness level for non-residential buildings, the database includes measurements results for about 3,000 non-residential buildings. They are mostly new buildings: 92% of them have been built since 2010. They are for the main part office buildings (29%) and schools (28%). Others are mainly health facilities, multi-purpose buildings, shops, gyms and restaurants. The entries of the “others” category (608) presented in figure 3 are lacking the “use” field. Half of the measured non-residential building are built in concrete (47%) and one quarter is built with wood (24%).



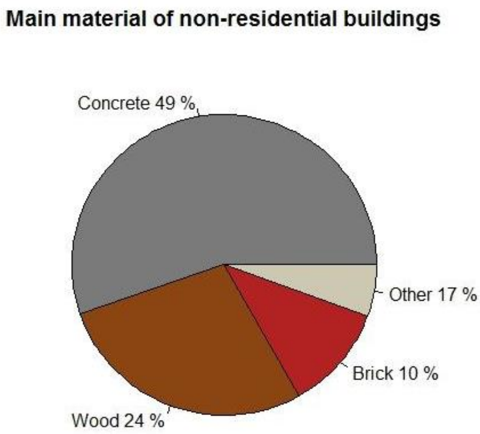


Figure 3: Uses of measured non-residential buildings

Figure 4: Main material of multi-family dwellings

3 EVOLUTION OF THE AIRTIGHTNESS ENVELOPE OF THE FRENCH BUILDINGS

In France, the reinforcement of buildings envelope airtightness has been pushed at first by the BBC-Effinergie label, which has imposed, since 2008, q_{a4} limit values depending of the type of dwelling. Then, RT2012 has imposed those limits as compulsory limits for all new residential buildings since 2013. The evolution of the airtightness level has been analyzed for each of the three building categories.

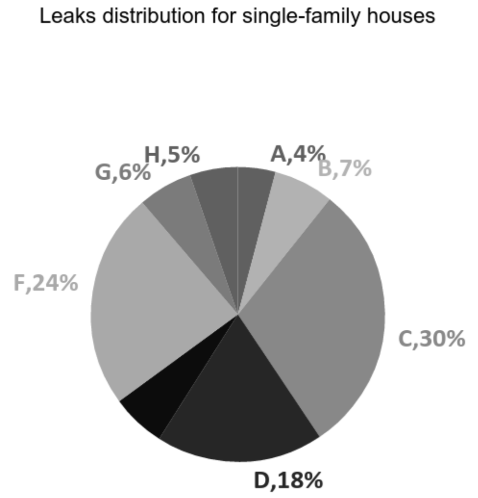
3.1 First analysis: leaks location

Eight leaks locations have been established in 2012:

- A: Main envelope area
- B: Wall, roof and floor junctions
- C: Doors and windows
- D: Building component penetrating the envelope
- E: Trapdoor
- F: Electrical components
- G: Door/wall and windows/wall junctions
- H: wood-burner, chimney, elevator, cooker hood…

For each of them, 3 to 8 subclasses are also defined. Those subclasses have been filled in for almost 45,000 buildings.

For single-family houses, an average of 5.4 types of leaks per house are reported by the tester. 30% of the leaks are observed on doors and windows, especially on junctions between windows and jamb and roller shutter box. Electrical components are also an important source of leaks (24%), just as all buildings components penetrating the envelope (18%), in particular ventilation terminal devices (leakage all around them), pipes and ducts (electrical, plumbing).



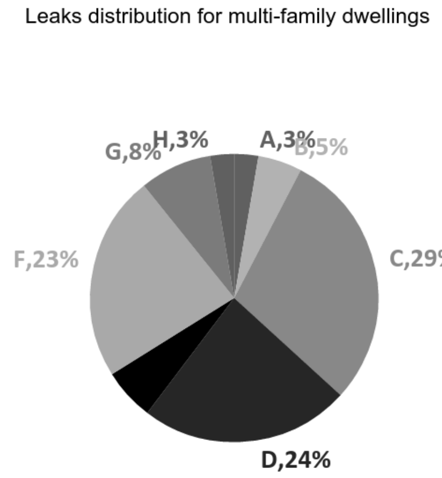


Figure 5: Leaks distribution for single-family houses

Figure 6: Leaks distribution for multi-family dwellings

The same analysis has been conducted on multi-family dwellings, which leads to an average of 4.8 types of leaks reported per building. The leaks are mainly observed at the same place as for single-family houses: doors and windows (29%), electrical components (23%) and components penetrating the envelope (23%).

Data are also available for 1,772 non-residential buildings. They confirm the previous results: on average, 5.7 types of leaks are reported per buildings, distributed in the same main categories as residential buildings.

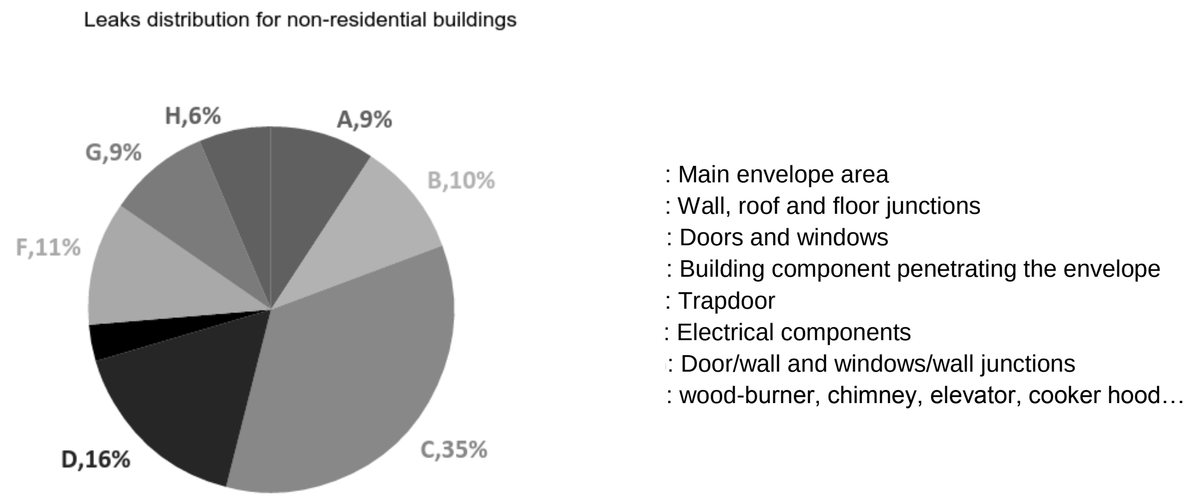


Figure 7: Leaks distribution for non-residential buildings

3.2 Evolution of measurements performed on residential buildings

The RT2012 regulation applies to all new buildings for which the building permit has been accepted since 1 January 2013. Only a few of them are built and have already been measured. Therefore, most of the measured buildings analysed in this study do not have to comply with RT2012. Nevertheless, most of them have an airtightness objective. Indeed, 72% of the single-family houses were applying for the BBC-Effinergie label, which included an airtightness requirement: q_{a4} has to be below 0.6 $m^3.h^{-1}.m^{-2}$, that is around n_{50}=2.3 h^{-1}. Only about 6,200 of the measured houses did not have a target airtightness value. The same observation is made for multi-family dwellings: 90% of them were applying for the BBC-Effinergie label, which imposed an airtightness level: q_{a4} has to be below 1.0 $m^3.h^{-1}.m^{-2}$.

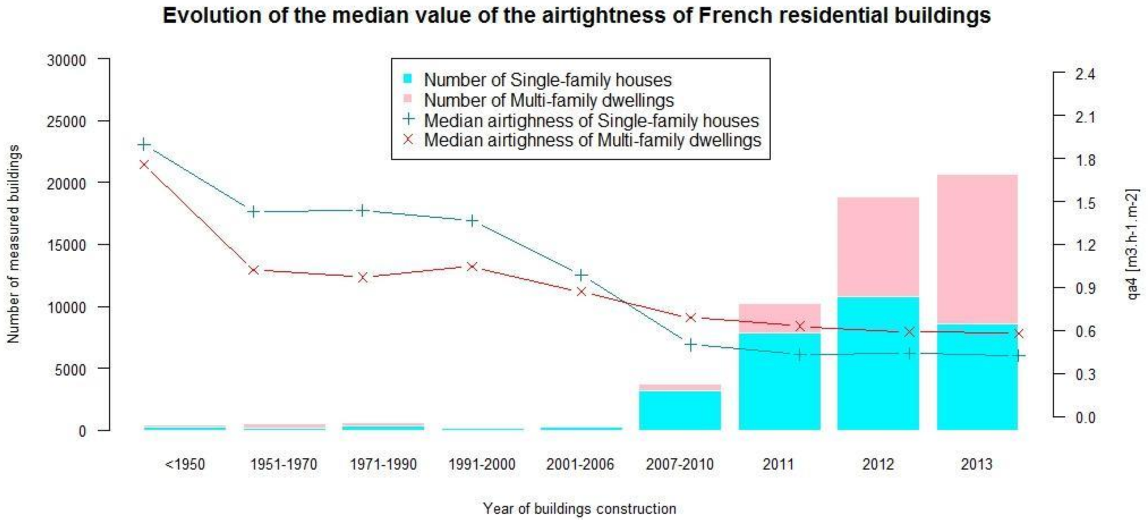


Figure 8: Evolution of the airtightness envelope measurements and their results for residential buildings

Those requirements have an impact on the measurements performed in France. First, the number of airtightness measurements performed on residential buildings is growing fast since 2007 (which corresponds to the beginning of the BBC-Effinergie label): from a few dozens of buildings tested each year before 2007 to about 20,600 in 2013. Secondly, buildings are becoming more and more airtight. Before 2007, measured multi-family dwellings were more airtight than single-family houses. But this does not hold for all buildings: in the sample of those years, more than 75% multi-family dwellings have been tested in order to comply with a certification, whereas less than 25% of the measured single-family houses were candidate to any certification. Since 2007, not only the BBC-Effinergie requirements are met, but the airtightness value is also better than the maximum value: in 2013, half of the measured airtightness envelope of single-family houses is below 0.4 $m^3.h^{-1}.m^{-2}$ and below 0.6 $m^3.h^{-1}.m^{-2}$ for half of the multi-family dwellings. Nevertheless, for most of the multi-family dwellings, the measurements have been performed by sampling, and not directly on the envelope building.

3.3 Evolution of measurements performed on non-residential buildings

Neither the BBC-Effinergie label nor the RT2012 regulation require an airtightness level for non-residential building. Moreover, only 36% of non-residential buildings in the database were applying for the BBC-Effinergie label, and 45% were not applying for any other certification.

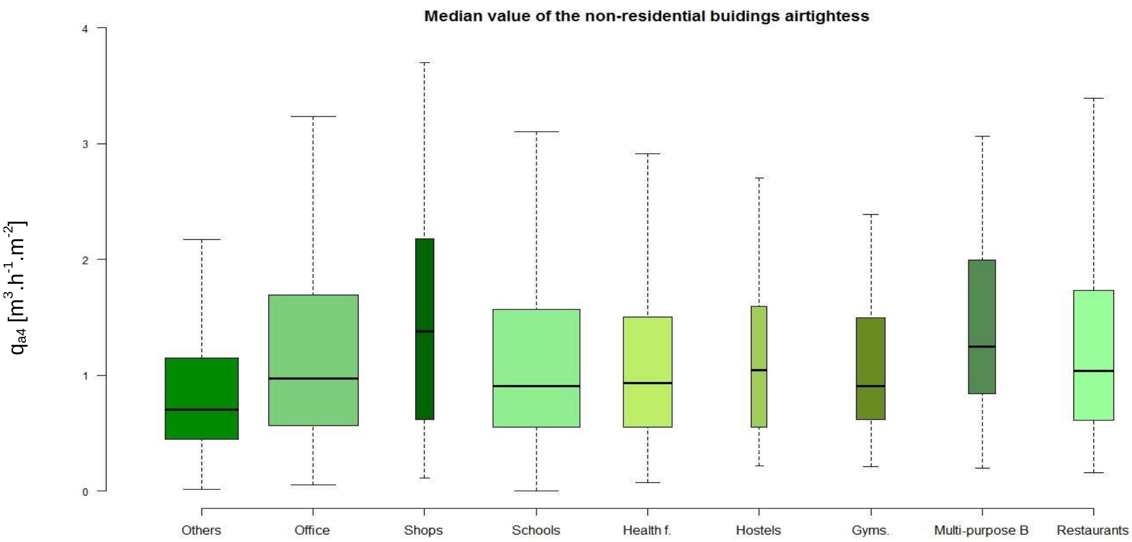


Figure 9: Non-residential buildings envelope airtightness depending on buildings' use

Therefore, depending of the use, the measured buildings envelope are more or less airtight. Nevertheless, for more than half of office buildings and schools, q_{a4} is below 1 $m^3.h^{-1}.m^{-2}$, whereas only 47% of office buildings and 55% schools were applying to a certification.

Among all those non-residential buildings, the measured volume varies quite a lot: from about 20 m^3 to several thousand cubic meters. Figure 10 illustrates that big buildings are not the least airtight.

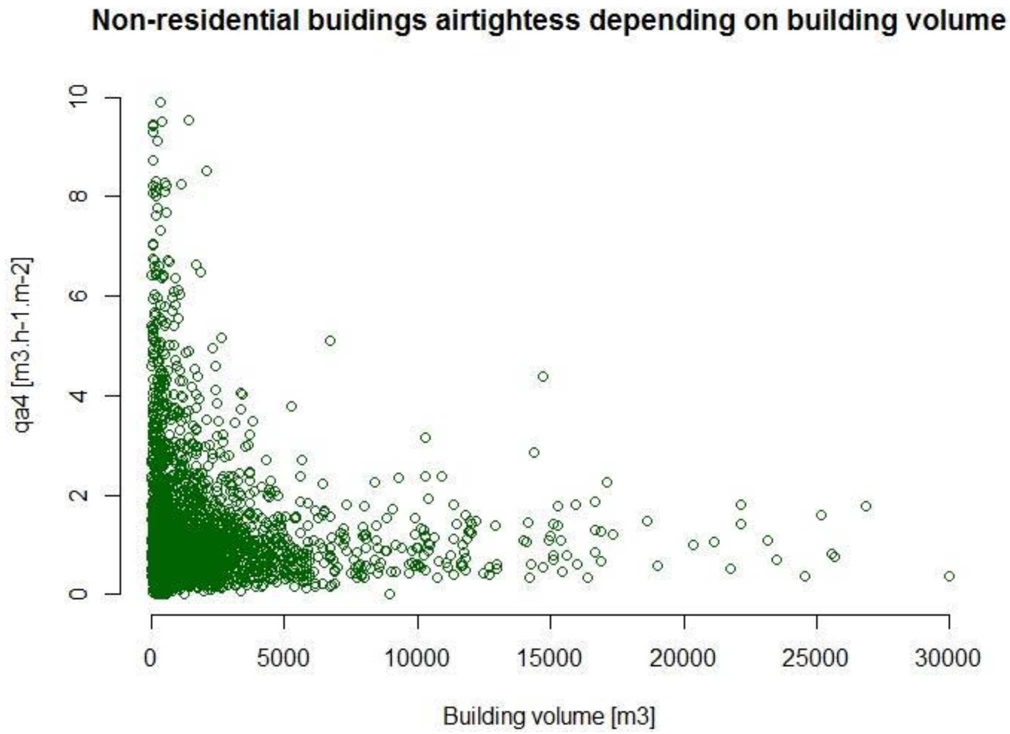


Figure 10: Non-residential buildings envelope airtightness depending on buildings' volume

Indeed, big buildings have the same leaks locations as others: doors and windows, electrical components, pipes and ducts (electrical, plumbing), which have no reason to be bigger when building volume increases.

3.4 Non bias induced by measurement season

The bias induced by thermal conditions could be evaluated through the important size of the database. Therefore, an analysis of the impact of the measurement season on the measured airtightness has been conducted. In France, standard temperatures range from -4°C to 13°C during the winter, and from 12°C to 29°C during the summer (source: meteofrance). Figure 11 presents results of airtightness measurements performed on single-family houses in 2013 (they were no particular climate conditions in 2013 in France).

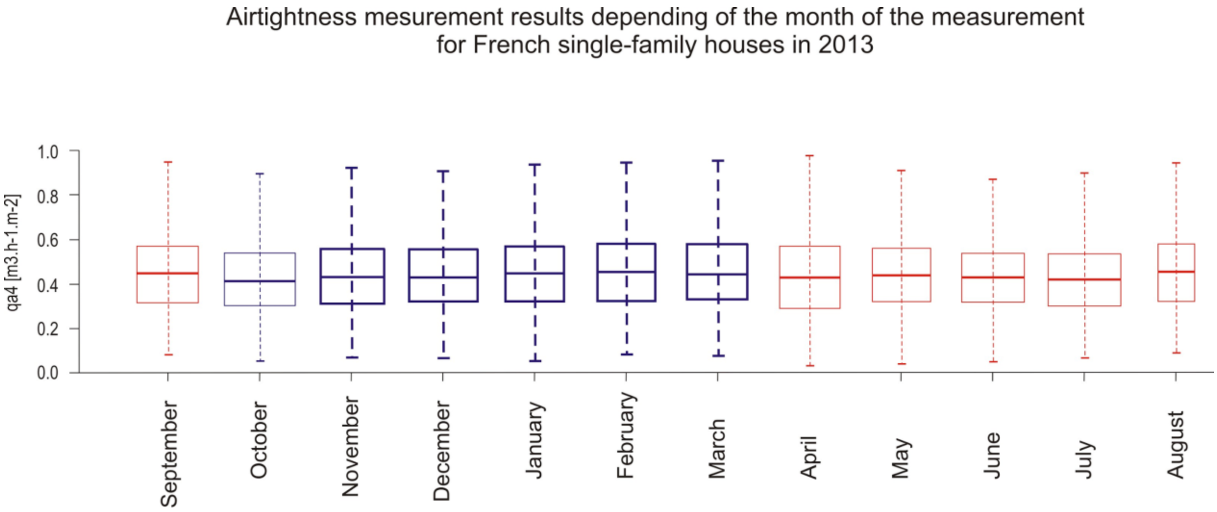


Figure 11: Single-family houses envelope airtightness depending on the measurement season (in 2013)

Those figures have been calculated from about 10,400 tests results. In those thermal conditions, the season of measurement significant impact on the result of the airtightness measurement.

4 CONSEQUENCES OF REQUIRING AIRTIGHTNESS LIMIT VALUES

As explained previously, since 2008, airtightness envelope measurements have been mostly performed on buildings applying for the BBC-Effinergie label and others certifications, especially residential buildings. Therefore, there were a target airtightness value. This calls for some questions:

- Do those buildings comply with the target value?
- What impact of those requirements on the construction sector?
- What impact on testers practices?

The following proposes partial answers to these questions.

4.1 Residential Buildings performance

The results of about 35,000 airtightness measurements performed on single-family houses have been analysed with respect to the BBC-Effinergie maximum value (q_{a4} = 0.6 $m^3.h^{-1}.m^{-2}$). Among those houses,

- 71 % were applying for the BBC-Effinergie label;

- 11% were applying for another certification;
- 18% did not have a target value (no certification).

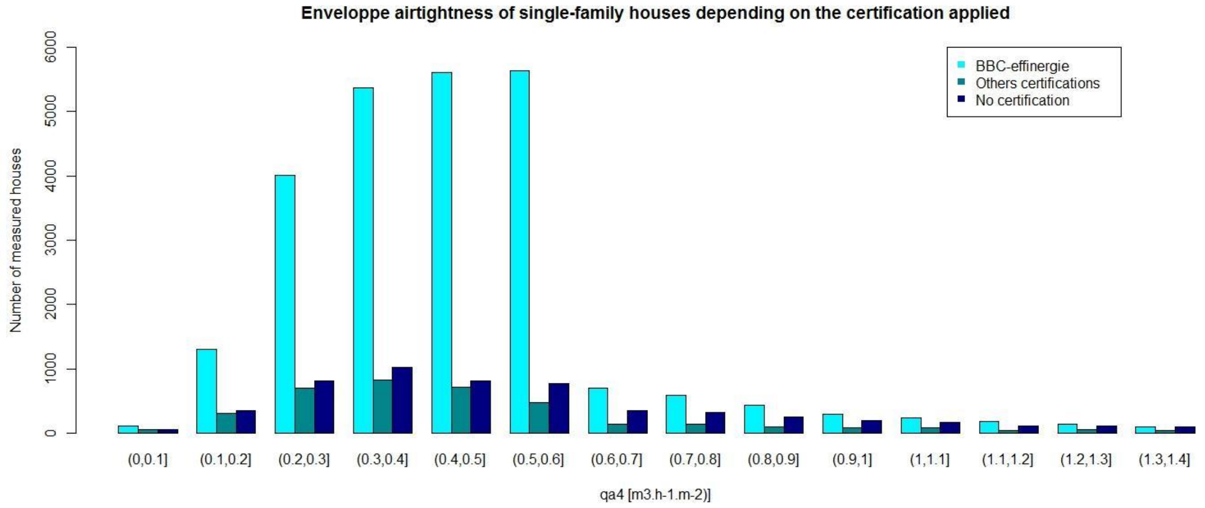


Figure 12: Single-family houses envelope airtightness depending on the certification applied

Figure 12 represents the number of measured houses in each airtightness interval. Almost all houses (87%) applying for the BBC-Effinergie label comply with the limit value, as well as 77% of houses applying for another certification. Moreover, about 70% of houses with no airtightness target meet the BBC-Effinergie label airtightness requirement. The same analysis has been conducted on multi-family dwellings, for which the BBC-Effinergie requirement is 1.0 $m^3.h^{-1}.m^{-2}$. The assessment is the same: 85% of buildings applying for BBC-Effinergie label comply with the limit value, as 69% of "others certifications". Moreover, 67% of buildings with no airtightness target meet the BBC-Effinergie label airtightness requirement.

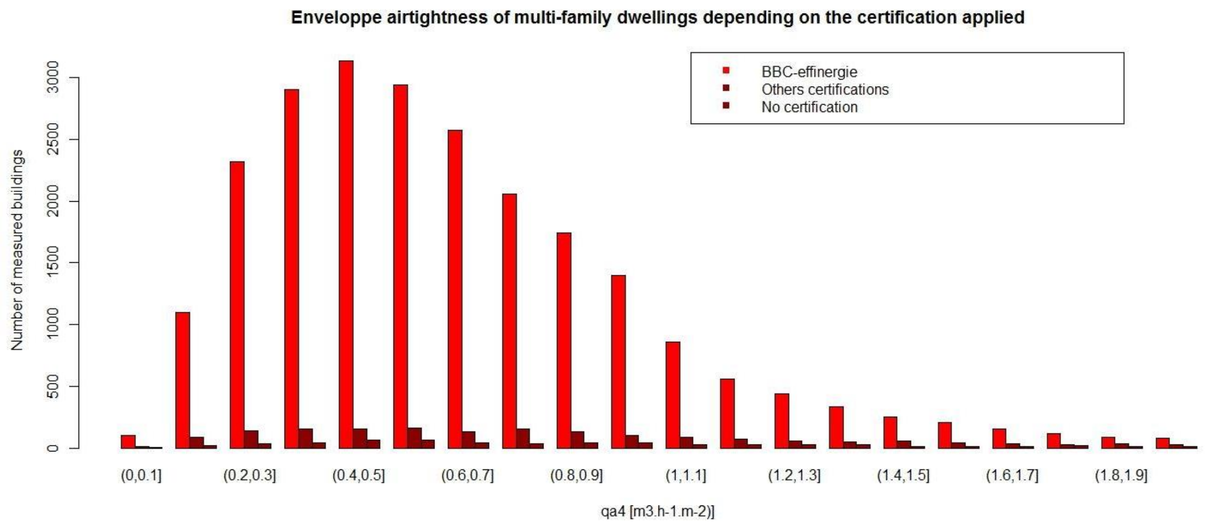


Figure 13: Multi-family dwellings envelope airtightness depending on the applied certification

Those results demonstrate that if the airtightness is taken into account during the design of the building with a reasonable target value, it is not difficult to comply with this objective.

4.2 Testers practices

The last point of this paper nuances the previous conclusion. Indeed, figure 12 shows a distinct “gap” at 0.6 $m^3.h^{-1}.m^{-2}$ for single-family houses applying for BBC-Effinergie label. Figure 14 proposes to zoom in on this “gap”.

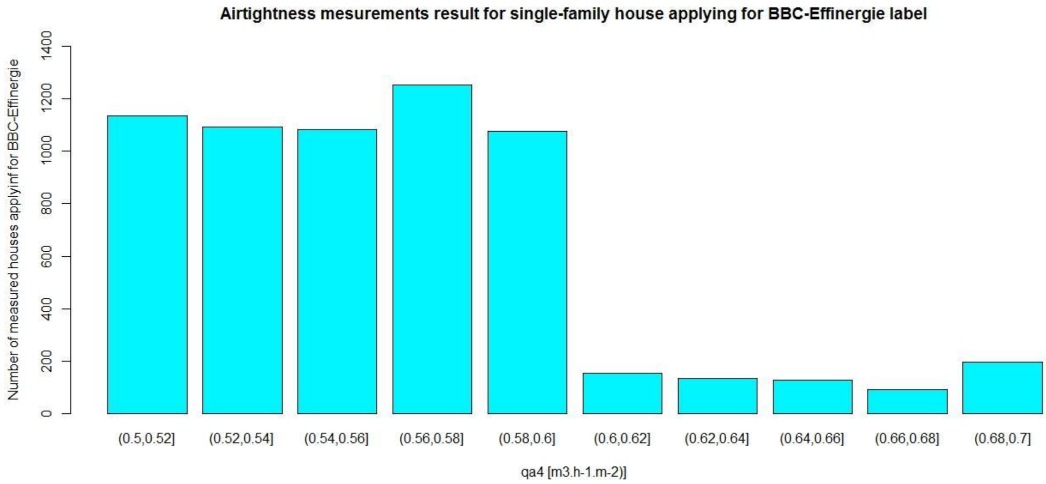


Figure 14: “Gap” of measured envelope airtightness for single-family houses applying for the BBC-Effinergie label

Even if no evidence of the cause of this gap has been provided, it seems reasonable to think that for those houses, the envelope airtightness has been modified just before the measurement. Actually, testers often perform a preliminary test, and the house owner or builder use mastic to seal off some leaks before the tester performs another measurement, in order to comply with the requirement. The pressure on testers could lead to some bad practices which will not ensure the durability of the envelope airtightness.

5 CONCLUSIONS & PERSPECTIVES

The French national airtightness database, with 65,000 measurements results as of today, is expected to be filled in with several thousands of entries each year. Many further analyses of those data should be conducted. First of all, analyses of the leaks location frequencies and of their impacts on the building envelope airtightness may lead to an improvement of the on-site work, through communications with building designers and workers. It may be completed with further analyses of the envelope airtightness, depending on main material, insulation and ventilation systems. Secondly, this database should be used to conduct more studies in order to acquire aeraulic information of building airtightness and leakage studying the *n* value. With this large database, it is now possible to develop a predictive model of building envelope airtightness, for example using principal components analysis.

Finally, several tools will be developed using those data:

- A national observatory website will be online in 2016: it will propose different analyses of the database, through various criteria as material, year of construction, label and certification, location, use of buildings. It will also geographically tag all certified testers.
- A new professional register: a new form will be proposed in order to make data more reliable, easier to process and analyse.

- A control tool with indicators calculated from individual testers statistics, with for instance, measurements results and number of measurements performed each day may be developed to make those checks easier.

6 ACKNOWLEDGEMENTS

The database is the property of the French Ministry for Ecology, sustainable development, and energy (MEDDE).
The analysis were performed by Cerema (ex- CETE de Lyon). The sole responsibility for the content of this publication lies with the authors.

7 REFERENCES

Charrier, S., Bailly, A. and Carrié, F.-R. (2015). *Buildings airtightness in France: regulatory context, control procedures, results*. QUALICHeCK, Factsheet #x.

Bailly, A., Jiang, Y., Guyot, G. and Desfougères, F. (2013). *Preliminary analysis of French buildings airtightness database.* Proceedings of the 34th AIVC - 3rd TightVent - 2nd Cool Roofs' - 1st venticool Conference, 25-26 September, Athens 2013

AIRTIGHTNESS DATA AND CHARACTERISTICS OF 752 RESIDENTIAL UNITS OF REINFORCED CONCRETE BUILDINGS IN KOREA

Jae Hun Jo[*1], Hyun kook Shin[1], Kyung Hwan Ji[1], and Myoung Souk Yeo[2]

1 Inha University
Yonghyun-Dong, Nam-Gu
Incheon, Korea

2 Seoul National University
Daehak-dong, Gwanak-gu,
Seoul, Korea
**Corresponding author: jhjo@inha.ac.kr*

ABSTRACT

This paper presents airtightness data measured for about 752 units of high-rise reinforced concrete buildings (apartment buildings) that have been recently constructed within five years in Korea. Target buildings were mainly constructed by using reinforced concrete walls/floors, and dry/wet walls were installed between units. Airtightness data of residential units were analysed based on values of ACH50 and air permeability. To identify the airtightness characteristics of reinforced concrete residential buildings, the measured airtightness data were analysed in terms of the dwelling unit sizes, wall types between units, and flow exponent n. The ACH50 values of the dwelling units were measured to range between 1.02 and 4.81, and the average value of the measured units was 2.39. It was found that the flow exponent of the measured units was 0.61 on average. The airtightness prediction model was derived by examining the correlation between a variable by component area and a variable by connection length, and performing regression analysis on these correlations. The ACH50 value tended to decrease as the floor area increased.

KEYWORDS

Airtightness, Reinforced concrete residential buildings, Fan pressurization method, Air leakage characteristic, Airtightness prediction

1 INTRODUCTION

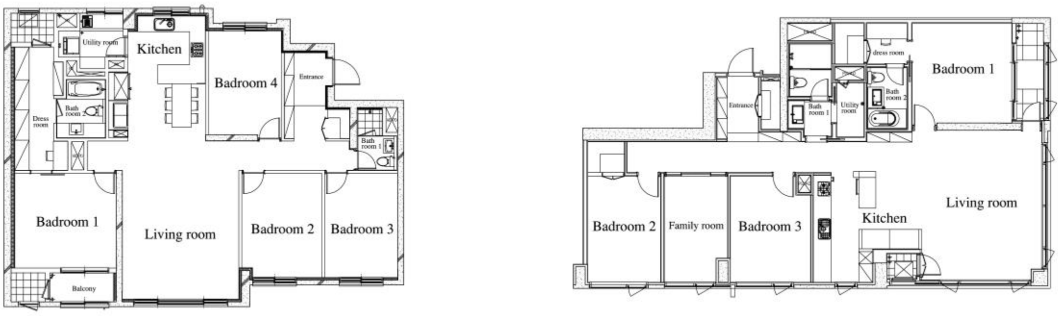
In a climatic zone similar to that in South Korea, infiltration and exfiltration in non-airtight buildings increase air-conditioning and heating energy costs and lead to problems such as condensation. For this reason, the importance of airtightness is being increasingly emphasized to reduce the energy consumption in buildings. Many countries have been proposing necessary building performance criteria regarding energy saving to make measurements in the completion step mandatory (Sherman et al. 2004, Chan 2013). Furthermore, the verification and evaluation processes on airtightness performance, which are the most crucial factors in terms of criteria for saving energy in newly constructed buildings, are being established and institutionalized. However, in Asian countries other than Japan (Yoshino 2008), an institutional base for building airtightness has either been absent or insufficient. Accordingly, studies on measuring airtightness of detached houses are being widely conducted in other countries, but it is difficult to find studies or data about the airtightness of high-rise residential buildings (i.e., apartments) of the type mainly used as residences in major Asian cities where high-density development is ongoing. This study presented the airtightness data of 752 residential units of four apartment complexes in Korea, to provide baseline data for establishing airtightness criteria for dwellings such as apartments. Airtightness of a unit was evaluated based on the ACH50 and air permeability values, and the correlation between them was analyzed. In addition, multiple regression analysis on the areas and the connection lengths, which are components of the building envelope, was performed using measurement data in order to derive an airtightness prediction model.

2 AIRTIGHTNESS MEASUREMENT SUMMARY

Buildings A to D, which were measurement targets, were constructed as high-rise residential buildings between 2010 and 2015 in Korea, and have a reinforced concrete structure. Airtightness of these buildings was measured based on Method B described in ISO9972 (2006) when all the main components of the envelope such as windows, doors, and facilities were installed, but before building completion. The test buildings are briefly described in Table 1. All the apartment complexes have structures of reinforced concrete. In terms of a building plan, in the tower type, there is a core mainly at the center, surrounded by units. In contrast, parts of A and D apartment complexes were of the plate-type building plan. As for the envelope, the A to D apartment complexes were considered to be of the punched window type. Regarding partition walls between units, wetwall construction (concrete wall) were used in the A and D apartment complexes and drywall construction in the B and C apartment complexes. Examples of typical floor plan in apartment complex are presented in Fig. 1.

Table 1: Test Building Summaries

Classification	Apartment A	Apartment B	Apartment C	Apartment D
Construction	Reinforced Concrete Structure, Flat Slab			
Building use	Residentials (apartment)			
Exterior wall type	Punched Window Type			
Number of stories	2 basements, 23~32 stories	2 basements, 12~28 stories	2 basements, 11~33 stories	2 basements, 25~32 stories
Height (m)	104.1 m	96.7 m	108.8 m	105.7m
Number of units	1401 units	1014 units	845 units	1861 units
Number of test units	210 units	148 units	129 units	265 units
View				



124m^2 type floor plan 129m^2 type floor plan

Figure 1: Examples of Floor Plan

Unit planning characteristics for high-rise residential buildings with structures of reinforced concrete are indicated in Table 2. The results from analyzing the envelopes of housing units based on their enclosed area, shows that the area of the ceiling and floor accounted for the largest part of the envelope, followed by that of the walls and windows. In each unit, there was one entrance door, and a louver in utility room containing the air conditioner. In terms of ventilation penetrations, a heat recovery ventilation system with 4~6 air inlets/outlets was installed in all the units, and one air exhaust vent was installed in kitchen and two installed in each bathroom. The area of the unit envelope changed significantly according to the unit size, but the number of mechanical penetrations for vents and drains was almost the same.

Table 2: Characteristics of the Target Apartment Units

Classification	Material or method	Number	Area Rate (%)	Apartment
Wall	Gypsum board	-	0	Apartment A , D
	Concrete	-	27 ~ 32	
	Gypsum board	-	4 ~ 15	Apartment B , C
	Concrete	-	13 ~ 24	
Floor/Ceiling (Slabs)	Concrete	-	30 ~ 35	All
Windows	PVC framed window	5 ~ 8	5 ~ 12	All
Entrance door	Steel door	1	0.3 ~ 0.8	All
Louver in utility room	Aluminium louver	1	0.4 ~ 0.7	All
Air/Pipe duct area	Gypsum board	3 ~7	4 ~ 11	All
Hear recovery ventilation	4~6 air inlets/outlets	1	-	All
Exhaust vents	Kitchen	1	-	All
	2 Bathrooms	2	-	All

Airtightness of the housing units was measured in accordance with measurement conditions proposed in ISO9972, as shown in Table 3. In contrast to detached houses, apartments have envelopes directly facing the exterior air, neighbouring units, and corridor, as well as floors and ceilings adjoining the upper and lower units. Thus, because adjacent units might affect the airtightness of the target units, the windows and entrance doors of the adjacent units were kept open to assume outdoor conditions. Openings such as doors and windows connected to the outside of target units were closed; whereas interior doors were kept open, to make a single zone. In addition, the various vents such as air inlets/ outlets and exhaust vents connected to the outside were all sealed.

Table 3: Test Conditions

Classification	Test Condition
Test space	To create a single zone
Zone height × indoor/outdoor air temperature difference	≤ 250 m K
Wind speed (near the ground)	≤ 6 m/s
Outdoor temperature	5 ~ 35 °C
The range of the induced pressure difference	10 ~ 60 Pa
Increments of induced pressure differences	5 ~ 10 Pa
Accuracy	± 5 %

3 AIRTIGHTNESS RESULTS

To obtain the airtightness of high-rise residential buildings with reinforced concrete structure, the airtightness of 752 units was measured. Six to twelve people were deployed in each apartment complex, including apartment A (210 units), apartment B (148 units), apartment C (129 units), and apartment D (265 units) to measure airtightness for four to six months. All the windows and doors of the adjacent units, as well as the windows of common corridors and stairway doors, were kept open to prevent negative effects on these units.

3.1 Ach50 and Air Permeability

To compare the airtightness of the target units, measurement results were analyzed based on the values of ACH50 and air permeability, which are used to represent airtightness. Fig. 2 and 4 present frequency distribution graphs (i.e., histograms), which indicate the number of relevant units by numerically classifying the values of ACH50 and air permeability measured. Figures 3 and 5 indicate measurement and mean values of each apartment complex in quartiles. It was found that the ACH50 values were between 1.02 and 4.81 (mean = 2.39). The ACH50 values were between 1.50 and 4.81 (mean = 2.85) for apartment A, between 1.41 and 3.60 (mean = 2.31) for apartment B, between 1.12 and 3.79 (mean = 2.48) for apartment C, and between 1.02 and 3.80 (mean = 2.02) for apartment D. These results indicate that apartment D was the most airtight and that apartment A was the least airtight. Given that apartments A to D all had envelopes of the punched window type, and that they were all recently constructed by the same construction company, we expected that the airtightness would also be similar. However, it seems that differences in window shapes and household partition (interior) walls affected the results. Moreover, because the value of ACH50 decreases as the floor area increases (as reported in previous studies), the ACH50 values of apartments A and D, which included relatively many small units, were higher than those of apartments B and C. It was also found that the value of air permeability was generally between 0.98 and 4.59 $m^3/h \cdot m^2$ (mean = 2.29). The value of air permeability was 1.39 and 4.59 $m^3/h \cdot m^2$ (mean = 2.70) for apartment A, 1.39 and 3.51 $m^3/h \cdot m^2$ (mean = 2.25) for apartment B, 1.11 and 3.57 $m^3/h \cdot m^2$ (mean = 2.42) for apartment C, AND 0.98 and 3.67 $m^3/h \cdot m^2$ (mean = 1.94) for apartment D.

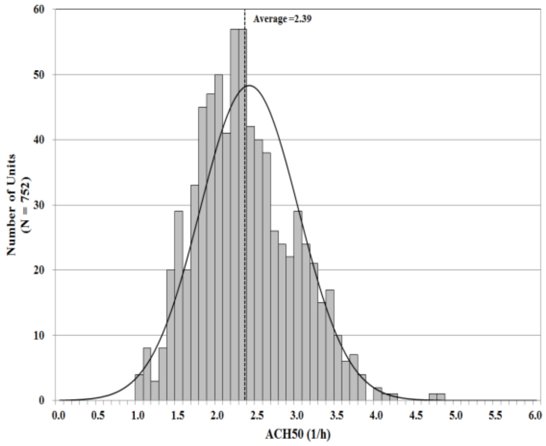


Figure 2: ACH50 Distribution

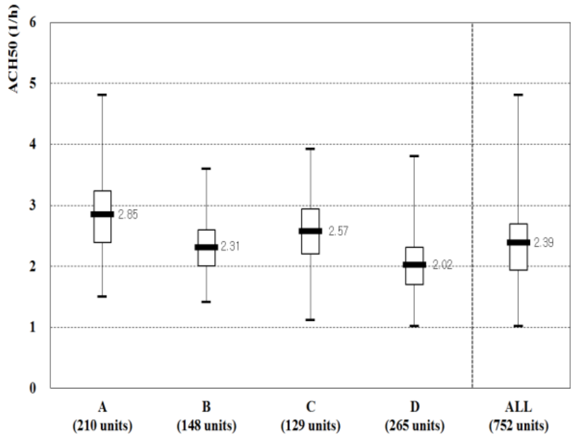


Figure 3: ACH50 Results

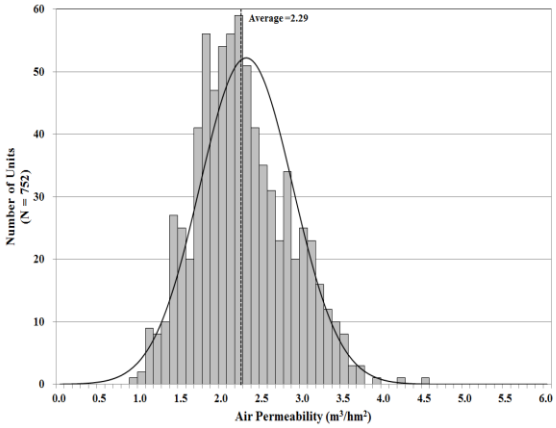


Figure 4: Air Permeability Distribution

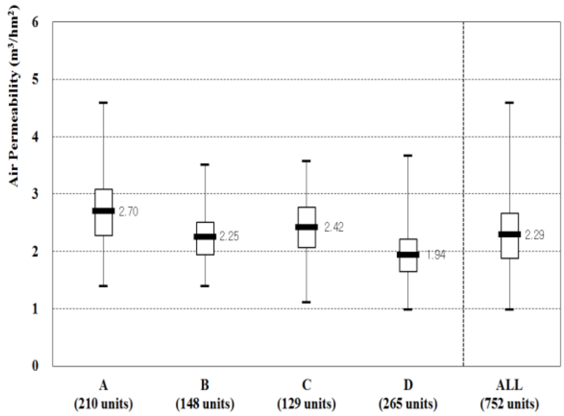


Figure 5: Air Permeability Results

To identify a correlation between ACH50 and air permeability, the two values were analyzed using a scatter diagram based on the analytic result shown above. As a result, the distribution shown in Fig. 6 was found, and the ratio of ACH50 to air permeability was close to 1.0. The volume and envelope area of the measured units are similar, and indeed, a surface to volume (S/V) ratio of the target units was verified to be between 0.965 and 1.100 (i.e., close to '1').

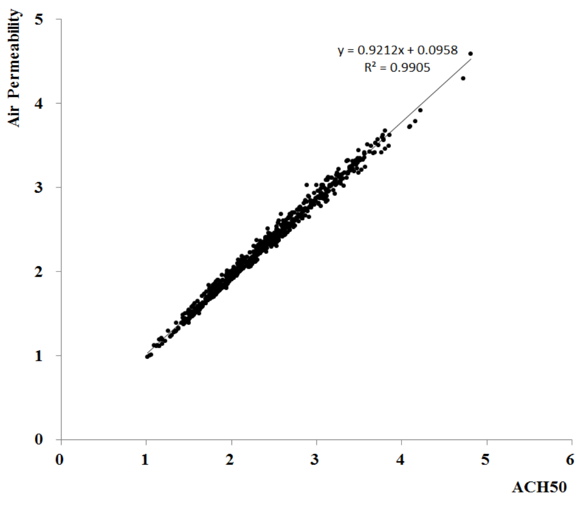


Figure 6: Relationship between ACH50 and Air Permeability

3.2 Airtightness Characteristics

To analyze the airtightness characteristics of housing units, airtightness of the target units of apartments A to D were examined according to the floor area and types of partition walls. To analyze the relationship between the floor area (size) and airtightness of a housing unit, airtightness of the target units was classified according to the floor area. ACH50 was used to represent the result of analyzing airtightness according to floor area, and the measurement data is shown in Fig. 7. The result of examining airtightness according to the area of housing units of each complex, based on one way ANOVA analysis, is indicated in Table 4. By calculating mean values of each floor plan and representing them using a trend line, it was found that the value of ACH50 tended to decrease as the floor area increased. This result indicates that the value of airtightness decreases as the floor area increases.

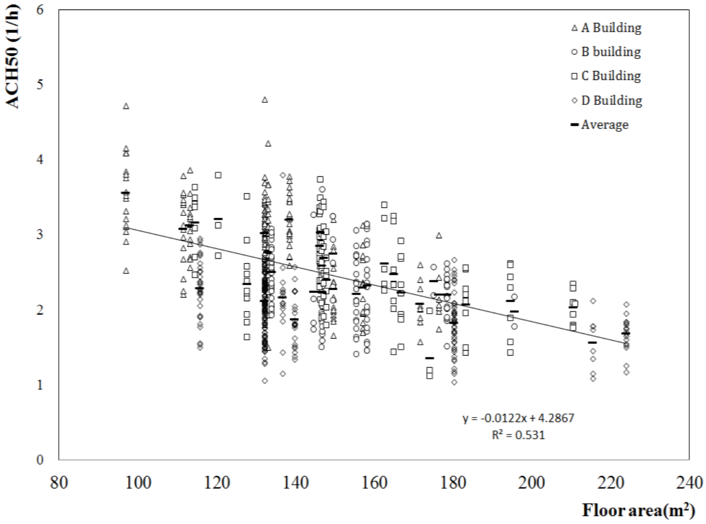


Figure 7: ACH50 According to the Floor area (Unit Sizes)

Table 4: One-way ANOVA Result for Airtightness for Each Unit Area

Apartment		Range(m^2)	Number of Units	Average of ACH50	Standard deviation	F value	Significance Probability
A	A1	80 ~ 100	17	3.55	0.546204	33.55711938	0.0000000000
	A2	100 ~ 120	29	3.09	0.425938		
	A3	120 ~ 140	114	2.95	0.502706		
	A4	140 ~ 160	34	2.28	0.433692		
	A5	160 ~ 180	16	2.13	0.348787		
	Sum	80 ~ 180	210	2.85	0.606387	-	-
B	B1	120 ~ 140	28	2.50	0.346872	2.677146627	0.0493836702
	B2	140 ~ 160	94	2.28	0.497214		
	B3	160 ~ 180	24	2.21	0.268802		
	B4	180 ~ 200	2	1.98	0.276479		
	Sum	120 ~ 200	148	2.31	0.448270	-	-
C	C1	100 ~ 120	7	3.16	0.436577	8.982701583	0.00000026811
	C2	120 ~ 140	15	2.51	0.608864		
	C3	140 ~ 160	51	2.71	0.457660		
	C4	160 ~ 180	31	2.28	0.615012		
	C5	180 ~ 200	15	2.09	0.439762		
	C6	200 ~ 220	10	2.03	0.223355		
	Sum	100 ~ 220	129	2.48	0.577864	-	-
D	D1	100 ~ 120	31	2.28	0.401743	11.48879037	0.00000001328
	D2	120 ~ 140	164	2.08	0.450523		
	D3	180 ~ 200	45	1.82	0.393204		
	D4	200 ~ 220	8	1.56	0.356926		
	D5	220 ~ 240	17	1.68	0.229318		
	Sum	100 ~ 2400	265	2.02	0.455082	-	-

To analyze the characteristics of cracks where exfiltration occurs, the flow exponent n was examined as shown in Fig. 8. The n value refers to a property value of a crack, and is irregular when the shape of the crack changes, according to the size of the pressure difference. The n value of typical buildings is between 0.6 and 0.8, and the average n value was 0.60 for apartment A, 0.62 for apartment B, 0.61 for apartment C, and 0.61 for apartment D. The mean n value of all target units was found to be 0.61, and 50% of all the n values were between 0.59 and 0.62. In the work of Chan et al. (2013) the n value was reported to be 0.65, whereas in the work of Derek Sinnott et al. (2012) it was reported to be 0.64. This means that apartments constructed with reinforced concrete walls have smaller cracks in air leakage area than wooden detached houses.

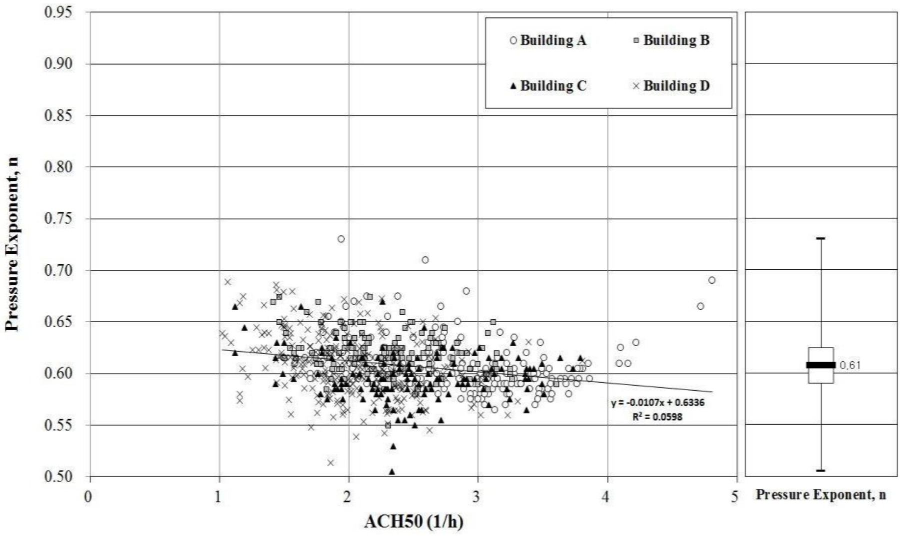


Figure 8: Distribution of Flow Exponent and ACH50 for Each Apartment

4 AIRTIGHTNESS PREDICTION MODEL

Among previous studies about predicting airtightness, Reinhold and Sonderegger (1983) presented the geometric information on buildings and the main elements of detached houses as variables to estimate airtightness. Montoya et al. (2010) derived a regression model by applying factors such as the structure type, floor area, building age, number of stories, insulation type, and heating system and Pan Wei (2010) analyzed a correlation of factors such as the building method, dwelling type, management context, and design target. In this study, we established variables regarding envelope components by classifying them into factors related to area, and factors related to joint length (connection length between the components), to derive a prediction model for airtightness of a housing unit. The airtightness prediction model was derived through regression analysis of the ACH50 value representing airtightness. Based on the analytic result of the envelope components of the housing units, variables were classified according to the area and joint length of the envelope components. Variables by area were classified into the slab, wetwall, drywall, AD/PD drywall, windows, entrance door, and louver in utility room. The variables by joint length were classified into dry and dry wall, wet and wet wall, dry and wet wall, wall and window, AD/PD room walls, wall and louver, and wall and the entrance door. Table 5 represents symbols and explanations for the variables.

Table 5: Variable Symbols According to the Area and the Joint Length Elements

Variable Symbol	Explanations
Aslab	The area of the floor and the ceiling
Asidewet	The area of wetwalls
Asidedry	The area of drywalls
Awin	The area of windows
AADPD	The area of Air Duct/Pipe Duct
Alouver	The area of louver
Adoor	The area of entrance door
Ldrydry	The length of the joint between two drywalls
Lwetwet	The length of the joint between two wetwalls
Ldrywet	The length of the joint between the drywall and wetwall
Lwin	The length of the joint between wall and window
LADPD	The length of the joint between walls for air duct/pipe duct room
Llouver	The length of the joint between wall and louver
Ldoor	The length of the joint between wall and entrance door

4.1 Multiple Regression Analysis: Area

Correlation between the variables was analyzed using a scatter diagram and the results indicated that the slab area and window area exhibited the most linear relationship and that they have a closer correlation with airtightness than do the other variables. It was also found that the slab area showed a highly negative correlation (explanation power coefficient R^2 = 0.61), followed by the window area, which showed a relatively high correlation compared to other variables. Through multiple regression analysis of the ACH50 value representing airtightness and area by factor, a regression model was derived, as shown in Eq. 1.

$$ACH50 = 4.654 - 0.007 \times Aslab + 0.004 \times Asidedry \quad (1)$$

In the regression model, the ACH50 value decreases as the slab area increases, whereas the ACH50 increases as the area of the drywall increases. As shown in the correlation analysis results above, the slab area has the most significant effect, followed by the area of drywall.

Standardized path coefficients of the slab area and drywall area were found to be –0.801 and 0.180, respectively.

4.2 Multiple Regression Analysis: Joint Length

In the scatter diagram, only the joint length of windows and doors exhibited a linear relationship with airtightness, whereas other variables did not. The R^2 value of windows and doors was 0.32 and showed a negative correlation. The result of analyzing correlation coefficients also showed that only the joint length of windows and doors had a correlation within the significance probability of 5%. By analyzing the correlation based on the scatter diagram and correlation coefficients, a correlation between windows and doors and airtightness was verified. Finally, a regression model (Eq. 2) was derived by performing multiple regression analysis of the ACH50 value representing airtightness and joint length by factor

$$ACH50 = 3.136 - 0.014 \times Lwin + 0.035 \times Ldrydry + 0.121 \times Llouver + 0.007 \times LADPD \quad (2)$$

In this model, the ACH 50 value decreases as the joint length of windows and doors and AD/PD drywalls increases, whereas the ACH50 increases as the joint length of the dry and wall and that of the wall and louver increase. The standardized path coefficient between variables was found to be - 0.544 for windows and doors, 0.258 for the dry and dry wall, 0.308 for the wall and louver, and - 0.216 for the joint length of the AD/PD dry walls. The explanation coefficient R^2 value representing goodness of fit of the prediction model was 0.45 for the prediction model regarding joint length, with low explanation power. On the other hand, the R^2 value for the prediction model regarding the area was 0.64 with greater explanation power. Based on this result, it seems that Eq. 1, the prediction model based on to the component area, is more appropriate than that based on the joint length of envelope components, and that this model could be used effectively to estimate airtightness in apartments where airtightness cannot be measured.

5 CONCLUSIONS

The airtightness of 752 units was measured to provide data on the airtightness of apartment constructed with reinforced concrete, and to analyze their airtightness characteristics. Multiple regression analysis was performed on the area and the length factors, which are components of the envelope, to derive the prediction model. By measuring the airtightness of 752 apartment units with reinforced concrete structure, it was verified that the ACH50 values were between 1.02 and 4.81 and that the overall mean ACH50 was 2.39. The air permeability value was between 0.98 and 4.59 $m^3/h{\cdot}m^2$ and the overall mean air permeability was 2.29 $m^3/h{\cdot}m^2$. The result of analyzing airtightness by classifying target units by floor area (size), showed that the ACH50 value decreased as the floor area increased. This result was due to the tendency toward an increase in the ratio of the volume of the housing unit, to the floor area. Multiple regression analysis was performed based on the measured airtightness data to derive an airtightness prediction model. The airtightness predictions according to the area of each component of the envelope, was found to be more accurate than that according to the joint length between components.

6 ACKNOWLEDGEMENTS

This work was supported by the National Research Foundation of Korea (NRF) grant funded by the Korea government (MSIP) (No. 2014051236).

7 REFERENCES

Francois R.C. and Bernd R. (2008). An Overview of National Trends in Envelope and Ductwork Airtightness. AIVC VIP29.

Sherman, M.H. and Rengie, C. (2004) Building Airtightness: Research and Practice. LBL Report No. 53356, LBNL.

Chan, W.R. (2013). Analysis of Air Leakage Measurements of US Houses. *Energy and Building*, 66, 616-625.

Yoshino, H. (2008) The current of air-tightness and ventilation system in Houses of Japan. Proceedings of the 29th AIVC Conference.

Sinnott, D. (2012). Air-tightness Field Data for Dwellings in Ireland. *Energy and Building*, 51, 269-275.

Kalamees, T. (2007). Air Tightness and Air Leakages New Lightweight Single-Family Detached Houses in Estonia. *Building and Environment*, 42, 2369-2377.

Reinhold, C. and Sonderegger, R. (1983). Component leakage areas in residential buildings, in: 4th AIVC Conference.

Pan, W. (2010). Relationships between Air-Tightness and its Influencing Factors of Post-2006 New-build Dwellings in the UK. *Building and Environment*, 45, 2387-2399.

Montoya, M.I. (2010). Air Leakage in Catalan Dwelling: Developing an Airtightness Model and Leakage Airflow Predictions. *Building and Environment*, 45, 1458-1469.

Sherman, M.H. (1998). The Use of Blower Door Data, LBL Report No. 35173.

ISO9972. (2006). *Thermal performance of buildings - Determination of air permeability of buildings - Fan pressurization method*.

AIRTIGHTNESS AND INDOOR AIR QUALITY IN SUBSIDISED HOUSING IN SPAIN

Jessica Fernández-Agüera[*1], Juan José Sendra[2], Rafael Suárez[2], Samuel Domínguez-Amarillo[2] and Ignacio Oteiza[1]

1 Instituto de Ciencias de la Construcción Eduardo Torroja (CSIC)
Av. Serrano Galvache, 4,
28033 Madrid, Spain
** jfernandezaguera@us.es*

2 Instituto Universitario de Arquitectura y Ciencias de la Construcción, University of Seville,
Av. Reina Mercedes, 2,
41012 Seville, Spain

ABSTRACT

Over three million subsidised dwellings were built in Spain between 1940 and 1980. Most of these buildings are now obsolete and fail to comply with thermal comfort and ventilation standards. A building's existing energy performance, including its airtightness, should be determined prior to conducting low-energy refurbishment, for those factors, particularly the latter, impact thermal comfort, energy demand and indoor air quality (IAQ) fairly heavily.

This paper introduces a study on airtightness and IAQ in subsidised housing built in Spain in the aforementioned 40-year period. Airtightness and CO_2 measurements taken in 2014-2015 in six units in multi-dwelling buildings, three each in Seville and Madrid, are described.

The results show that in a building in Madrid, the number of air changes per hour at a pressure of 50 Pa (n_{50}) ranges from 3.2 to 8.3. The winter time CO_2 concentration in bedrooms is 1900 ppm and in living rooms 1400 ppm, with peaks of 5000 ppm and 4700 ppm, respectively. The number of air changes per hour at 50 Pa (n_{50}) in Seville, ranges from 5.0 to 9.5. The winter time CO_2 concentration in bedrooms is 1500 ppm and in living rooms 800 ppm, with peaks of 6000ppm and 4000 ppm, respectively. In the summer, however, when users tend to open windows at night primarily to let in cooler air, the CO_2 concentration values observed in Seville drop to 700 ppm and in Madrid to 1100 ppm.

As those values are much higher than recommended in the standards on good indoor air quality, the inference is that winter time housing ventilation must not be allowed to rest solely on users' voluntary opening of windows.

KEYWORDS

Carbon dioxide concentration, infiltration, airtightness, blower door, dwellings

1 INTRODUCTION

Given the impact of building envelope airtightness (measured as air infiltration) on energy efficiency, occupants' thermal comfort and indoor air quality, it has become a standard area of research in the United States in general and the Lawrence Berkeley National Laboratory (LBNL) in particular (Sherman et al, 2007), and in regions of Europe with cold climates (Montoya et al, 2010; Jokisalo et al; 2009, Pan, 2010). In Southern Europe, however, studies on which to base a sound analysis and interpretation of the interaction between these factors are largely lacking (d'Ambrosio et al, 2012, Pinto et al, 2011). Spain's Technical Building

Code (CTE, 2006), for instance, does not specify the airtightness to be met by building envelopes, but only the ex-factory permeability of the carpentry, i.e., irrespective of its subsequent on-site assembly.

Most of the studies on the relationship between IAQ and airtightness conducted in the US (Prince et al., 2006, Less et al., 2015) and other cold climates (Sharpe et al, 2014; Mickaël et al, 2014; McGill et al, 2015) conclude that resident behaviour is instrumental to effective ventilation and that mechanical ventilation is needed to improve the quality of indoor air. The LBNL is pioneering research on the link between indoor air quality and minimum ventilation rates (Turner et al, 2013; Turner et al, 2012; Sherman et al, 2011).

Many papers have been published on low-energy refurbishment in Spain, although the focus is primarily on improving envelope insulation (Dominguez et al, 2012; Cuerda et al, 2014), with scant attention in airtightness to its impact on indoor air quality. In light of that paucity of research, this study analysed envelope airtightness in subsidised housing built in Madrid and Seville in 1940-1980 and its relationship to indoor air quality. During much of that (post-Civil War) period, the construction industry's primary objective was to provide housing for large swathes of the population in a context in which environmental requirements were simply not envisaged. The end date, the early nineteen eighties, marked the institution of the country's earliest legislation limiting energy demand.

Moreover, as most of the housing built in the period studied has no specific ventilation system, air replenishment depends on the voluntary opening of windows in mild weather. Against that backdrop, the primary aims of this study were to characterise air tightness in a group of social housing developments erected in the period 1940-80 and assess air quality at various levels of air tightness in these dwellings in different seasons of the year.

2 METHODOLOGY

The methodology followed included:
-choosing and defining the housing units for the study sample
-taking in situ measurements, including blower door airtightness tests, smoke tests to locate air leaks and CO_2 concentration measurements to determine indoor air quality

These tasks are described in greater detail below.

2.1 Case studies

Six units in multi-dwelling buildings, three in Seville and three in Madrid, were chosen for the airtightness tests and CO_2 measurements. The sampling criterion for the units chosen was that they had to be sufficiently representative of the architectural typologies and construction solutions for façades most commonly deployed in 1940-1980. A number of user profiles were chosen, given their effect on use and occupancy. The buildings sampled also exhibited different degrees of envelope refurbishment.
Figure 1 shows the location of the units in Madrid (A, B, C) and Seville (D, E, F), while Table 1 lists the dwelling characteristics and user profiles.

Figure 1: Location of units studied in Madrid and Seville

Table 1: Case study

Case study	Quarter	Year built	Type of façade	Floor area (m^2)	Façade area (m^2)	Volume (m^3)	No. occupants	User profile	Extent of refurbish-ment
A	Manoteras	1960	1- ft brick	63	36	145	2	Couple	Basic
B	Manoteras	1976	0.5-ft brick+air chamber+partition	64	69	159	3	Couple with child	Major (outer)
C	Hispano-américa	1965	0.5-ft brick+air chamber+partition	76	45	193	4	Couple with children	Intermediate
D	Bami	1963	1- and 0.5-ft brick	63	49	205	3	Stu-dents	Basic
E	San Pablo	1965	0.5-ft brick+air chamber+partition	51	31	112	2	Couple	Basic
F	Diez Mandamientos	1964	0.5-ft brick+air chamber+partition	64	47	163	4	Couple with children	Intermediate

The clearance height in all the units ranged from 2.20 to 2.50 m, except in unit D, where it was 3.25 m.

Basic refurbishment consisted in everyday maintenance with no changes in the envelope other than repainting; intermediate in upgrading the unit and replacing the carpentry; and major in replacing the carpentry and adapting the building envelope. Major refurbishment in the unit in Madrid even involved removing the kitchen vents with on-site inspection during the work.

2.2 Fan pressure trials

The blower door test, a technique consolidated among the scientific community, was conducted in this study further to the procedure set out in Spanish and European standard UNE EN 13829:2002 (UNE-EN 13829:2002). The test was performed in all the units in the sample to determine building envelope airtightness. The findings were subsequently used to ascertain its possible effect on air quality, defined as indoor CO_2 concentration. Table 2 shows Parameters defined in UNE 13829

The test consisted in measuring the air flow when a fan was positioned on the outside door (Figure 2) to remove air out of (depressurise) or force air into (pressurise) the dwelling to a positive or negative pressure of around 50 Pa. All windows and any other outer doors in the unit were sealed and all the inner doors kept open during the test. Outdoor air consequently flowed into the dwelling across any infiltrations in the envelope. Smoke generators and infrared thermographic analysers were used to locate such air leaks. Envelope infiltrations were accurately located by this joint use of blower door trials and thermographic and smoke analyses.

Figure 2: Blower door position during airtightness trials

Table 2: Parameters defined in UNE 13829

Parameter	Name	Unit	Definition
V50	Air Leakage rate at 50 Pa	m^3/h	Air flow across the building envelope including the flow rate through joints, fissures and the superficial porousness
n50	Air change rate at 50 Pa	h^{-1}	Infiltration air flow rate per internal volume (V)
w50	Specific leakage area at 50 Pa	m/h	Relationship between the rate of infiltrated air and the usable area (S).

2.3 Monitoring environmental conditions

The environmental conditions were monitored in all the units studied with a multi-parametric data logger that recorded ambient temperature, relative humidity and air quality (with a CO_2 probe) at 10-minute intervals. Two such data loggers were positioned in each unit, one in the daytime and the other in the night time living area. The loggers had USB connectivity to download the data to a computer. Their positions in one of the units are depicted in Figures 3 and 4.

Figure 3. Plan view of unit E, showing data logger positions

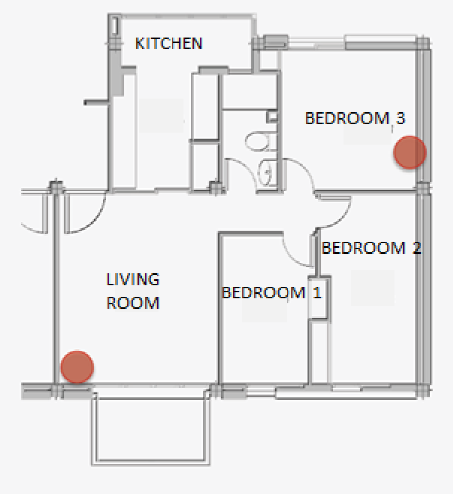
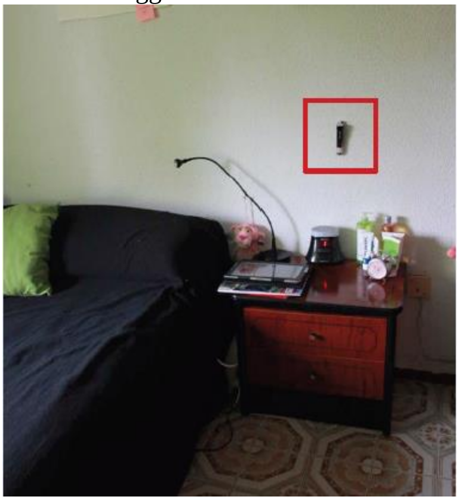


Figure 4: Data logger installed in bedroom in unit E



3 ANALYSIS AND DISCUSSION OF THE RESULTS

3.1 Airtightness measurements

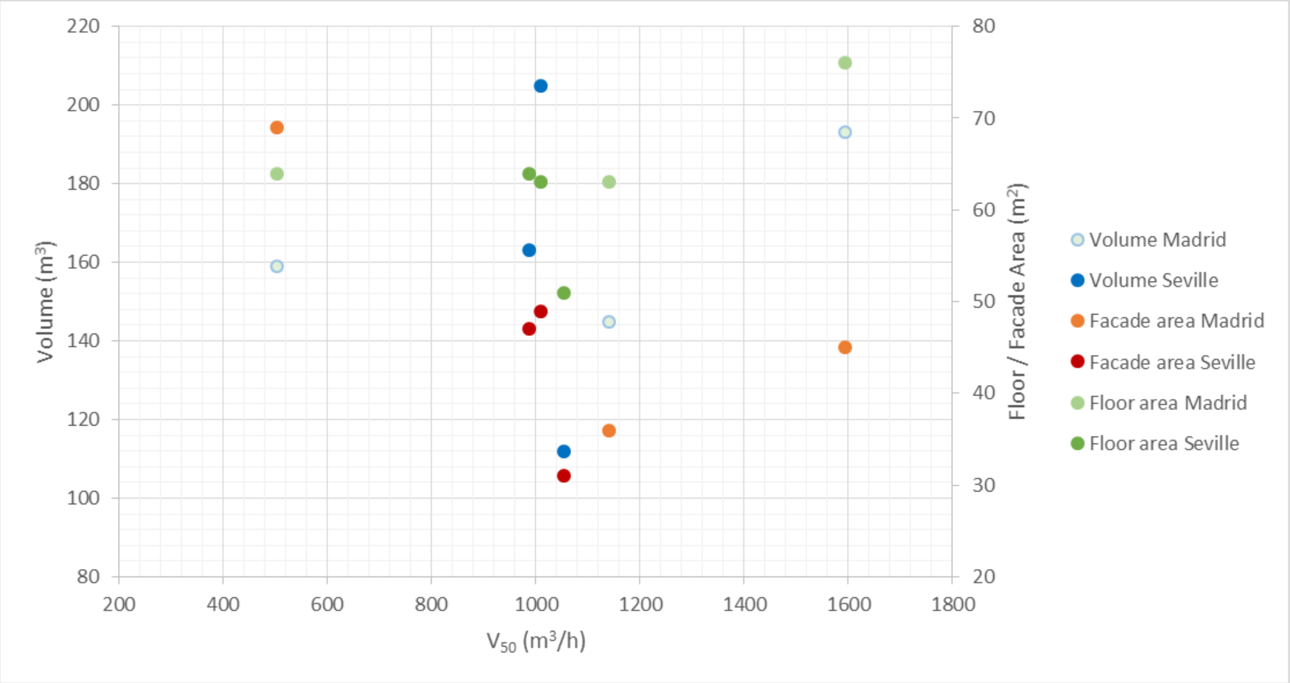
Table 3 gives the blower door trial findings for the six units studied, which are plotted against a number of parameters in Figure 5. No relationship was observed between infiltration air flow (V_{50}) and any of the other parameters: façade area, dwelling area, dwelling volume or extent of refurbishment.

In Madrid n_{50} ranged from 8.9 to 3.2 and w_{50} from 7.9 to 21. The unit that underwent the most extensive refurbishment had the lowest infiltration rate, due to the exhaustive control conducted during the works. In Seville, given the greater disparity in clearance heights, w_{50} (which ranged from 15.4 to 20.6) was deemed to be a more suitable parameter, inasmuch as most air leaks occurred at abutments between construction elements rather than on the continuous areas of the façade. Here the values were similar for the unit with basic maintenance only and the one where the windows had been replaced because workmanship during installation of the new windows went largely uncontrolled. That further to the smoke test findings most of the air leaks were located at the connection between the window and the enclosure stands as proof of the importance of on-site quality control.

Table 3: Blower door test results

Case Study	Floor area (m^2)	Facade area (m^2)	Volume (m^3)	V_{50} (m^3/h)	n_{50} (m/h)	w_{50} (h^{-1})
A	63	36	145	1140	7.9	18.1
B	64	69	159	504	3.2	7.9
C	76	45	193	1594	8.3	21.0
D	63	49	205	1010	4.9	16.0
E	51	31	112	1053	9.4	20.6
F	64	47	163	988	6.1	15.4

Figure 5: Infiltration air flow vs dwelling volume, façade area and dwelling area.



3.2 Air quality measurements

Figure 6 shows that in a typical winter month, the highest CO_2 concentrations were recorded in the most air-tight dwelling in Madrid, with mean values on the order of 2500 ppm. In all the other cases studied, the mean ranged from 1100 to 1800 ppm (absolute concentration in air).

The peak CO_2 concentration values, which ranged from 4000 to 5000 in Madrid and from 4000 to 6000 in Seville, were recorded in the bedrooms in the winter. The highest peaks were observed in the most airtight units.

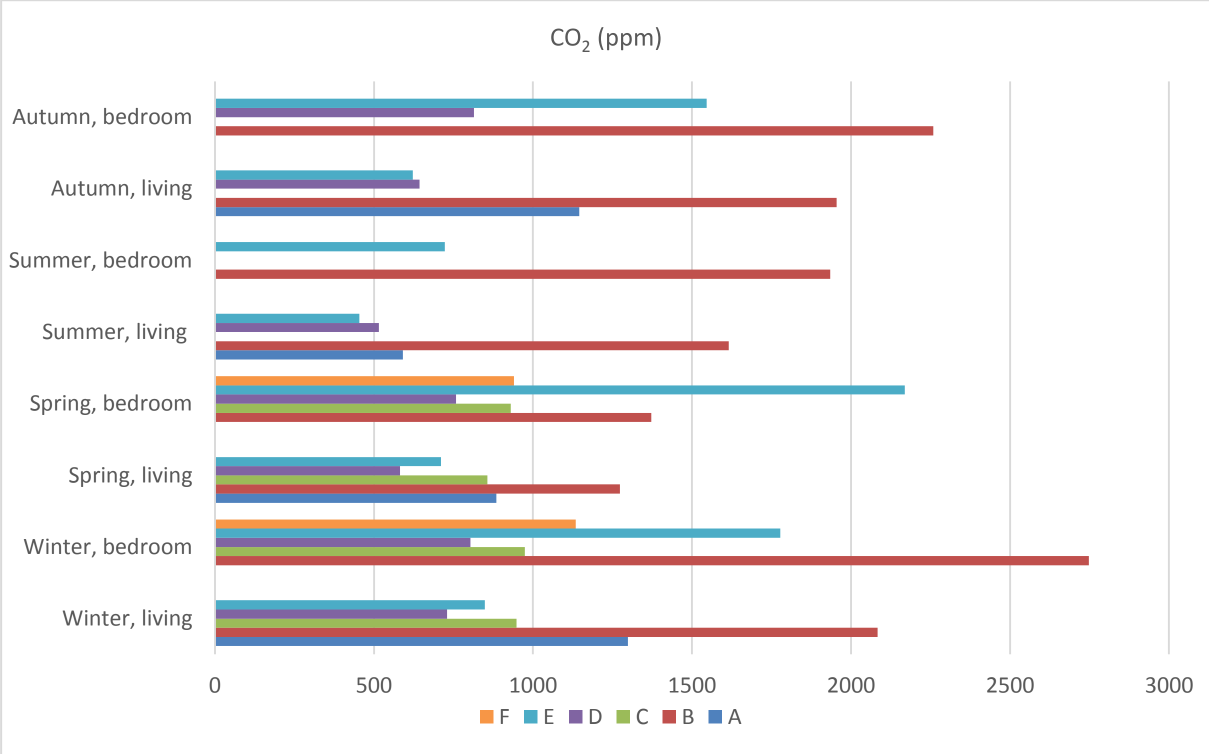
The highest living room concentrations were logged late at night, with peaks of 3000 to 4700 in Madrid and 3000 to 4000 in Seville.

Figure 6: Winter time CO_2 concentration in ppm in bedrooms occupied by two people in selected units

The CO_2 concentrations found for the dwellings are shown in Figure 7. The values recorded were lower in summer and spring than in winter and autumn. The concentration was substantially higher in the bedrooms than in the living rooms in all seasons, as would be expected given the longer periods of time, at night especially, that the former were continually occupied. The highest values in the living room were observed in the middle hours of the day as well as in the hours shortly before bedtime.

Seasonal fluctuation differed in Madrid and Seville. In the former, the variation between winter-autum and summer was approximately 30 % in the living room and 40 % in the bedroom. In Seville, the living room winter-to-summer values fluctuated by 60 % and the autumn/spring-to-summer concentrations by 30 %, while the bedroom values were twice as high in winter as in summer. These findings have to do with user-mediated ventilation (by opening windows), a more frequent practice in Seville where the climate is warmer than in Madrid.

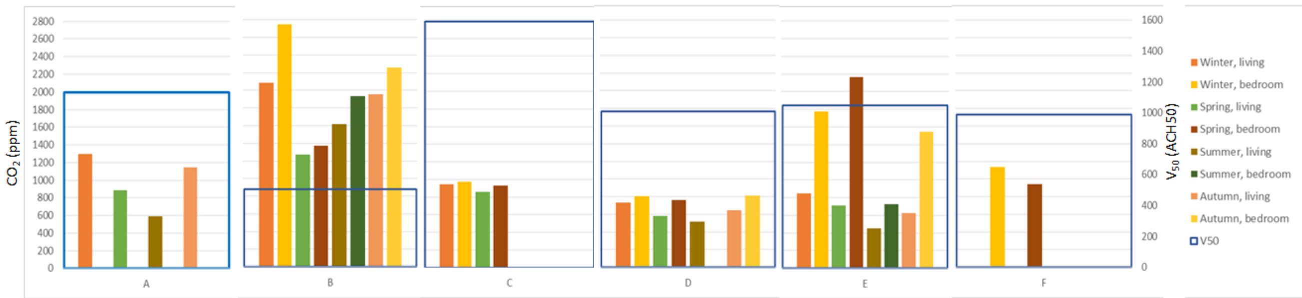
Figure 7: Mean CO_2 concentration (ppm) in all four seasons in all the units studied



3.3 Relationship between air tightness and air quality

Figure 8 shows mean CO2 concentration (ppm) vs air tightness. As noted, occupant behaviour differed in Madrid and Seville. In the former, where temperatures are usually lower, users tend to open and close windows less routinely. As a result, envelope airtightness in Madrid is related more closely to air replenishment, which is clearly insufficient most of the time, unless windows are opened. In Seville, where annual temperatures are more temperate, windows and doors are opened more frequently to replenish the air. The outcome is a more random distribution of indoor CO_2 concentrations, which depend less on airtightness, for indoor replenishment is observed to vary widely among dwellings with similar airtightness values.

Figure 8: Mean CO_2 concentration (ppm) vs air tightness in the units studied



Note: The bedroom in dwelling D was occupied by only one person.

4 CONCLUSIONS

The V_{50} values measured were apparently unrelated to dwelling area and volume, as well as to façade area. With one exception, however, w_{50} and the extent of refurbishment were found to be related. Case dwelling B (Madrid), where extensive refurbishment had been conducted, had the lowest w_{50} value, at around 7.9 h^{-1}, lower even than required by European standards. Case dwelling F (Seville), also extensively refurbished, exhibited a value of around 15.4 h^{-1}, clearly lower than in the dwellings in Madrid and Seville with scant refurbishment, where it ranged from 16 to 20.6 h^{-1}.

The V_{50} values, together with the lack of specific ventilation systems outside of user-dependent opening and closing of windows, led to low n_{50} values and a generally deficient quality of indoor air with high CO_2 concentrations, particularly in bedrooms in the winter. This was especially apparent in housing where extensive refurbishment had been performed: dwellings B (Madrid) and F (Seville) exhibited the highest CO_2 concentrations throughout the year. Concentration values were widely scattered, however, due especially (in addition to dwelling use and room occupancy) to the voluntary opening and closing of windows (particularly in Seville), which was the sole system of ventilation in the six dwellings analysed.

Users failed to perceive the deficient indoor air quality. That, in conjunction with the lack of ventilation systems in these homes, even in the ones with extensive refurbishment, gave rise to generally poor and often unhealthy indoor environments with high CO_2 concentrations and the risk of condensation damp, especially in the most air-tight dwellings. Low-energy refurbishment in housing must, then, aim not only to improve the thermal resistance of the building envelope and its airtightness, but also to ensure that ventilation systems are suitable and efficient.

5 ACKNOWLEDGEMENTS

This study was funded by the Spanish Ministry of Economy and Competitiveness under project BIA2012-39020-C02-01. One of the authors benefited from a pre-doctoral contract, reference BES-2013-063097.

6 REFERENCES

d'Ambrosio, F.R., Dell'Isola, M. and Tassini, G. (2012). Experimental analysis of air tightness in Mediterranean buildings using the fan pressurization method. Building and Environment, 53, 16-25.

Código Técnico de la Edificación, CTE, Real Decreto 314/2006: España, Real Decreto 314/2006, por el que se aprueba el Código Técnico de la Edificación, de 17 de marzo. Boletín Oficial del Estado, 28 de marzo de 2006, núm. 74, p. 11816.

Cuerda, E. et al. (2014). Facade typologies as a tool for selecting refurbishment measures for the Spanish residential building stock. Energy and Buildings, 76, 119–129.

Domínguez, S., Sendra J.J., León, A.L., Esquivias, P.M. (2012). Towards Energy Demand Reduction in Social Housing Buildings: Envelope System Optimization Strategies. Energies, 5(7), 2263-2287.

Less, B., Mullen, N., Singer, B. and Walker, I. (2015). Indoor Air Quality in 24 California Residences Designed as HighPerformance Homes. Ernest Orlando Lawrence Berkeley National Laboratory.

McGill, G., Oyedele, L. O. and McAllister, K. (2015). Case study investigation of indoor air quality in mechanically ventilated and naturally ventilated UK social housing. International Journal of Sustainable Built Environment, 4, 58-77.

Mickaël, D., Bruno, B., Valérie, C., Murielle, L., Cécile, P. and Jacques, R. (2014). Indoor air quality and comfort in seven newly built, energy-efficient houses in France. Building and Environment, 72, 173–187.

Montoya, M. I., Pastor, E., Carrié, F. R., Guyot, G. & Planas, E., (2010). Air leakage in catalan dwellings: Developing an airtightness model and leakage airflow predictions. Building and Environment, 45, 1458-1469.

Pan, W. (2010). Relationships between air-tightness and its influencing factors of post-2006 new-build dwellings in the UK. Building and Environment, 45, 2387-2399.

Pinto, M., Viegas, J. and de Freitas, V.P., Air permeability measurements of dwellings and building components in Portugal. Building and Environment, 46, 2480-2489.

Price, P. N. and Sherman, M. H. (2006).Ventilation Behavior and Household Characteristics in New California Houses, Environmental Energy Technologies Division. Ernest Orlando Lawrence Berkeley National Laboratory.

Sharpe, T., McQuillan, J., Howieson, S., Farren, P. and Tuohy, P. (2014). Research Project to Investigate Occupier Influence on Indoor Air Quality in Dwellings. Local Government and Communities, Livingston.

Sherman, M. H. and McWilliams, J. A., (2007). Air Leakage of U.S. Homes: Model Prediction. US DOE, Lawrence Berkeley National Laboratory.

Sherman, M. H., Walker, I.S. and Logue, J.M. (2011). Equivalence in Ventilation and Indoor Air Quality, Environmental Energy Technologies Division. Ernest Orlando Lawrence Berkeley National Laboratory.

Turner, J. N., Sherman, M. H. and Walker I. S. (2012) Infiltration as Ventilation: Weather Induced Dilution, Environmental Energy Technologies Division. Ernest Orlando Lawrence Berkeley National Laboratory.

Turner , W. J.N., Logue, J. M. and Wray, C. P.(2013). A combined energy and IAQ assessment of the potential value of commissioning residential mechanical ventilation systems. Building and Environment, 60, 194-201.

UNE-EN 13829:2002. Thermal performance of buildings. Determination of air permeability of buildings. Fan pressurization method. Madrid: AENOR 2002.

Jokisalo J, Kurnitski J, Korpi M, Kalamees T and Vinha J. (2009). Building leakage, infiltration, and energy performance analyses for Finnish detached houses. Building and Environment, 44, 377-387.

EVALUATION TOOL OF CLIMATE POTENTIAL FOR VENTILATIVE COOLING

Annamaria Belleri*[1], Theofanis Psomas[2], Per Heiselberg[2]

1 EURAC research
Druso 1
Bolzano, Italy
**Corresponding author:*
annamaria.belleri@eurac.edu

2 Aalborg University
Sofiendalsvej 11
Aalborg, Denmark

ABSTRACT

The new initiatives and regulations towards nearly zero energy buildings forces designers to exploit the cooling potential of the climate to reduce the overheating occurrence and to improve thermal comfort indoors. Climate analysis is particularly useful at early design stages to support decision making towards cost-effective passive cooling solution e.g. ventilative cooling. As buildings with different use patterns, envelope characteristics and internal loads level do not follow equally the external climate condition, the climate analysis cannot abstract from building characteristics and use.

Within IEA Annex 62 project, national experts are working on the development of a climate evaluation tool, which aims at assessing the potential of ventilative cooling by taking into account also building envelope thermal properties, internal gains and ventilation needs.

The analysis is based on a single-zone thermal model applied to user-input climatic data on hourly basis. The tool identifies the percentage of hours when natural ventilation can be exploited to assure minimum air change rates required by state of the art research, standards and regulations and the percentage of hours when direct ventilative cooling is useful to reduce overheating risk and improve thermal comfort. The tool also assesses the night cooling potential and highlights other useful climate performance indicators such as the day-night temperature swing.

Furthermore, the analysis method has also been devised to provide building designers with useful information about the level of ventilation rates needed to offset given rates of internal heat gain.

The paper also presents several analysis performed on a reference room in a case study (Aarhus town hall office in Denmark) in order to validate the analysis method development. Specifically we analysed the influence of using dynamic loads, building thermal mass and ventilation control in the heat transfer model and on the calculation method for the heating balance point temperature of the building.

Finally, the ventilative cooling potential tool outputs are compared with the predictions of a state of the art building performance simulation model of the reference room, highlighting several possible improvements in the evaluation criteria.

KEYWORDS

Ventilative cooling potential, climate analysis, early design stages, overheating, airflow rates

1 INTRODUCTION

The new initiatives and regulation towards low energy buildings forces designers to exploit the cooling potential of the climate to reduce the overheating occurrence and to improve

thermal comfort indoors. Climate analysis is particularly useful at early design stages to support decision making towards cost-effective ventilative cooling solutions. As buildings with different use patterns, envelope characteristics and internal loads level do not follow equally the external climate condition, the climate analysis cannot abstract from building characteristics and use.
Within International Energy Agency (IEA) Annex 62 project (IEA EBC Annex 62 - Ventilative cooling, 2014-2017), national experts are working on the development of a ventilative cooling potential tool (VC tool), which aims at assessing the potential of ventilative cooling by taking into account also building envelope thermal properties, internal gains and ventilation needs.

2 THE VENTILATIVE COOLING POTENTIAL TOOL

The ventilative cooling potential tool is an excel-based tool intended to be used during early design stages for estimating the potential of ventilative cooling.

2.1 Theory

The ventilative cooling potential tool refers to the method proposed by NIST (Axley J.W., Emmerich S.J., 2002) (Emmerich S. J., 2011), further developed within the IEA Annex 62 activities.
This method assumes that the heating balance point temperature (T_{o-hbp}) establishes the outdoor air temperature below which heating must be provided to maintain indoor air temperatures at a defined internal heating set point temperature (T_{i-hsp}).
Therefore, when outdoor dry bulb temperature (T_{o-db}) exceeds the heating balance point temperature, direct ventilation is considered useful to maintain indoor conditions within the comfort zone. At or below the heating balance point temperature, ventilative cooling is no longer useful but heat recovery ventilation should be used to meet minimum air change rates for indoor air quality control and reduce heat losses.
The heating balance point temperature (T_{o-hbp}) can be calculated using Equation 1.

$$T_{o-hbp} = T_{i-hsp} - \frac{q_i}{\dot{m}_{min} c_P + \sum UA} \tag{1}$$

where:

T_{o-hbp} = heating balance point temperature [°C]
T_{i-hsp} = heating set point temperature [°C]
q_i = total internal gains [W/m²]
c_p = air capacity [J/kg-K]
$\dot{m}_{min}$ = minimum required mass flow rate [kg/s]
$\sum UA$ = envelope heat exchange [W/K]
U = average U-value of the envelope [W/m²K]

The minimum required ventilation rate refers to indoor air quality standards, i.e. EN 15251:2007.
The equation derives from the energy balance of a well-mixed single-zone and relies on the assumption that the accumulation term of the energy balance can be negligible. It is a reasonable assumption if either the thermal mass of the zone is negligibly small or the indoor temperature is regulated to be relatively constant. Under these conditions, the energy balance of the zone is steady state and can may provide an approximate mean to characterize the ventilative cooling potential of a climate.

The comfort zone is determined according to the adaptive thermal comfort model proposed in the EN 15251:2007 standard. The upper and lower temperature limits of the comfort zone are calculated using equations 2 and 3.

$$T_{i-max} = 0.33 \cdot T_{rm} + 18.8 + K \quad (2)$$

$$T_{i-min} = 0.33 \cdot T_{rm} + 18.8 - K \quad (3)$$

where

T_{i-max} = upper temperature limit of the comfort zone [°C]
T_{i-min} = lower temperature limit of the comfort zone [°C]
T_{rm} = outdoor running mean temperature [°C]
K = constant depending on required comfort Category: $K = 2$ if comfort cat. I, $K = 3$ if comfort cat. II, $K = 4$ if comfort cat. III.

Below an outdoor running mean temperature of 10°C, the upper temperature limit is set as the upper temperature limit for heating recommended by EN 15251:2007 (Table A.3). Below an outdoor running mean temperature of 15°C, the lower temperature limit is set as the lower temperature limit for heating recommended by EN 15251:2007 (Table A.3).

2.2 Input

The tool requires basic information about a typical room of the building, the building use and the climate.

Figure 1 shows a screenshot of the input data sheet. Orange cells should be fulfilled by the user. Grey cells provide different options for the user (e.g. type of the building). The tool automatically calculates data in grey cells.

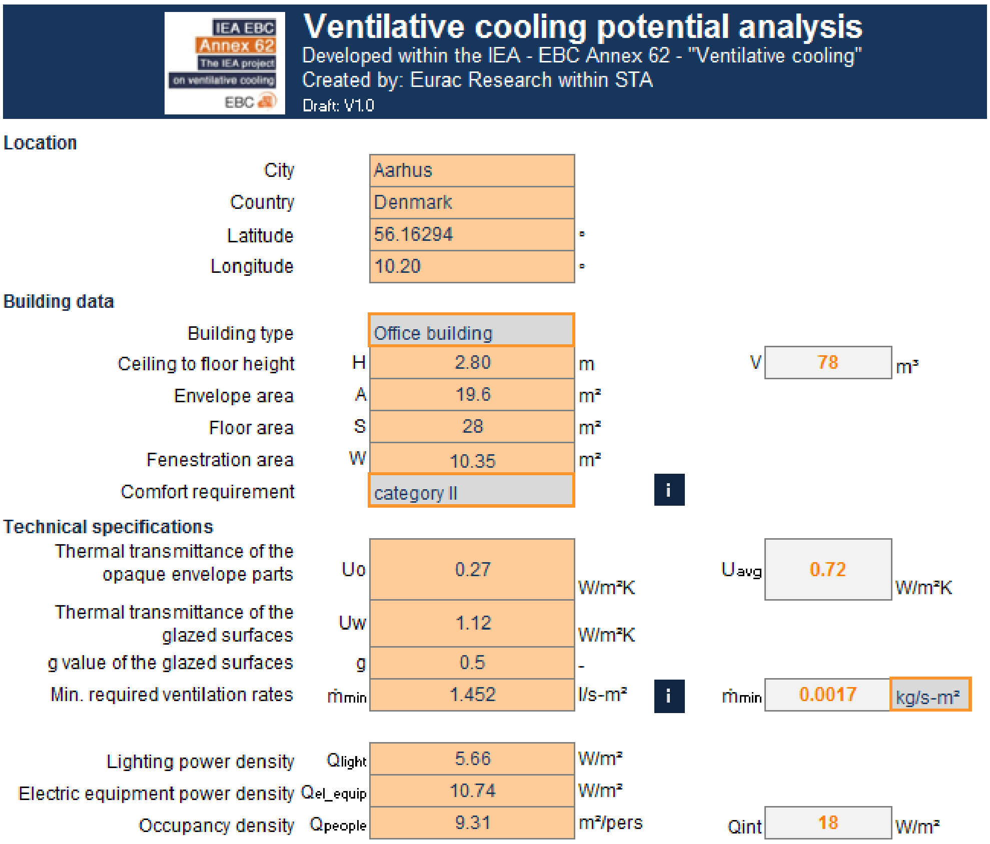


Figure 1: Input data sheet of the ventilative cooling potential tool

Within the building data section, the user is required to input basic internal geometry data of the reference room as well as the type of the building and the comfort category.
Comfort requirements refer to the comfort categories defined by the EN 15251:2007 standard (K parameter Equation 2 and 3). Recommended input values given for each of the different comfort categories are included in the tool and automatically selected.
Various thermal and technical properties specifications about the envelope features are required to determine the transmission losses and the solar gains. Minimum required air change rates (l/s-m^2) determine the ventilation losses within the energy balance of the reference room.
The tool includes a database of standard load profiles of occupancy (Table 2), lighting (Table 3) and electric equipment (Table 4) for different building typologies, which are under publication by REHVA organization. According to the selected building type, the tool sets automatically the typical corresponding occupied time and load profiles on hourly basis due to occupancy, lighting and electric equipment. Internal gains are calculated according to the lighting and electric equipment power density and the occupancy density input by the user.
The tool determines internal gains for each hour of the year according to the load profile, the lighting and electric equipment power density and the occupancy density input by the user.

Climatic data

Annual record of climatic data is user-input on hourly time steps. The climatic data used on this tool are the dry bulb temperature and the global horizontal solar radiation. The weather data should be representative of the examined location's typical meteorological year for the given location.
Several sources for typical meteorological year weather files are available: Meteonorm software (Meteonorm, 2015), International Weather for Energy Calculations (IWEC) data set (ASHRAE, 2001), Typical Meteorological Year (TMY) data set (Wilcox S., 2008) and many others (Energylus Energy Simulation software: Weather Data sources, 2015).

2.3 Evaluation criteria

The analysis is based on a single-zone thermal model applied to user-input climatic data on hourly basis. For each hour of the annual climatic record of the given location, an algorithm splits the total number of hours when the building is occupied into the following groups:

1. **Ventilative Cooling mode [0]**: ventilative cooling is not required when the outdoor temperature is below the heating balance point temperature no ventilative cooling can be used since heating is needed;

   If $T_{o-db} < T_{o-hbp}$ then $\dot{m} = 0$

2. **Ventilative Cooling mode [1]**: Direct ventilative cooling with airflow rate maintained at the minimum required for indoor air quality when the outdoor temperature exceeds the balance point temperature, yet falls below the lower temperature limit of the comfort zone;

   If $T_{o-hbp} \leq T_{o-db} < T_{o-hbp} + (T_{i-max} - T_{i-min})$ then $\dot{m} = \dot{m}_{min}$

3. **Ventilative Cooling mode [2]**: Direct ventilative cooling with increased airflow rate when the outdoor temperature is within the range of comfort zone temperatures.

   If $T_{o-hbp} + (T_{i-max} - T_{i-min}) \leq T_{o-db} \leq T_{i-max} - \Delta T_{crit}$ then $\dot{m} = \dot{m}_{cool}$

The airflow rate required to maintain the indoor air temperature within the comfort zone temperature ranges is computed as in Equation 4. A ΔT_{crit} of 3 K is introduced in order to prevent unrealistic airflow rates;

$$\dot{m}_{cool} = \frac{q_i}{c_p(T_{i-max} - T_{o-db})} \quad (4)$$

4. **Ventilative Cooling mode [3]**: Direct ventilative cooling is not useful when the outdoor temperature exceeds the upper temperature limit of the comfort zone;

   If $T_{o-db} > T_{i-max} - \Delta T_{crit}$ then $\dot{m} = 0$

   If direct ventilative cooling is not useful for more than an hour during the occupied time, the night-time cooling potential over the following night is evaluated as the internal gains that may be offset for a nominal unit night-time air change rate (Equation 5).

$$NCP = \frac{H\rho c_P(T_{i-max} - T_{o-db})}{3600} \quad (5)$$

where:
NCP = night-time cooling potential [W/m²-ach]
H = floor height [m]
ρ = air density [kg/m³]

Compared to the climate suitability analysis method developed by NIST, the ventilative cooling potential analysis tool presented in this research includes two main new features:

- Dynamic load profiles and heating balance point temperature calculation;
- Adaptive thermal comfort based control.

2.4 Outputs

The VC tool calculates the following performance indicators:

- the percentage of time within each month when the building is occupied and:
  - ventilative cooling is not required (VC mode [0]) according to the evaluation criteria described in par. 2.3;
  - direct ventilative cooling with airflow rate maintained at the minimum is required (VC mode [1]) according to the evaluation criteria described in par. 2.3;
  - direct ventilative cooling with increased airflow rate is required (VC mode [2]) according to the evaluation criteria described in par. 2.3;
  - direct ventilative cooling is not useful (VC mode [3]) according to the evaluation criteria described in par. 2.3;
- the night-time cooling potential over the night following the days when direct ventilative cooling is not useful (VC mode [3]) for at least an hour;
- the required ventilation rates (average and standard deviation over each month) to cool the building during occupied hours when direct ventilative cooling with increased airflow rate is required (VC mode [2]);
- the night-time Cooling Degree Hours (CDH);
- the monthly average temperature swing between day and night;
- the monthly average global horizontal radiation.

From climate classification point of view, the outputs are useful to compare the ventilative cooling potential in different climates for different building typologies.

From design point of view, the outputs support the decision making by selecting the most efficient ventilative cooling strategy and by providing rough estimation of the airflow rates needed to cool down the building in relation to internal gains, comfort requirements and envelope characteristics.
The tool enables also to analyse the effect of other energy efficiency measures, like internal gains reduction, solar gains control and envelope performance, on ventilative cooling effectiveness.

3 CASE STUDY: AARHUS TOWN HALL OFFICE (DENMARK)

The case study used for the validation of the analysis is an office room located in the Aarhus municipality building in Denmark.

3.1 Ventilative Cooling potential tool

Input data

The reference office is 3.99 m x width x 7 m large x 2.8 m height (volume 78 m^3) and is occupied by three persons. Lighting and electric equipment power density amounts at 5.7 W/m^2 and 10.7 W/m^2 respectively.
The room has only one external wall (facing south) with 53% Glass to Wall Ratio (GWR). Considering the external wall (U_{wall} = 0.27 W/m^2K) and window constructions (U_{window} = 1.12 W/m^2K) and assuming adiabatic conditions for the other envelope components, the average U-value of the external walls is 0.72 W/m^2K.
The examined required comfort level is category II (new or renovated buildings). According to the EN 15251:2007 standard, the minimum required air change rates to assure an indoor air quality within category II are 1.452 l/s-m^2 (1.9 h^{-1}).
The weather file used for the analysis refers to the city of Copenhagen and derives from the International Weather for Energy Calculations (IWEC) database (ASHRAE, 2001). The climate of Denmark is temperate with small differences from city to city.
Since the solar gain calculation is still under development, we input to the VC tool the solar gains calculated by the building energy simulation model in EnergyPlus (see par. 3.2).

Output data

The graph in Figure 3 reports the ventilative cooling mode distribution in terms of the percentage of time when the building is occupied.
Direct ventilative cooling is useful for more than 85% of the time during the period May - September.

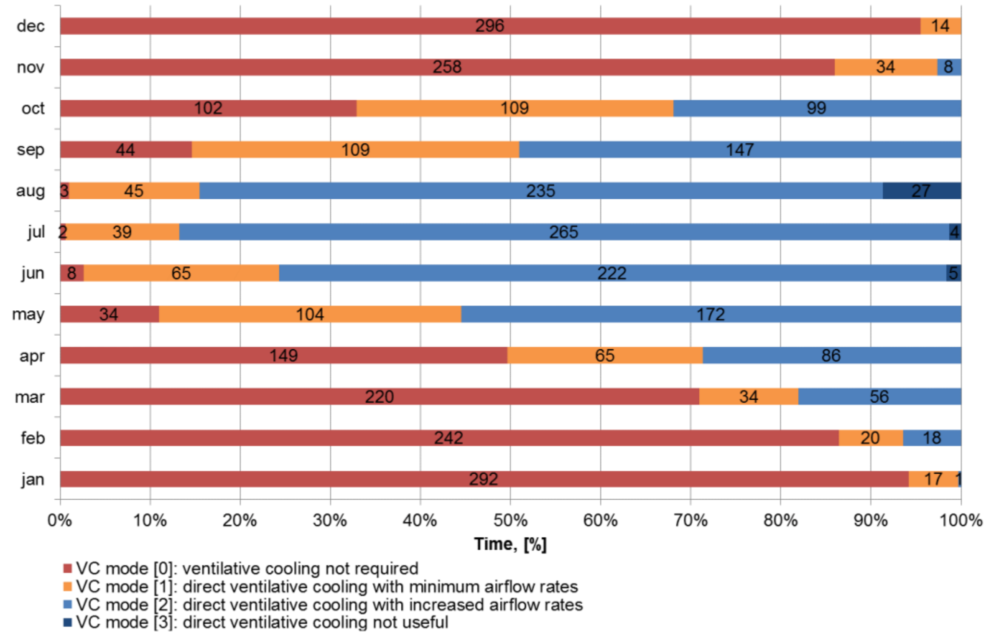


Figure 3: Tool output: percentage of working hours when ventilative cooling is not required, direct ventilative cooling is useful or not useful.

Table 1: Required ventilation rates (average and standard deviation over each month) to cool the building during occupied hours when direct ventilative cooling with increased airflow rate is required (VC mode [2]).

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Average airflow rate	2.68	2.63	2.98	3.73	4.30	4.47	4.68	4.82	3.59	3.17	2.89	0
Standard deviation	0	0.17	0.41	1.08	1.58	1.94	2.22	2.75	0.99	0.61	0.38	0
Nr of hours when VC mode [2] is on	1	18	56	86	172	222	265	235	147	99	8	0

Table 1 reports the required ventilation rates (average and standard deviation over each month) to cool the building during occupied hours when direct ventilative cooling with increased airflow rate is required (VC mode [2]). These statistics provide design guidance for preliminary considerations about the ventilation system and the control strategy. For example, according to the results for Copenhagen, an average airflow rate of 4.82 ± 2.75 h^{-1} is expected to assure that indoor temperatures are within the comfort zone during August for more than 80% of the time. Furthermore, by decreasing the solar and internal loads level, the airflow rate required to provide ventilative cooling would decrease as well and therefore the passive cooling of the building might be possible or more effective using commonly available ventilation strategies.

During wintertime, outdoor temperatures are too cold and a direct ventilative cooling strategy would cause higher heating demand and/or draught problems due to too low indoor temperatures.

Direct ventilative cooling is not useful due to too high outdoor temperature for only 2%, 1% and 9% of the time in June, July and August respectively. In these cases, the Night-time Cooling Potential is around 8 W/m^2-h^{-1}, which means that an airflow of one air change per hour can offset 8 W/m^2 of internal gains produced during the previous day. The average monthly diurnal temperature swing is around 3K during summer.

3.2 Building Energy Simulation model

In order to validate the ventilative cooling potential tool outputs, we modelled the reference office room in EnergyPlus simulation software and compared the simulation results with the tool outputs.

The zone settings are the same as the reference office room described in par. 3.1.1. The schedules of internal gains are defined in order to perfectly match the load profiles used by the tool.
The design flow rates are input as hourly values in a schedule file that reports the required airflow rates, both minimum and increased, calculated by the ventilative cooling potential tool.
The simulation is run in free-floating mode.
The predicted indoor temperatures on hourly frequency were compared with the comfort ranges set in the tool according to the following assumptions:

- If the predicted indoor temperature is lower than the lower temperature limit of the comfort zone and the airflow rates are set at the minimum, then direct ventilative cooling is not useful (VC mode [0]);
- If the predicted indoor temperature is within the comfort zone and the airflow rates are set at the minimum, then direct ventilative cooling is useful if airflow rates are maintained at the minimum required (VC mode [1]). Also time steps when the predicted indoor temperature is lower than the lower temperature limit of the comfort zone and the airflow rates are set at an increased value, are classified as VC mode [1];
- If the predicted indoor temperature is within the comfort zone and the airflow rates are set at an increased value, then direct ventilative cooling is considered useful (VC mode [2]);
- Finally, if the predicted indoor temperature is higher than the higher temperature limit of the comfort zone and the airflow rates are set at an increased value, then direct ventilative cooling is not enough to cool down the reference zone (VC mode [3]).

3.3 Building Energy Simulation model predictions vs VC tool outputs

The graph in Figure 4 shows the percentage of working hours when ventilative cooling is not required, direct ventilative cooling is useful or not useful based on the analyses of building energy simulation model (EnergyPlus) predictions as described above. The results are directly compared with the ventilative cooling potential tool outputs.
This comparison allows us to validate the VC tool outputs as well as to analyse the effect of thermal mass on output results.
Table 2 reports the differences in terms of number of days between the EnergyPlus predictions and the ventilative cooling potential tool outputs.
Highest differences occur during middle seasons for the time when ventilative cooling is not useful (VC mode [0]) and the time when ventilative cooling with increased airflow rates is useful (VC mode [2]). Generally, the VC tool underestimates the number of hours when ventilative cooling is not useful (apart from June, July, August) and overestimates the number of hours when ventilative cooling with increased airflow rates is useful (apart from July). This underestimation exceeds 5 working days per month during spring and fall time, but does not exceed 3 working days per month during summer and winter time.
The differences are mainly related to the evaluation criteria and the simplifications in the heating balance point temperature calculation. According to the indoor temperature prediction of the EnergyPlus model, the average heating balance point temperature is around 15°C. The VC tool calculates an average heating balance point temperature of 12°C.

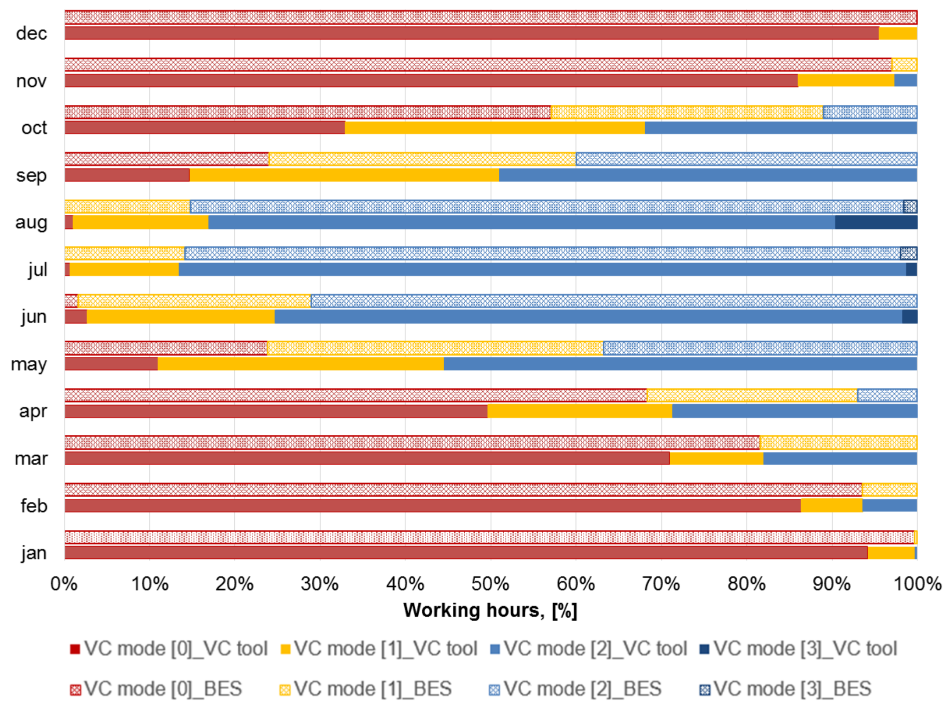


Figure 4. EnergyPlus model output (BES) analysed according to the tool evaluation criteria compared to the ventilative cooling potential tool (VC tool) outputs.

Table 2: Number of days (considering 10hrs/day) difference between EnergyPlus model predictions and ventilative cooling potential tool outputs.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
VC mode [0]	-1.7	-2	-3.3	-5.6	-4	0.3	0.2	0.3	-2.8	-7.5	-3.3	-1.4	-30.8
VC mode [1]	1.6	0.2	-2.3	-0.9	-1.8	-1.7	-0.5	-0.1	0.1	1	2.5	1.4	-0.5
VC mode [2]	0.1	1.8	5.6	6.5	5.8	0.4	0.1	-5.1	2.7	6.5	0.8	0	25.2
VC mode [3]	0	0	0	0	0	0.5	-0.2	2.2	0	0	0	0	2.5

Furthermore, we analysed the effect of the new features introduced in the VC tool compared to the original method developed by NIST (Emmerich S. J., 2011), namely:

1. Adaptive thermal comfort based control instead of standard comfort zone;
2. Constant loads and heating balance point temperature.

The graph in Figure 5 shows the analysis results over the whole for the following cases:

- *BES (EnergyPlus)*: building energy simulation model results;
- *BES (EnergyPlus) with increased thermal mass*: results of the building energy simulation model with additional 8200 kg of thermal mass (corresponding to the mass of a 20cm concrete slab with area equal to the floor area);
- *VC tool*: output of the ventilative cooling potential tool;
- *VC tool: standard comfort zone*: output of the ventilative cooling potential tool considering the standard comfort zone, with lower temperature limit of 20°C and upper temperature limit of 24°C;
- *VC tool: no dynamic load and T_{hbp}*: output of the ventilative cooling potential tool considering constant internal gains (18 W/m²) and heating balance point temperature (12°C).

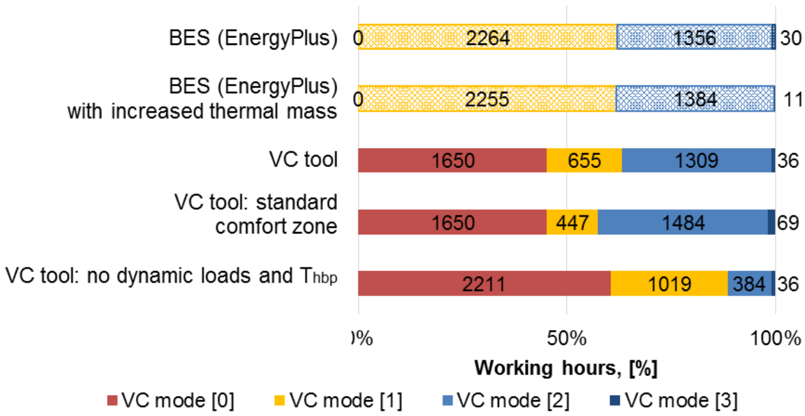


Figure 5. EnergyPlus and VC tool output according to different evaluation criteria.

No significant differences are observed between the original EnergyPlus model and the one with increased thermal mass, meaning that in this case the effect of thermal mass can be neglected.

The use of a standard comfort zone within the evaluation criteria of the VC tool causes an additional overestimation of the ventilative cooling with increased airflow rate mode compared to the case with adaptive thermal comfort based control. Since the upper temperature limit does not vary according to the outdoor temperatures, the time when ventilative cooling is not considered useful is overestimated as well because of the high temperatures.

Higher differences occur when internal gains and heating balance point temperature are considered constant over the whole time. The ventilative cooling potential is up to three times under estimated.

4 DISCUSSION

The steady-state assumptions seem to be acceptable in this case. The Aarhus municipality building town hall office does not have massive constructions. The assumptions validity needs to be further tested on other building types located in different climates (hot, temperate).

Highest differences between EnergyPlus model predictions and VC tool outputs occur during middle season. The evaluation criteria allow direct ventilative cooling with increased rates even when the outdoor temperatures are too low. Analysing the EnergyPlus model results on thermal comfort, we observed when ventilative cooling with increased rates is activated at low outdoor temperatures, the model predicts discomfort due to too cold temperatures, meaning that the increased airflow rates have a too high cooling effect.

The introduction of an outdoor temperature limit condition for VC mode [2] would prevent this issue.

Since the current European standard on thermal comfort does not provide for any recommendation on relative humidity, the evaluation criteria adopted by the tool does not include consideration about relative humidity. According to the Copenhagen weather file, less than 1% of the time the dew point temperature exceeds the 17°C limit proposed by the NIST methodology (Emmerich S. J., 2011). Therefore, the introduction of a control based on air humidity would not affect the results for the present case study. The relative humidity based control is still under discussion within the IEA Annex 62 experts and will be introduced in the next versions of the tool.

Furthermore, as mentioned before, a simplified solar radiation model for solar gains calculation is under implementation.

5 CONCLUSIONS

The paper presents the ventilative cooling potential tool (VC tool) which is under development within the IEA Annex 62 project. The tool analyse the potential of ventilative cooling by taking into account not only climate conditions, but also building envelope thermal properties, internal gains and ventilation needs.

The analysis is based on a single-zone thermal model applied to user-input climatic (hourly) basis and thermal data. For each hour of the annual climatic record of the given location, an algorithm identifies over the occupied time the number of hours when ventilative cooling is useful and estimates the airflow rates needed to prevent building overheating.

The tool is particularly suitable for early design phases, as it requires only basic information about a typical room of the building, the building use and an annual climatic record.

Furthermore, the tool provide building designers with useful information about the level of ventilation rates needed to offset given rates of internal heat gains.

As validation of results, the ventilative cooling potential tool outputs are compared with the predictions of a building energy simulation model of the reference room, highlighting the following aspects:

- The steady-state assumptions seem to be acceptable in case of no massive constructions and cold climates, but their validity needs to be further tested on other case studies located in different climates;
- Dynamic internal loads and calculation of the heating balance point temperature need to be considered in order to have realistic results;
- The introduction of an outdoor temperature limit condition for ventilative cooling mode with increased airflow rates would improve further the tool outputs reliability.

Further improvements of the tool such as internal calculation of solar gains and evaluation criteria based on relative humidity are under discussion within the Annex 62 national experts.

6 ACKNOWLEDGEMENTS

The work presented in this paper has been largely developed in the context of the IEA EBC Annex62: Ventilative cooling. The EURAC contribution has been funded by the Stiftung SüdTiroler Sparkasse.

7 REFERENCES

ASHRAE. (2001). International Weather for Energy Calculation (IWEC Weather files) Users manual and CD-rom. Atlanta.

Axley J.W., Emmerich S.J. (2002). A method to assess the suitability of a climate for natural ventilation of commercial buildings. Proceedings of indoor air, (pp. 854-859). Monterey, California.

Emmerich S. J., P. B. (2011). Impact of adaptive thermal comfort on climatic suitability of natural ventilation in office buildings. Energy and Buildings, 43(2101-2107).

Energylus Energy Simulation software: Weather Data sources. (2015, 07 07). Retrieved from http://apps1.eere.energy.gov/buildings/energyplus/weatherdata_sources.cfm

IEA EBC Annex 62 - Ventilative cooling. (2014-2017). http://venticool.eu/annex-62-home/.

Meteonorm. (2015, 07 07). Retrieved from http://meteonorm.com/

Wilcox S., M. W. (2008). User's Manual for TMY3 Data Sets. NREL/TP-581-43156. Golden, Colorado: National Renewable Energy Laboratory.

8 APPENDIX

Table 3: Occupancy load profile. Source: REHVA (under publication)

Time	Residential building	Department store	Hospital	Hotel	Office building	Restaurant	School	Sport, terminal, theatre
00:00-01:00	1	0	0.4	0.9	0	0	0	0
01:00-02:00	1	0	0.4	0.9	0	0	0	0
02:00-03:00	1	0	0.4	0.9	0	0	0	0
03:00-04:00	1	0	0.4	0.9	0	0	0	0
04:00-05:00	1	0	0.4	0.9	0	0	0	0
05:00-06:00	1	0	0.4	0.9	0	0	0	0
06:00-07:00	0.5	0	0.4	0.7	0	0.1	0	0
07:00-08:00	0.5	0	0.5	0.4	0.2	0.4	0	0
08:00-09:00	0.5	0.1	0.6	0.4	0.6	0.4	0.6	0.6
09:00-10:00	0.1	0.3	0.8	0.2	0.6	0.4	0.7	0.6
10:00-11:00	0.1	0.3	0.8	0.2	0.7	0.2	0.6	0.6
11:00-12:00	0.1	0.7	0.8	0.2	0.7	0.5	0.4	0.6
12:00-13:00	0.1	0.6	0.8	0.2	0.4	0.8	0.3	0.6
13:00-14:00	0.2	0.5	0.8	0.2	0.6	0.7	0.7	0.6
14:00-15:00	0.2	0.6	0.8	0.2	0.7	0.4	0.6	0.6
15:00-16:00	0.2	0.6	0.8	0.3	0.7	0.2	0.4	0.6
16:00-17:00	0.5	0.9	0.8	0.5	0.6	0.25	0.2	0.6
17:00-18:00	0.5	0.9	0.6	0.5	0.2	0.5	0	0.6
18:00-19:00	0.5	1	0.5	0.5	0	0.8	0	0.6
19:00-20:00	0.8	0.9	0.5	0.7	0	0.8	0	0.6
20:00-21:00	0.8	0.7	0.4	0.7	0	0.8	0	0.6
21:00-22:00	0.8	0	0.4	0.8	0	0.5	0	0.6
22:00-23:00	1	0	0.4	0.9	0	0.35	0	0
23:00-00:00	1	0	0.4	0.9	0	0.2	0	0

Table 4: Lighting load profile. Source: REHVA (under publication)

Time	Residential building	Department store	Hospital	Hotel	Office building	Restaurant	School	Sport, terminal, theatre
00:00-01:00	0	0	0.5	0.22	0	0	0	0
01:00-02:00	0	0	0.5	0.17	0	0	0	0
02:00-03:00	0	0	0.5	0.11	0	0	0	0
03:00-04:00	0	0	0.5	0.11	0	0	0	0
04:00-05:00	0	0	0.5	0.11	0	0	0	0
05:00-06:00	0	0	0.5	0.22	0	0	0	0
06:00-07:00	0.15	0	0.5	0.44	0	0.1	0	0
07:00-08:00	0.15	0	0.5	0.56	0.2	0.4	0	0
08:00-09:00	0.15	1	0.9	0.44	0.6	0.4	0.6	0.6
09:00-10:00	0.15	1	0.9	0.44	0.6	0.4	0.7	0.6
10:00-11:00	0.05	1	0.9	0.28	0.7	0.2	0.6	0.6
11:00-12:00	0.05	1	0.9	0.28	0.7	0.5	0.4	0.6
12:00-13:00	0.05	1	0.9	0.28	0.4	0.8	0.3	0.6
13:00-14:00	0.05	1	0.9	0.28	0.6	0.7	0.7	0.6
14:00-15:00	0.05	1	0.9	0.28	0.7	0.4	0.6	0.6
15:00-16:00	0.05	1	0.9	0.28	0.7	0.2	0.4	0.6
16:00-17:00	0.2	1	0.5	0.28	0.6	0.25	0.2	0.6
17:00-18:00	0.2	1	0.5	0.28	0.2	0.5	0	0.6
18:00-19:00	0.2	1	0.5	0.67	0	0.8	0	0.6
19:00-20:00	0.2	1	0.5	0.89	0	0.8	0	0.6
20:00-21:00	0.2	1	0.5	1	0	0.8	0	0.6
21:00-22:00	0.2	0	0.5	0.89	0	0.5	0	0.6
22:00-23:00	0.15	0	0.5	0.67	0	0.35	0	0
23:00-00:00	0.15	0	0.5	0.41	0	0.2	0	0

Table 5: Electric equipment load profile. Source: REHVA (under publication)

Time	Residential building	Department store	Hospital	Hotel	Office building	Restaurant	School	Sport, terminal, theatre
00:00-01:00	0	0	0.5	0.22	0	0	0	0
01:00-02:00	0	0	0.5	0.17	0	0	0	0
02:00-03:00	0	0	0.5	0.11	0	0	0	0
03:00-04:00	0	0	0.5	0.11	0	0	0	0
04:00-05:00	0	0	0.5	0.11	0	0	0	0
05:00-06:00	0	0	0.5	0.22	0	0	0	0
06:00-07:00	0.15	0	0.5	0.44	0	0.1	0	0
07:00-08:00	0.15	0	0.5	0.56	0.2	0.4	0	0
08:00-09:00	0.15	1	0.9	0.44	0.6	0.4	0.6	0.6
09:00-10:00	0.15	1	0.9	0.44	0.6	0.4	0.7	0.6
10:00-11:00	0.05	1	0.9	0.28	0.7	0.2	0.6	0.6
11:00-12:00	0.05	1	0.9	0.28	0.7	0.5	0.4	0.6
12:00-13:00	0.05	1	0.9	0.28	0.4	0.8	0.3	0.6
13:00-14:00	0.05	1	0.9	0.28	0.6	0.7	0.7	0.6
14:00-15:00	0.05	1	0.9	0.28	0.7	0.4	0.6	0.6
15:00-16:00	0.05	1	0.9	0.28	0.7	0.2	0.4	0.6
16:00-17:00	0.2	1	0.5	0.28	0.6	0.25	0.2	0.6
17:00-18:00	0.2	1	0.5	0.28	0.2	0.5	0	0.6
18:00-19:00	0.2	1	0.5	0.67	0	0.8	0	0.6
19:00-20:00	0.2	1	0.5	0.89	0	0.8	0	0.6
20:00-21:00	0.2	1	0.5	1	0	0.8	0	0.6
21:00-22:00	0.2	0	0.5	0.89	0	0.5	0	0.6
22:00-23:00	0.15	0	0.5	0.67	0	0.35	0	0
23:00-00:00	0.15	0	0.5	0.41	0	0.2	0	0

NIGHT TIME COOLING BY VENTILATION OR NIGHT SKY RADIATION COMBINED WITH IN-ROOM RADIANT COOLING PANELS INCLUDING PHASE CHANGE MATERIALS

Eleftherios Bourdakis[1], Bjarne W. Olesen[*1], Fabio Grossule[1]

[1]*Technical University of Denmark*
International Centre for Indoor Environment and Energy
Nils Koppels Allé, Building 402
Kongens Lyngby, Denmark
** bwo@byg.dtu.dk*

ABSTRACT

Night sky radiative cooling technology using PhotoVoltaic/Thermal panels (PVT) and night time ventilation have been studied both by means of simulations and experiments to evaluate their potential and to validate the created simulation model used to describe it. An experimental setup has been constructed at the Technical University of Denmark, where the outside PVT panels are connected through a storage tank to in-room radiant ceiling panels. The radiant ceiling panels include phase change material (PCM) and embedded pipes for circulating water. Due to the phase change material it is possible to store the heat generated during the day from internal sources. Then during the night the panels can be cooled down again and regenerated. The possibility of cooling down the panels during the night with outside air was also studied. The night cooling power of the PVT panels ranged from 92 to 119 W/m^2 depending on the sky clearness. This cooling power was enough to remove the stored heat and regenerate the ceiling panels. The validation simulation model results related to PCM were close to the corresponding results extracted from the experiment, while the results related to the production of cold water through the night sky radiative cooling differed significantly. The possibility of night time ventilation was studied through simulations for three different latitudes. It was concluded that for Danish climatic conditions night time ventilation would also be able to regenerate the panels while its contribution is not sufficient in warmer South-European climates.

KEYWORDS

Night Time Ventilation, Night Sky Radiative Cooling, Phase Change Materials, Thermal Comfort, Renovation

1 INTRODUCTION

World energy consumption has been increasing rapidly the last decades, mainly due to the population growth and the industrial and technological development. In order to address this issue, European Union (EU) has put into force the agreement “20-20-20” (European Commission, 2007), which sets several ambitious targets to be met from the member states by the year 2020.

According to the International Energy Agency (IEA) (2013), energy consumption in buildings accounts for more than 40% of the primary energy consumption in many IEA member states. Therefore, vast changes have to be made in the building sector and especially in the Heating, Ventilation and Air-Conditioning (HVAC) systems, in order to reach the aforementioned targets. Most of the buildings which will exist by the year 2020 have already been built, since a building has a life span of more than 50 years. Thus, much effort must be put into retrofitting existing buildings.

One solution that could contribute in reducing the energy consumption caused by HVAC systems is the Thermo Active Building Systems (TABS). Main advantages of TABS are the distribution of cooling over a longer period which results in reduced peak loads, reduction in the buildings materials due to the reduced suspended ceiling for ventilation purposes since the required air flow rate is reduced. Moreover, the fact that the cooling temperature is close to room temperature increases the energy efficiency of heat pumps and ground heat exchangers (Babiak et al., 2007). Many modern office buildings in Europe are using TABS to store heat in the slabs during the day and remove the stored heat during the night (Kolarik et al., 2015).

Nevertheless, this system is in most cases not applicable for energy renovation of buildings. A solution that could be effective in cases of buildings renovation is a radiant cooling system with the implementation of Phase Change Material (PCM) since it has the benefits of a heavyweight construction with the thickness of a lightweight construction (Koschenz & Lehmann, 2004). PCMs are organic or inorganic substances that absorb heat while melting and release it while solidifying. The most important advantages of the implemention of PCMs in the structure, as they have been reported in the literature (Cabeza et al., 2007; BASF, 2010; Pavlov, 2014; Grossule, 2015), are the improved thermal inertia compared to conventional concrete, the reduction of peak cooling load, the shift of part of the cooling demand to night time where lower electricity prices occure, the attenuation of the interior air temperature fluctuations and the reduction of the size of the HVAC system.

One passive method that could be utilized for discharging the PCM is the night sky radiative cooling. During night time the effective sky temperature is lower than the one of the solar panels surface, therefore the panels release heat in the form of radiation towards the atmosphere (Meir et al., 2002). There are several ways with which night sky radiative cooling can be exploited (Grossule, 2015), but in this study only a closed water based system will be examined. The advantages of such a system are the reduction of energy consumption, since the only energy required is that for the water pumps, a higher utilization factor for solar thermal panels is achieved since they are exploited also during night time, cooling demand and cold water production through night sky radiative cooling are in phase since clear sky occurs more often during summer time and it can be coupled with thermal storage materials like PCM (Meir et al., 2002; Eicker & Dalibard, 2011; Hosseinzadeh & Taherian, 2012). Another passive method that could be exploited is the night time natural ventilation when the ambient air temperature is low enough.

The purpose of this study was to examine and evaluate different methods for discharging the PCM, and to couple the PCM with solar panels for discharging it passively through night sky radiative cooling. The results of the experiment were used to validate the corresponding simulation model, which was then used to simulate the possibility of discharging the PCM by exploiting night time ventilation under Copenhagen's, Milan's and Athens' weather conditions.

2 EXPERIMENT

2.1 Experimental setup

For the purpose of the experiments a climatic chamber at the facilities of the International Centre for Indoor Environment and Energy (ICIEE) was used. The floor surface was 22.7 m^2 and the height from the floor to the suspended ceiling 2.5 m. The walls and the roof are made of two steel sheets separated by 10 cm of mineral wool for insulation, while the whole chamber is not exposed directly to the ambient weather conditions since it is enclosed in a bigger building.

The suspended ceiling consisted from panels made from a mixture of clay with 26% of PCM by weight. The PCM used was the type DS 5040X produced from BASF with melting point of 23^{o}C (BASF, 2010). As it is shown in Figure 1, above the PCM ceiling panels there was a 0.5 m height plenum, from where the ventilation air was supplied in the chamber through the gaps between the ceiling panels. For discharging the PCM, the ceiling panels had embedded pipes inside for circulating water during night time. The chamber was designed accordingly to simulate a two-person office, with the use of heat dummies for the occupants and the equipment. Therefore, the heat dummies were activated during typical office hours, namely from 9:00 to 17:00. The total power of the heat gains was 540 W.

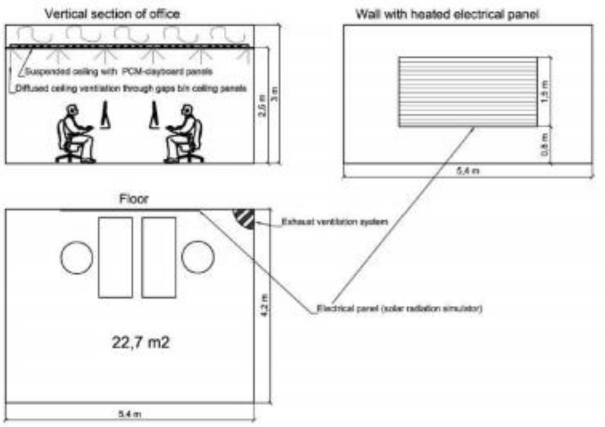


Figure 1: Chamber’s layout (Pavlov, 2014)

Figure 2: Experimental Setup (Grossule, 2015)

The ventilation flow rate was set to 30 l/s, sized according to Annex B.1.2 of the standard DS 15251 (2007) for removing pollutants and providing fresh air, counting only on the performance of the PCM for removing the heat. The air supply temperature was set to 18.5^{o}C. As in the case of the heat dummies, the ventilation was operating from 9:00 to 17:00. The exhaust of the ventilation can be seen in the far corner of the chamber in Figure 2. In the second experimental case the ventilation was also used from 22:00 to 06:00 as a method for discharging the PCM.

For the simulation of the solar heat gains, an electrical heating panel was used on the wall which was supposed to be facing south, as it can be seen in Figure 2. The schedule of the solar heat gains is illustrated in Figure 3.

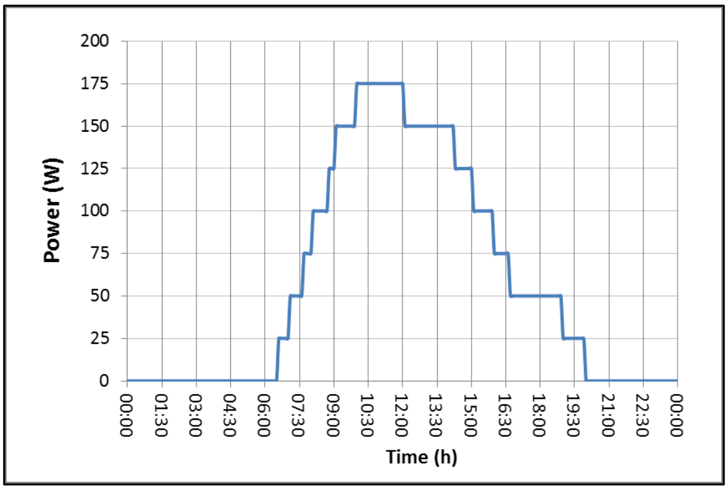


Figure 3: Daily solar heat gains profile

The second method used for discharging the PCM was by circulating water to the embedded pipes during night time. In experiment 1 the cold water was directly supplied from the main chiller of the laboratory's facilities. In experiment 2 the system was upgraded with the implementation of 3 PhotoVoltaic/Thermal (PVT) panels for the production mainly of cold water, by circulating water in the PVT panels during night time. In this study the production of hot water and electricity was not taken into consideration. Each PVT panel had a surface area of 1.3 m^2, a tilt angle of 45° and they were facing south.

The PVT panels were connected with two storage tanks through a heat exchanger. Both tanks had a volume of 255 l and the one was used for storing cold water (CWT) while the other one for storing hot water (HWT). The reason why a heat exchanger was installed in between was due to different settings required in the PVT panels and the tanks regarding water pressure. Furthermore, in this way a smaller quantity of glycol was required which was used as antifreeze. The water from the heat exchanger was circulated to a heat exchanger enclosed in the upper part of the CWT, while a second enclosed heat exchanger was connected with the main chiller of the laboratory's facilities. The main chiller was used as an ancillary system for ensuring the production of cold water in case where the production from the night sky radiative cooling was not enough. The water from the bottom of the CWT was circulated to the PCM panels and from there it was returned to the top of the CWT. In Figure 4 the schematic drawing of the hydraulic system is presented.

In both cases (water provided either from the main chiller or the CWT) the circulation of the water would start at 20:00 and would continue until 7:30 if none of the two following conditions was met earlier:

- The average bottom surface temperature of the PCM panels dropped below 21^{o}C
- The operative temperature of the room dropped below 21^{o}C

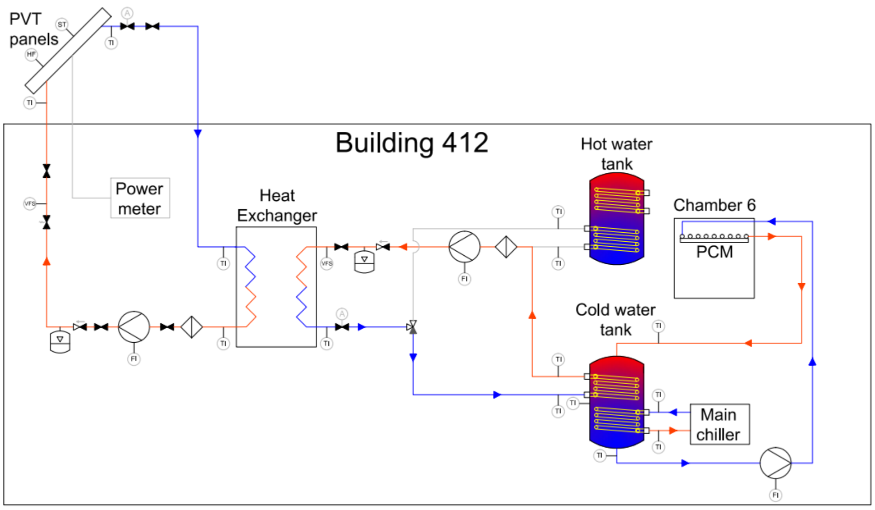


Figure 4: Schematic drawing of the hydraulic system

2.2 Experiment 1

As it was mentioned before, for experiment 1 the cold water for discharging the PCM was provided from the main chiller of the laboratory. Four cases were examined in the first experiment; the first two cases were identical to the cases A1 and B1 examined by Pavlov (2014), in order to ensure that the system works in the same way. In the third case the embedded pipes were used for discharging the PCM and two floor fans were installed in the corners to examine the impact of improved air mixing in the operative temperature and the PCM bottom surface temperature. Lastly, in the fourth case the two desks with the heat dummies where placed closer to the centre of the room and then split towards the walls, in order to distribute the heat sources more evenly in the room. The settings for the four cases are presented in Table 1. Each experimental case lasted four days.

Table 1: Examined cases of experiment 1

Case	**Night ventilation (22:00 – 6:00)**	**Water air flow**
1	Off	0.15 m^3/h
2	62 l/s	Off
3	Off	0.15 m^3/h
4	Off	0.15 m^3/h

2.3 Experiment 2

For the second experiment two cases were examined. In the first one the air temperature inside the chamber was kept constantly at 26^{o}C from 9:00 to 18:00, while in the second case it was kept constantly at 28^{o}C for the same time period. During the rest of the day the ventilation was deactivated. In this experiment the heat dummies and the solar panel were deactivated and the room air temperature was controlled through the ventilation system. Each case lasted for three days.

2.4 Simulation study

The results extracted from experiment 2 were compared with a simulation model made in TRNSYS 17, in order to validate it. In order for the simulation model to be more accurate, the extracted data from a weather station installed next to the PVT panels were used as an input to the PVT component in the simulation model. All the rest of the settings of the simulation model were as close as possible to the conditions of experiment 2. Afterwards, the model was used for simulating the possibility of exploiting night time ventilation for discharging the PCM under the weather conditions of Copenhagen, Milan and Athens. The night time flow rate that was used was 62 l/s as it was set in the second case of experiment 1, while the air supply temperature was the ambient air temperature. The period simulated was the whole cooling period, namely from 1st of May until the 30th of September.

3. RESULTS

3.1 Experiment 1 results

In the Figure below the operative temperature in the interior of the chamber during the four examined cases of experiment 1 is presented. As it can be seen, both the improvement of the air mixing (case 3) and the separation of the internal heat sources (case 4) caused a reduction in the peak temperature during the occupancy period.

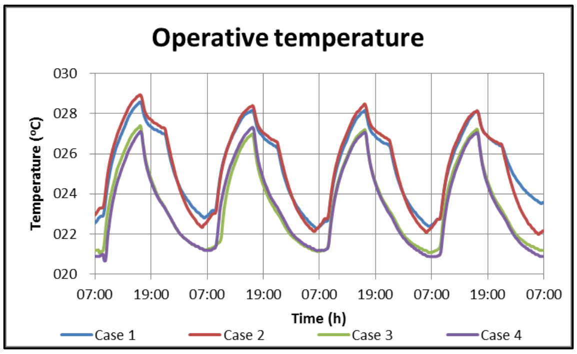


Figure 5: Operative temperature of experiment 1

In Figure 6 the temperature of the bottom surface of the PCM panels is presented. As before, a reduction in the peak temperature is observed in cases 3 and 4. Furthermore, the minimum surface temperature is lower in these two cases, which means that a higher percentage of PCM was discharged during the night time. On the other hand, the examined combination of air supply volume and temperature during night time proved to be insufficient for discharging the PCM completely.

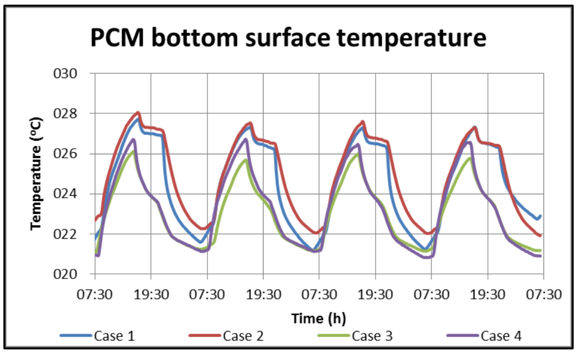


Figure 6: Bottom surface temperature of the PCM panels

One of the drawbacks of PCMs is their low thermal conductivity. In order to examine the impact of the changes examined during cases 3 and 4 to the energy absorbed by the PCM, the percentage of the utilised PCM was calculated. For this calculation the bottom and the top PCM surface temperatures were used, and the results can be seen in Table 2. As it can be seen the changes implemented in case 3 and 4 increased the percentage of the utilised PCM.

Table 2: Percentage of PCM utilization level

	Case 1	Case 2	Case 3	Case 4
Percentage of PCM utilized (%)	90.8	78.1	95.1	96.7

3.2 Experiment 2 and validation simulations results

The results from the second experiment will be presented in the following figures combined with the results from the validation simulations, in order to be compared directly. In Figure 7 the air temperature at height 0.6 m and the operative temperature from a representative experimental day of case 1 are presented. As it can be seen the results from the simulation match satisfactorily with the results extracted from the corresponding experiment case. In Figure 8 the average temperature of the bottom surface of the PCM panels is presented. As before, the two results are satisfactorily close. The three spikes that are observed in the simulation curve in the morning were caused by the fact that a deadband was not implemented.

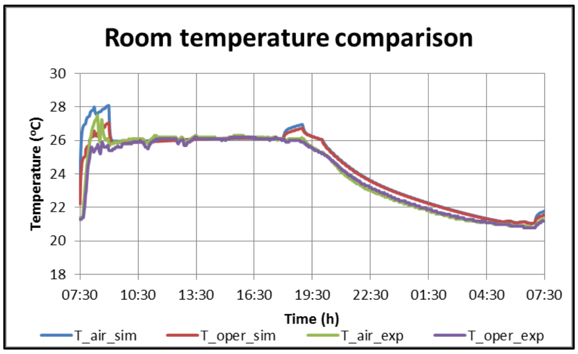


Figure 7: Air and operative temperature of simulation and experiment for case 1

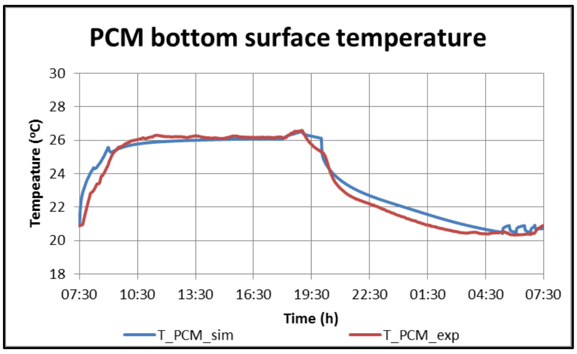


Figure 8: Bottom surface temperature of PCM panels of simulation and experiment for case 1

In Figure 9 the air temperature at height 0.6 m and the operative temperature from a representative experimental day of case 2 are shown. The downwards pikes that are observed in the experiment curves were caused by the opening of the door of the chamber. In Figure 10 the average temperature of the bottom surface of the PCM panels is presented. As in the previous case, the pike at the end of the curve is caused by the absence of a deadband. From both Figure 9 and Figure 10 it can be seen that the simulation results of case 2 matched to a large extent with the results from the corresponding experimental case.

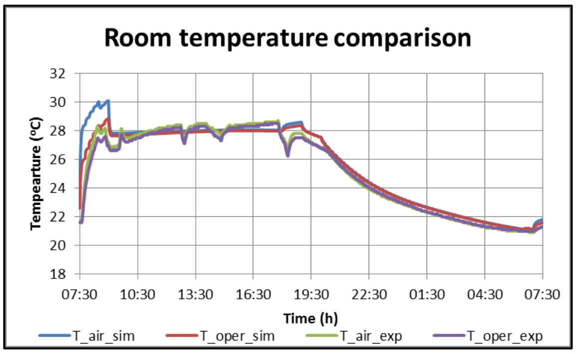


Figure 9: Air and operative temperature of simulation and experiment for case 2

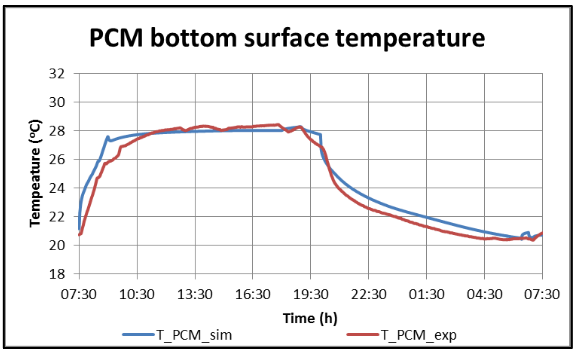


Figure 10: Bottom surface temperature of PCM panels of simulation and experiment for case 2

In the figures below the average specific cooling power of each night of the second experiment and the corresponding energy that was released towards the atmosphere are presented. As it can be seen, the results extracted from the simulations are underestimated compared to those extracted from the experiment.

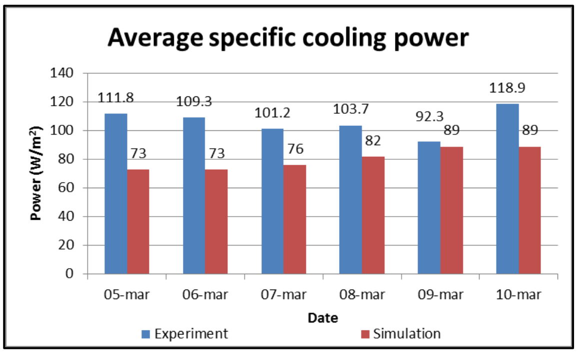


Figure 11: Average specific cooling power

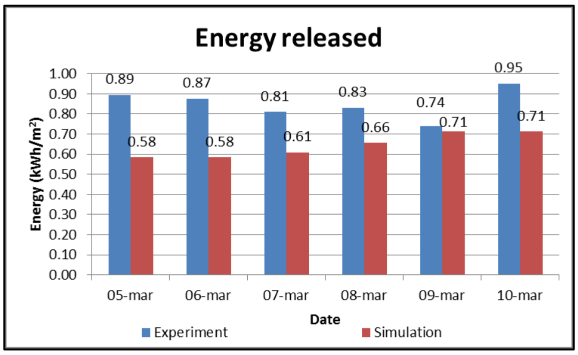


Figure 12: Energy released towards the atmosphere

The reason why this inaccuracy was observed was caused by the effective sky temperature values implemented in the TRNSYS model. Since this parameter was not possible to be measured by the weather station, a theoretical calculation was used as it is described by Grossule (2015) in Appendix 12.1.3.

Moreover, the cooling output is affected by the difference between the water temperature entering and exiting the PVT panels. As it can be seen in **Error! Reference source not found.** the ΔT in the case of simulation is significantly lower compared to the case of the experimental results.

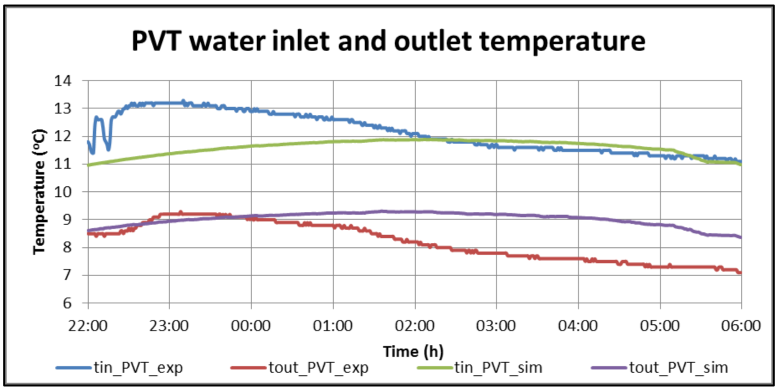


Figure 13: Water inlet and outlet temperature at the PVT panels

3.3 Simulation study results

In Table 3 the performance of the office room in terms of operative temperature in the three simulated location is presented. At this point, it should be reminded that standard DS 15251 requires the 95% of the occupancy period to be within the range 23.5 – 25.5^{o}C, 23 – 26^{o}C or 22 – 27^{o}C in order for a building to be evaluated as category I, II or III respectively. The presented results are the percentages in which the operative temperature was within the suggested range. It can be observed that Athens had the best thermal performance out of the three examined locations, but it was still not good enough for satisfying the requirements for category III of standard DS 15251.

Table 3: Operative temperature performance based on standard DS 15251

	Category I (23.5 – 25.5^{o}C)	Category II (23 – 26^{o}C)	Category III (22 – 27^{o}C)
COP	25%	39%	43%
MIL	28%	42%	46%
ATH	29%	42%	72%

In Figure 14 the daily average performance of the PCM is illustrated. The percentage of charged PCM refers to the end of the occupancy period, namely shows the percentage of PCM that was utilized during that period. On the other hand, the percentage of the discharged PCM refers to the beginning of the occupancy period, thus it reflects the percentage that is still not discharged after the night time ventilation. It can be seen that while in Copenhagen and Milan PCM was almost fully discharged, in the case of Athens approximately the one third of it was not discharged.

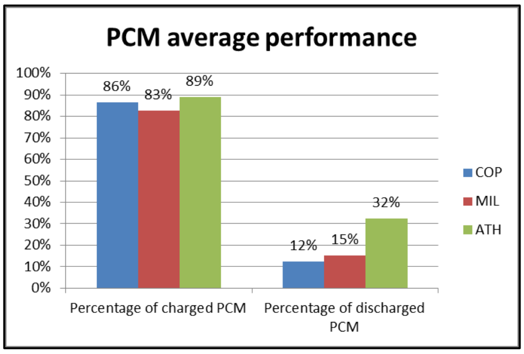


Figure 14: Average daily percentage of charged and discharged PCM

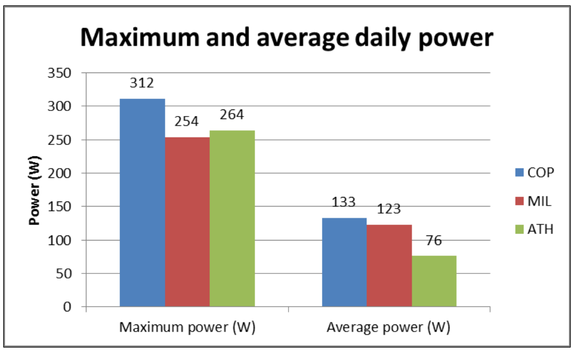


Figure 15: Maximum and daily average power

In Figure 15 the maximum and the daily average cooling power are presented. As it can be seen, although the maximum cooling power in Athens is comparable with the one of Copenhagen, the daily average cooling power of Athens is approximately the half of the one of Copenhagen. The corresponding results of Milan are closer to the results of Copenhagen than those of Athens.

4. DISCUSSION

Regarding the experiment 1, the calculation regarding the percentage of utilization is just an indication, since the temperature distribution inside the panels is not known. This can be observed from the values for cases 1 and 2, which were expected to be closer, since the interior of the experiment was identical. Since the air flow rate that was examined as a discharging method proved to be insufficient for discharging the PCM panels, a higher air flow rate could be examined, or a lower air supply temperature. A lower supply temperature would be feasible in northern European climate areas, but not in Mediterranean climate areas.

The second experiment was conducted during March 2015, a period in which the ambient weather conditions were in favour for producing cold water through night sky radiative cooling. Thus, the experiment should be repeated during summer time, when it is expected that the cooling demand would be higher and the ambient air temperature as well.

The reason why Copenhagen performed worse than Milan and Athens in terms of thermal environment was the low position of the sun which increased significantly the solar heat gains in this location. The same simulations should be repeated with the implementation of a solar shading system.

5. CONLUSIONS

In this study two experiments were conducted to examine the performance of a radiant ceiling cooling system with PCM combined with PVT panels for discharging the PCM passively, while the results extracted from the second experiment were used to validate the corresponding TRNSYS simulation model. The validated model was used to simulate in three different locations the possibility of discharging the PCM by exploiting night time ventilation.

From the first experiment it can be concluded that the circulation of water in the embedded pipes is more effective in discharging the PCM compared to the tested night time ventilation flow rate. Furthermore, it can be concluded that the location of the PCM compared to the heat gain sources affects significantly the performance of the PCM. Therefore, special attention has to be given when designing a cooling system which includes PCM.

From the second experiment it can be concluded that the cold water produced through the night sky radiative cooling was sufficient for discharging the PCM for both examined cases. On the other hand, since – as it was mentioned before – the experiment was conducted during March, this discharging method should also be examined during summer time or ideally during a whole year in order to have a more comprehensive view of the performance of the PVT panels. In order to have the complete performance of the PVT panels, also the production of hot water and electricity needs to be taken into consideration.

Regarding the simulation model, the part simulating the chamber with the PCM ceiling panels proved to be satisfactorily accurate, while the solar loop containing the PVT panels needs to be further investigated and improved. The model should be compared with an experiment conducted during summer time for a more complete and accurate validation.

From the simulation study it can be concluded that the night time ventilation can be exploited for discharging the PCM passively in areas with Nordic climate but is insufficient for Mediterranean climate conditions.

6. REFERENCES

Babiak, J., Olesen, B. W., & Petráš, D. (2007). Low temperature heating and high temperature cooling (2nd ed., Vol. 7). Forssa, Finland: Forssan Kirjapaino Oy.

BASF. (2010). Micronal ® PCM Intelligent temperature management for buildings. Ludwigshafen, Germany: BASF.

Cabeza, L. F., Castellón, C., Nogués, M., Medrano, M., Leppers, R., & Zubillaga, O. (2007). Use of microencapsulated PCM in concrete walls for energy savings. Energy and Buildings, 39(2), 113 - 119.

Dansk standard. (2007). DS 15251 Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics. Charlottenlund, Denmark: Dansk Standard.

Eicker, U., & Dalibard, A. (2011). Photovoltaic–thermal collectors for night radiative cooling of buildings. Solar Energy, 85(7), 1322 - 1335.

European Commission. (2007). The EU climate and energy package. Retrieved June 30, 2015, from http://ec.europa.eu/clima/policies/package/index_en.htm

Grossule, F. (2015). The use of phase change materials for cooling of buildings combined with night sky radiant cooling. Kongens Lyngby, Denmark: Technical University of Denmark.

Hosseinzadeh, E., & Taherian, H. (2012). An experimental and analytical study of a radiative cooling system with unglazed flat plate collectors. International Journal of Green Energy, 9, 766 - 779.

International Energy Agency. (2013). Modernising Building Energy Codes to secure our Global Energy Future: Policy Pathway. Retrieved 6 26, 2015, from http://www.iea.org/publications/freepublications/publication/policy-pathways-modernising-building-energy-codes.html

Kolarik, J., Toftum, J., & Olesen, B. W. (2015). Operative temperature drifts and occupant satisfaction with thermal environment in three office buildings using radiant heating/ cooling system. Healthy Buildings Europe 2015.

Koschenz, M., & Lehmann, B. (2004). Development of a thermally activated ceiling panel with PCM for application in lightweight and retrofitted buildings. Energy and Buildings, 36(6), 567 - 578.

Meir, M. G., Rekstad, J. B., & Løvvik, O. M. (2002). A study of a polymer-based radiative cooling system. Solar Energy, 73(6), 403 - 417.

Pavlov, G. K. (2014). Building thermal energy storage. Kongens Lyngby: Technical University of Denmark.

STACK DRIVEN VENTILATIVE COOLING FOR SCHOOLS IN MILD CLIMATES: ANALYSIS OF TWO CASE STUDIES

Nuno M. Mateus[1], Guilherme Carrilho da Graça*[2]

1 Instituto Dom Luiz, Faculdade de Ciências, Universidade de Lisboa
Campo Grande, Ed. C8
1149-016 Lisboa, Portugal

2 Instituto Dom Luiz, Faculdade de Ciências, Universidade de Lisboa
Campo Grande, Ed. C8
1149-016 Lisboa, Portugal
**Corresponding author: gcg@fc.ul.pt*
Presenting author

ABSTRACT

This paper presents two case studies of stack driven ventilative cooling systems implemented in kindergarten schools located in the mild Subtropical-Mediterranean climate of Lisbon, Portugal. Both systems rely on stack driven natural ventilation supplemented by a larger, single-sided ventilation opening to be used in the warmer months. In both systems air enters the rooms at a low level, directly in front of the heating passive convector systems, and is exhausted in the back of the room, through a chimney. In addition to the smaller opening configuration, that is sized for the heating and mild seasons (1-3% of room floor area) both designs have larger openings to be used during the cooling season. This larger opening is fundamental to meet the minimum code requirement for total ventilation opening area (5% of floor area). The designs were developed and fine-tuned using dynamic thermal simulation (EnergyPlus). This approach allowed for straightforward statistical analysis of expected system performance, assessed in terms of thermal comfort and indoor air quality. Measurements in steady state mode show a good agreement between simulated and measured airflow rate. During the warmer months the smaller and protected heating season openings are opened for night cooling effect. The chimney exhausts were optimized to avoid opposing stack and wind effects. Both systems are user controlled. The importance of effective commissioning of passive systems is discussed and an example of a simple user manual is provided. The performance of these systems shows that a well-designed natural ventilation system can insure adequate levels of indoor air quality in kindergartens.

KEYWORDS

Ventilative cooling systems; natural ventilation; thermal chimney; EnergyPlus; dynamic thermal simulation.

1 INTRODUCTION

In the last decades, heating, ventilation and air conditioning systems (HVAC) assumed an important role in buildings energy consumption as the increasing demand for thermal comfort, combined with the rising time that people spend indoors lead to a significant increase in HVAC related energy consumption in services buildings (up to 50% of the total energy consumed in buildings, Zhao, 2012). In this context, natural ventilative cooling systems can be essential tool to limit this growth and reach the Nearly Zero Energy Buildings (NZEB) target.

Natural ventilation systems can be driven by buoyancy, wind or a combination of the two mechanisms, generating pressure differences across different spaces inside a building or between interior and exterior, driving airflow from high to low pressures zones.
There are three natural ventilation system geometries: displacement ventilation (DV), single-sided ventilation (SS) and cross-ventilation (CV). In DV, air is introduced near floor level where buoyancy forces induced by temperature differences (between inflow and room air temperature) make the colder air supply spread over the floor until it reaches a heat source, where it will expand and rises as a thermal plume. Ideally, the air movement induced by the thermal plumes, from low to high level, will be capable to transport heat and pollutants above the occupied zone, promoting a vertical temperature and contaminants stratification (Mateus, 2015).
Single-sided and cross-ventilation are characterized by mainly use the wind pressure on building shape to induce air currents through the openings on the façade. SS uses one or more windows in the same façade and is also affected by local buoyancy effects that can promote bi-directional flows (Wang, 2012). In contrast, CV uses openings in opposite façades (Graça, 2015). Although Cross-ventilation can provide larger flow rates there are implementation difficulties in kindergartens due to draft, noise and room configuration issues.
The application of natural ventilation concepts in schools has been extensively studied (Santamouris, 2010; Wang, 2014; Jamaludin, 2014). These studies identified issues that could affect system performance, such as: system control (manual or automatic), the unpredictability of weather conditions, noise and ingress of outdoor pollution into the indoor environment. In spite of these issues, in many cases, natural ventilation systems are capable of maintaining comfortable indoor temperature as well as acceptable CO_2levels.
This paper presents two case studies of stack driven user-controlled ventilative cooling systems for kindergarten schools in Lisbon (Portugal) that use a combination of DV and SS techniques. The thermal building simulation software EnergyPlus (EnergyPlus, 2013) was used in the design and fine-tuning of both natural ventilation systems.

2 CASE STUDIES

This section presents the two case studies analysed in this paper: CML Kindergarten (Figure 1) and German School (Figure 2). Both buildings are situated in urban area of Lisbon (Portugal). The CML Kindergarten was built in 2013, with a total built area of 680 m^2 distributed into two floors with 3.1m floor to ceiling height each. The building was south-west oriented, with capacity for 42 children, and each ground floor classroom have direct access to the exterior courtyard, partially covered (courtyard view, Figure 1).
The second case study, the German School of Lisbon, has a main façade facing north (to minimize solar heat gains impact), with two floors, and capacity for 160 children (1200m^2, 3.3m floor to ceiling height). The building structure is concrete, with external insulation, creating a high thermal inertia building. The windows are low-emissivity double glazed with solar control and movable blinds (interior in German school and exterior in CML kindergarten case). In south oriented areas overhangs were installed to control high solar gains. Both schools were designed with large glazed areas for allow the use of natural lighting.
These two schools are located in the mild Subtropical-Mediterranean climate of Lisbon, Portugal (Figure 3), characterized by mild winters (minimum temperature ≈ 4ºC) and dry summers with high levels of solar radiation (maximum temperature ≈ 37ºC). In spring and summer there are many days with large thermal amplitude (up to 18ºC), that can potentially make a night cooling approach very effective. In a typical schools building in Lisbon it is expected that the main comfort problems occur when high direct radiation levels and the

maximum outdoor temperatures are combined with high internal gains, easily leading to cooling loads of up to $100W/m^2$.

Figure 1: Inside, exterior and courtyard views of the CML Kindergarten.

Figure 2: Lateral, front and inside views of the German school.

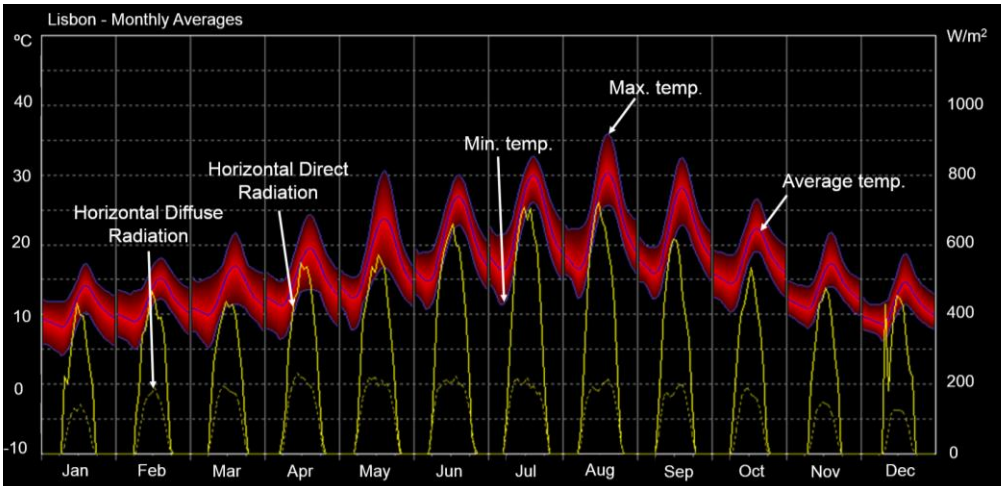


Figure 3: Typical year of Lisbon weather (outdoor temperature and radiation).

Both schools have no mechanical ventilation system installed, the fresh air is introduced into the spaces through low level grilles on the façade and will be exhausted in the back of the room, through a chimney. For the thermal conditioning of the spaces, the buildings Portuguese national code (RECS, 2013) only requires the installation of an active system for the heating period. For this purpose an hydraulic is installed in each classroom. For optimal performance of the ventilative cooling systems designed two operation modes were considered (winter and summer), as shown in Figure 4.

During heating period (winter mode), due to the buildings regulation impositions the airflow grilles should be opened to provide the required minimum airflow (fresh air) in order not to exceed CO_2 concentration limit (average below 1625ppm over an 8h period). In this mode,

the air that enters though the grilles and was pre-heated directly in front of the passive heating convector that maintain the interior air temperature always above 19ºC.
In summer mode (during the cooling period), all the openings on the façade (low level grilles and openable windows) will be available to be opened, in order to enable larger flow rates to remove the higher heat gains. Taking advantage of the exposed concrete structure and building thermal inertia, passive night cooling used to pre-cool the building during non-occupied periods. In both modes blinds are user-controlled and can be used to minimize solar gains.

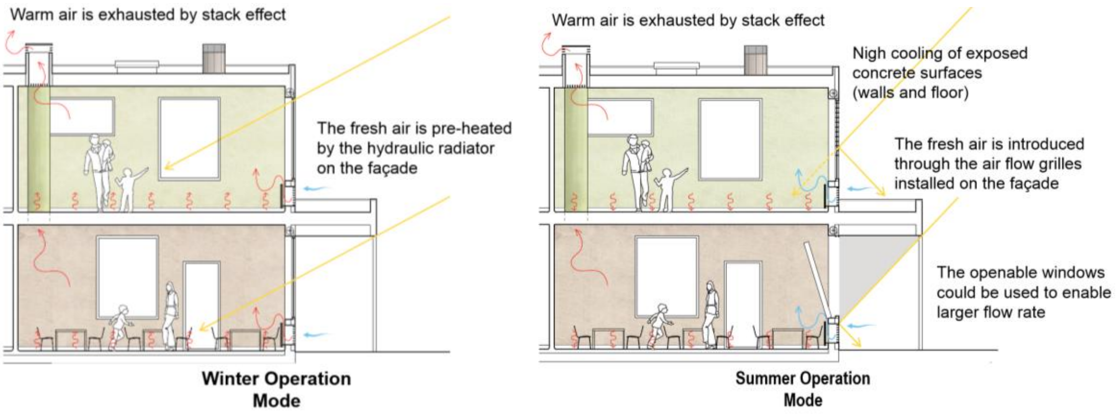


Figure 4: Kindergartens ventilative cooling systems operation modes (winter and summer).

3 METHODOLOGY - THERMAL SIMULATION

The dynamic thermal simulations were performed using the open source thermal building simulation software EnergyPlus. To simulate natural ventilation the airflow network approach (EnergyPlus, 2013) was used, modelling infiltration and openings in detail. This tool has been previously validated for natural ventilation of free-running buildings (Mateus, 2014; Zhai, 2011) and applied in SS (Wang, 2013; Schulze, 2013) and DV ventilation studies (Mateus, 2013).
Both kindergartens are shielded by surrounding buildings and for that reason wind effects were neglected and only buoyancy was considered in simulations (a conservative approach). In both cases, to analyse the performance of the natural ventilation systems only a representative classroom of each building was considered. These spaces including the principal features of natural ventilation systems: airflow grilles, thermal chimney and openable windows. Figure 5 shows the EnergyPlus model used to simulate CML Kindergarten and German School.

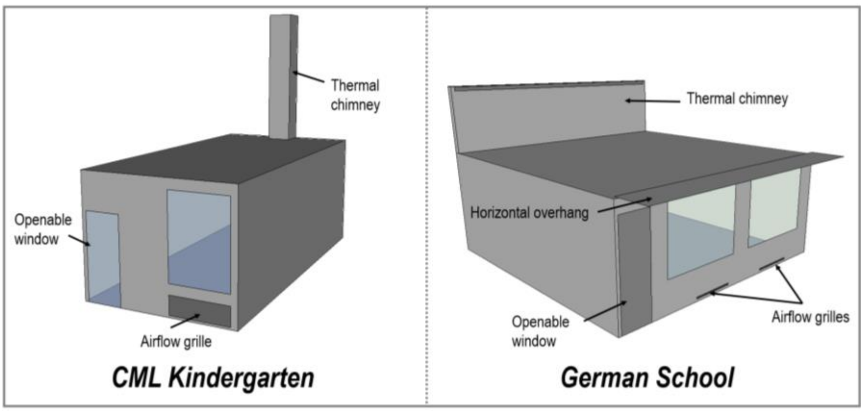


Figure 5: CML Kindergarten and German School EnergyPlus model.

In order to consider the thermal stratification effects both models are composed by two thermal zones (connected by a virtual horizontal window that is always open): room and thermal chimney.
The simulations were performed for a whole year, using the TMY weather file for Lisbon (EnergyPlus Weather), the rooms IAQ level promoted by the natural ventilation systems should be in agreement with buildings Portuguese code (RECS, 2013) and the users thermal comfort was analysed considering two international standards (CEN, 2007; ASHRAE, 2010):

- The rolling average of CO_2 concentration in 8 consecutive hours should not exceed 1625 ppm (RECS).
- Operative temperature range between 19-26ºC, (kindergartens limits, EN 15251).
- Adaptive comfort model (80% acceptability limits for naturally conditioned spaces, ASHRAE 55-2010).

In the simulation, the airflow grilles were open when the outdoor temperature is below the interior temperature. The hydraulic radiator is used during the heating months (from October to April) to ensure an interior air temperature above 19ºC. The openable windows will be used to increase the air change rates during the warmer months (from May to September) when additional heat removal will be needed. The Table 1 presents the internal heat loads considered for each case. Table 2 shows the size of opening areas considered for the winter and summer operation modes.
The smaller opening configuration is sized for the heating and mild seasons, corresponding to 1-3% of room floor area, while in the cooling season the total opening area should meet the minimum code requirement of 5% of floor area, that in the case of CML kindergarten reach the 8%.

Table 1: CML Kindergarten and German School heat loads scenarios used in simulation.

School	**Occupancy**	***Internal Gains***		
		Occupants (W/occupant)	**Lighting (W/m^2)**	**Total (W/m^2)**
CML Kindergarten	19 children + 1 adult	70 + 100	8	53
German School			7	33

Table 2: Opening areas summary.

Floor Area (m^2)		***German School***	***CML Kindergarten***
		55	32
Opening Area (m^2)	*Winter mode*	0.8	1.0
	Summer Mode	2.7	2.6
Max. Opening Area/Floor Area (%)	*Winter mode*	**1.5**	**3.1**
	Summer Mode	**5.0**	**8.1**

4 RESULTS: NATURAL VENTILATION SYSTEMS PERFORMANCE

This section presents the simulation results for the two kindergartens (CML on section 4.1 and German School on section 4.2). Figures 6-7 and 10-11 present the predicted performance for a typical day, in both operation modes (winter and summer). Finally, figures 8-9 and 12-13 show the predicted yearly operative temperature and indoor air quality (CO2 concentration), evaluated according to the EN 15251 and ASHRAE 55-2010 criteria.

4.1 CML Kindergarten

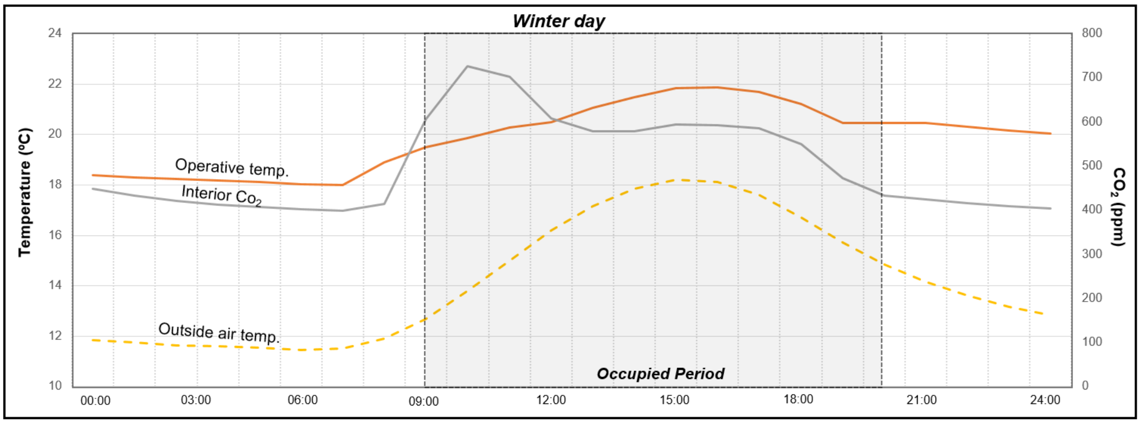


Figure 6: CML Kindergarten results: Operative temperature and CO_2 level (winter operation day).

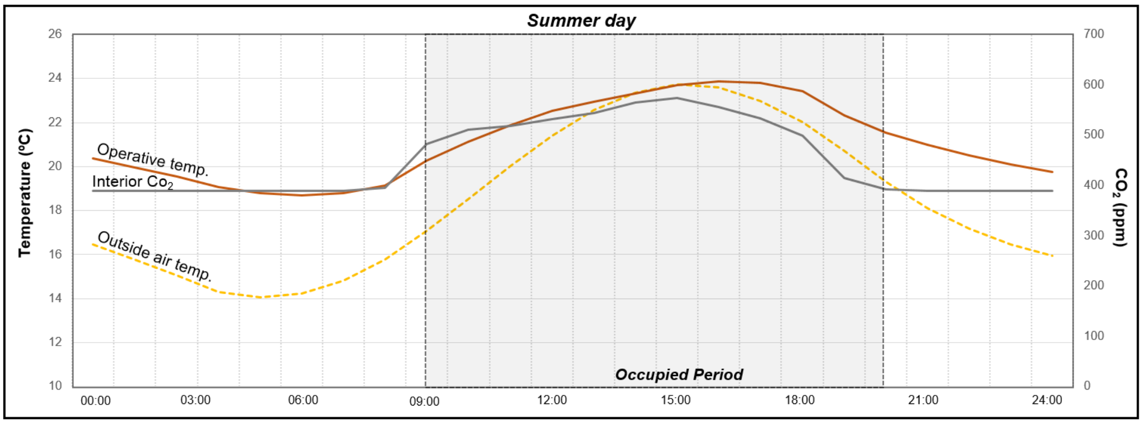


Figure 7: CML Kindergarten results: Operative temperature and CO_2 level (summer operation day).

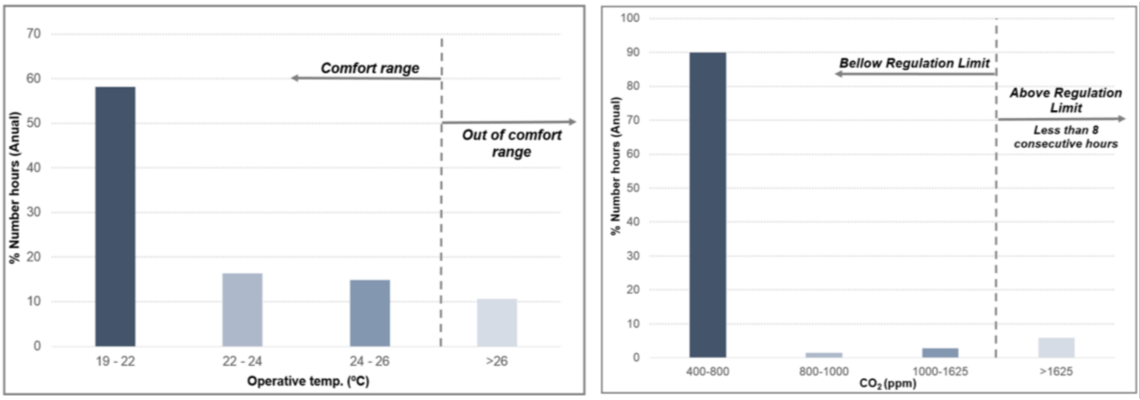


Figure 8: CML Kindergarten statistical analysis: operative temperature (EN 15251) and indoor air quality (RECS).

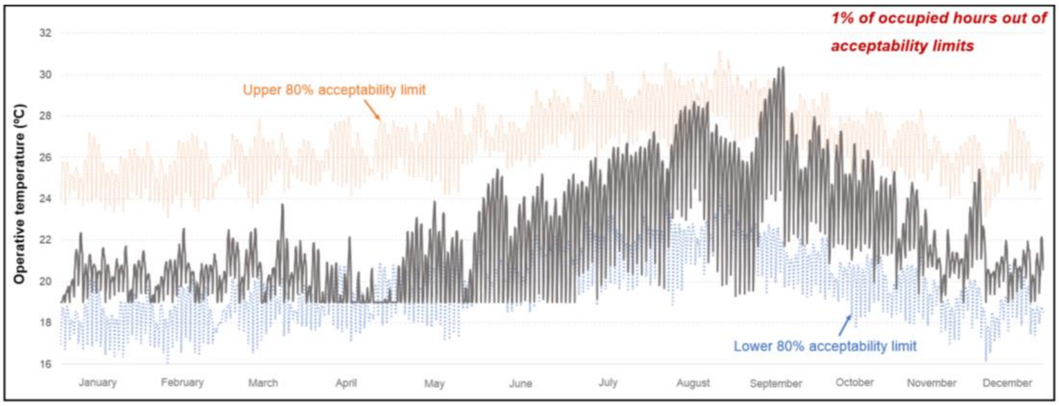


Figure 9: CML Kindergarten operative temperature adaptive comfort analysis (ASHRAE 55-2010).

4.2 German School

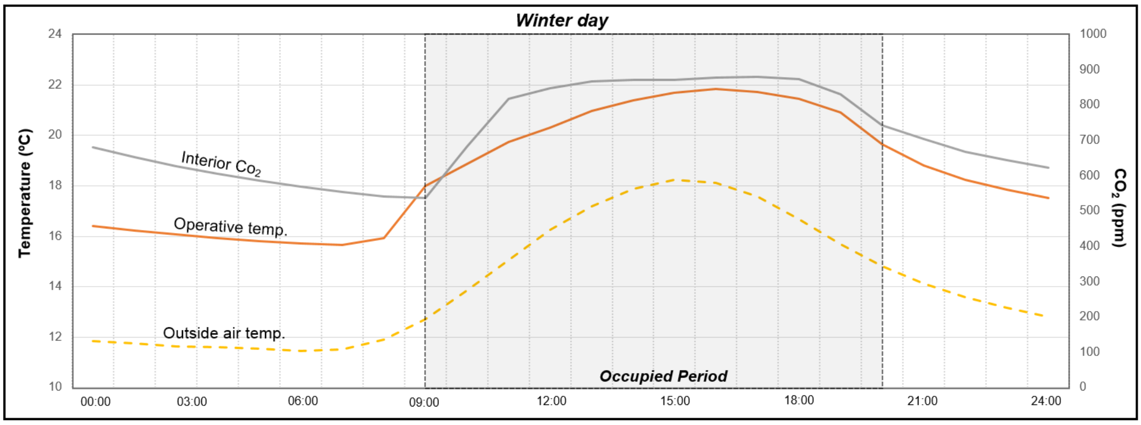


Figure 10: German School results: Operative temperature and CO_2 level (winter operation day).

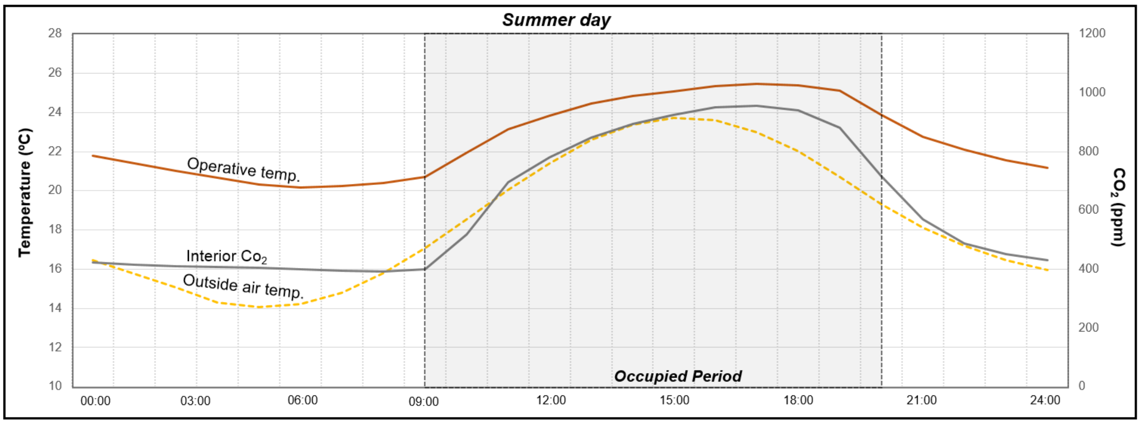


Figure 11: German School results: Operative temperature and CO_2 level (summer operation day).

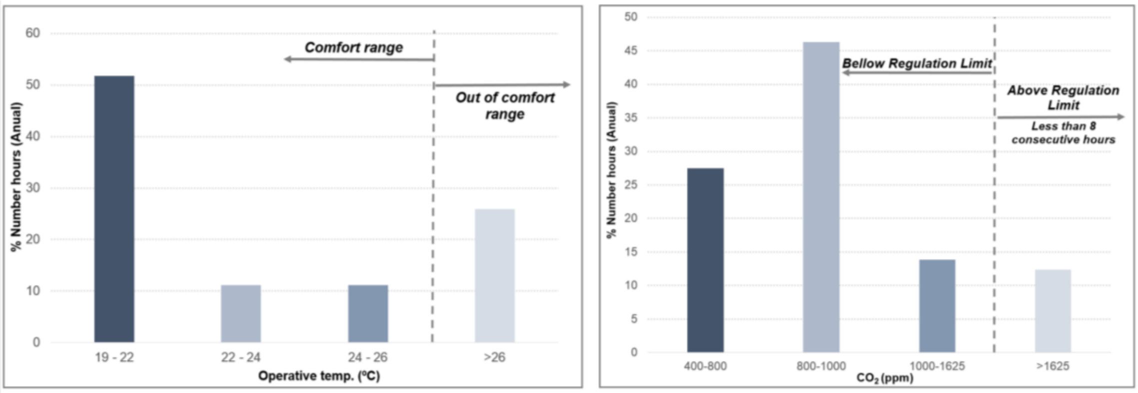


Figure 12: German School statistical analysis: operative temperature (EN 15251) and indoor air quality (RECS).

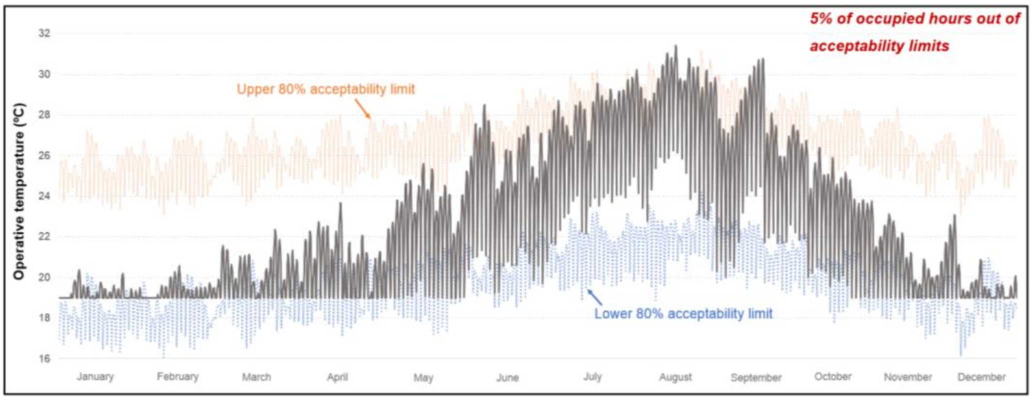


Figure 13: German School operative temperature adaptive comfort analysis (ASHRAE 55-2010).

The typical summer and winter operation days results (Figures 6-7 and 10-11) reveal that the system is capable to provide a natural ventilation airflow that is sufficient to ensure the thermal comfort and IAQ level in the majority of occupied periods. As expected, in both schools, when the occupants get into the space CO_2 concentration and operative temperature begin to increase until in the end of the working day. The night cooling approach effectively pre-cools the spaces until the beginning of the next occupied period. The proposed ventilative cooling designs meet the IAQ requisites defined by buildings Portuguese national code: the rolling average of CO_2 concentration in the 8 consecutive hours does not exceed 1625ppm.
The main problems occur in the summer, when it is necessary to promote the interior air renewal (to maintain acceptable CO_2 concentration) but the outside air is warmer. Ideally, in

these moments all the openings should be closed to achieve comfortable interior air temperature but open to do not exceed 1625ppm (CO_2 concentration). In these cases, the users will determine what comfort parameter is more relevant to his comfort and to define if the openings should be maintained closed or be opened.
Analyzing the annual operative temperature results using the adaptive comfort model shows that the occupants comfort is obtained 99% of the occupied hours in CML and 95% in German school. However, when these results are compared with those obtained when using the limits proposed by EN 15251 (19-26ºC) the CML kindergarten obtain similar performance results (10% hours in discomfort), but the German school occupants are expected to be out of comfort during 28% of occupied periods. This poor performance is probably due to the lower opening area per floor area ratio that is used in this design. In light of these results, the designers recommended the installation of an active cooling system in German School to be used in conjunction of natural ventilation system. This recommendation was not followed by the building manager. As a result there have been systematic user discomfort complaints in the warmer weeks of the year. These results indicate that the adaptive thermal comfort model may be "optimistic" in the proposed limits. To obtain a high performance ventilative cooling system (like in the CML kindergarten case), we recommend the use of EN 15251 instead of to ASHRAE 55-2010 standard to analyse the occupant's thermal comfort, due to the more restricted temperature limits.

4.3 Limited validation of flow rate prediction

This section presents a comparison between a steady state airflow measurement and the correspondent EnergyPlus simulation results for the German School case. The measurements were performed using very high internal sensible heat gains (6kW). Table 3 presents the comparison between predicted and measured airflow rate and the correspondent percentage error. The results of the comparison between simulation and measurements (performed in steady state mode) show good agreement: 4% error.

Table 3: Steady state airflow measurements vs EnergyPlus simulation.

Steady-state	**German school**
Outdoor-Indoor temp. dif. (K)	8.8
Inflow - measured airflow (m^3/h)	194
Simulated airflow (m^3/h)	187
Error (%)	**4**

5. CONCLUSIONS

In the two case studies presented in this paper, thermal and airflow simulation was used to fine-tune the designs and diagnose possible thermal comfort and IAQ problems. The opening sizes used in both designs depend on system operation mode: 1-3% of room floor for winter mode and 5-8% for summer mode. Both projects meet the requirements imposed by Portuguese buildings code CO_2 concentration bellow 1625ppm (during 8 consecutive hours).
The capability to meet thermal comfort goals depends on the criteria used: both designs perform well when assessed using the adaptive thermal comfort standard. However only the CML kindergarten meets the thermal comfort standards proposed by ASHRAE 55-2010 and EN 15251(1% and 10% of occupied hours out of limits, respectively). Limited user feedback indicates that the stricter assessment is more accurate. The German school, deemed adequate

by the adaptive thermal comfort analysis, has had excessive air temperature related user complaints since its inauguration (in 2008).

Sizing and controlling ventilative cooling systems is a complex task that is affected by many uncertainties that impact system performance. In this context, the use of a simulation tool such as EnergyPlus can be beneficial.

6. ACKNOWLEDGEMENTS

The authors are grateful for the support provided by the Fundação Calouste Gulbenkian (through Ph.D. Grant No. 126724) and FCT (Grant UID/GEO/50019/2013).

7. REFERENCES

Adi Ainurzaman Jamaludin, Hazreena Hussein, Ati Rosemary Mohd Ariffin, Nila Keumala, A study on different natural ventilation approaches at a residential college building with the internal courtyard arrangement, Energy and Buildings, Volume 72, April 2014, Pages 340-352, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2013.12.050.

ASHRAE, ASHRAE standard 55-2010. In: Thermal environmental conditions for human occupancy. ASHRAE Atlanta, GA; 2010.

CEN, CEedN., CEN Standard EN 15251. In: Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics. CEN: Brussels; 2007.

EnergyPlus (2013). Energy Plus Documentation: Getting Started with EnergyPlus, EnergyPlus Engineering Reference, Input and Output Reference.

G. Carrilho da Graça, N.C. Daish, P.F. Linden, A two-zone model for natural cross-ventilation, Building and Environment, Volume 89, July 2015, Pages 72-85, ISSN 0360-1323, http://dx.doi.org/10.1016/j.buildenv.2015.02.014.

Hai-xiang Zhao, Frédéric Magoulès, A review on the prediction of building energy consumption, Renewable and Sustainable Energy Reviews, Volume 16, Issue 6, August 2012, Pages 3586-3592, ISSN 1364-0321, http://dx.doi.org/10.1016/j.rser.2012.02.049.

Haojie Wang, Qingyan Chen, A new empirical model for predicting single-sided, wind-driven natural ventilation in buildings, Energy and Buildings, Volume 54, November 2012, Pages 386-394, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2012.07.028.

Haojie Wang, Qingyan Chen, A semi-empirical model for studying the impact of thermal mass and cost-return analysis on mixed-mode ventilation in office buildings, Energy and Buildings, Volume 67, December 2013, Pages 267-274, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2013.08.025.

Nuno M. Mateus, Armando Pinto, Guilherme Carrilho da Graça, Validation of EnergyPlus thermal simulation of a double skin naturally and mechanically ventilated test cell, Energy and Buildings, Volume 75, June 2014, Pages 511-522, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2014.02.043.

N. Mateus, G. Carrilho da Graça. 2013. Thermal and Airflow Simulation of the Gulbenkian Great Hall. Proceedings of 13th International Conference on Building Simulation, Chambery, France.

Nuno M. Mateus, Guilherme Carrilho da Graça, A validated three-node model for displacement ventilation, Building and Environment, Volume 84, January 2015, Pages 50-59, ISSN 0360-1323, http://dx.doi.org/10.1016/j.buildenv.2014.10.029.

RECS, Regulamento de Desempenho Energético dos Edificios de Comércio e Serviços, Decreto-Lei nº 118/2013 de 20 de Agosto. Diário da República nº159 - Ministério da Economia e do Emprego, Lisboa, 2013.

M. Santamouris, A. Synnefa, M. Asssimakopoulos, I. Livada, K. Pavlou, M. Papaglastra, N. Gaitani, D. Kolokotsa, V. Assimakopoulos, Experimental investigation of the air flow and indoor carbon dioxide concentration in classrooms with intermittent natural ventilation, Energy and Buildings, Volume 40, Issue 10, 2008, Pages 1833-1843, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2008.04.002.

Tobias Schulze, Ursula Eicker, Controlled natural ventilation for energy efficient buildings, Energy and Buildings, Volume 56, January 2013, Pages 221-232, ISSN 0378- 7788, http://dx.doi.org/10.1016/j.enbuild.2012.07.044.

Yang Wang, Fu-Yun Zhao, Jens Kuckelkorn, Xiao-Hong Li, Han-Qing Wang, Indoor air environment and night cooling energy efficiency of a southern German passive public school building operated by the heat recovery air conditioning unit, Energy and Buildings, Volume 81, October 2014, Pages 9-17, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2014.06.008.

Zhiqiang (John) Zhai, Mary-Hall Johnson, Moncef Krarti, Assessment of natural and hybrid ventilation models in whole-building energy simulations, Energy and Buildings, Volume 43, Issue 9, September 2011, Pages 2251-2261, ISSN 0378-7788, http://dx.doi.org/10.1016/j.enbuild.2011.06.026.

VENTILATIVE COOLING AND ENERGY USE IN SUPERMARKETS

Zoi Mylona*, Maria Kolokotroni, and Savvas Tassou

RCUK National Centre for Sustainable Energy Use in Food Chains (CSEF)
Brunel University London
Kingston Lane, Uxbridge, UB8 3PH, UK
**Corresponding author: zoi.mylona@brunel.ac.uk*

ABSTRACT

Supermarkets are a category of non-domestic buildings with high energy use because of their operation. Recent work indicates that by improvements to the energy delivery systems through which internal environmental conditions are maintained such as thermal properties of external envelope including airtightness, HVAC systems and lighting, substantial energy savings can be achieved. Work to date has focused on typical supermarkets while the present paper examines frozen food supermarkets which include more refrigeration cabinets and therefore result in higher energy use per sales floor area. The work is based on measured energy and environmental data from a newly built supermarket in South London that was used to create and optimise an energy and thermal model of the supermarket using EnergyPlus. With the calibrated model, a parametric analysis was carried out to determine strategies for improved energy performance. Results indicate that changes in the operational times of the existing HVAC system, as well as different temperature set points in the sales area can lead to energy savings of 6.5%. Night Ventilation has the potential for energy demand reduction across the majority of the operating systems and could save 81 kWh/m^{2}sales area annually and up to 51% cooling energy. Improved envelope airtightness is also being investigated and in combination with thermal insulation retrofits, a further reduction is predicted to the energy demand. Light intensity monitoring data have indicated that additional energy savings can be achieved by introducing a daylighting control strategy. Compared to the baseline supermarket model as it operates currently, the above changes can reduce the energy use and CO_2 emissions by 13.1% annually. Finally, the potential of the implementation of different HVAC systems (CAV and VAV) to the store indicated that the VRF system can maintain more efficiently the indoor air temperature of the store with the minimum total energy use. Night Ventilation strategies through r the different HVAC systems showed that the CAV system presents higher dependence on the air flow rates of night ventilation.

KEYWORDS

Supermarket, Energy Use, HVAC, Night Ventilation, EnergyPlus

1 INTRODUCTION

Supermarkets are energy intensive buildings and have a major impact on energy demand compared to other commercial buildings. In the USA they represent a 5% of total commercial building primary energy use (Clark, 2015). According to Enova statistics (2008), supermarkets consume nearly twice the energy of office buildings. Electricity used for heating, ventilation and air conditioning (HVAC), hot water and refrigeration covers about 4% of the country total in USA and France and 3% in Sweden. Currently, there are over 1 million supermarkets in Europe (CREATIV, 2014). Thus, just a small percentage reduction in energy use can result to substantial savings. A 25 % energy saving in supermarkets in Europe will result in 31 TWh of annual electricity savings which equates to carbon reductions of 16.2 million tons CO_2 (CREATIV, 2014).

Recent work in the UK indicates that supermarkets have improved energy efficiency resulting from improvements to the energy delivery systems. According to (Sullivan, 2013) the majority of the UK supermarkets achieved a reduction to their emissions from 2005 to 2010. Analysis results from NREL (2015) demonstrate significant energy savings with the implementation of advanced HVAC strategies.

Literature review reveals that there is limited information on the energy use by sub-systems in supermarkets based on operational data. Available information indicated that refrigeration uses the largest percentage of energy followed by lighting and HVAC (Mavromatidis, 2013) (Tassou et al., 2011). A recent study shows that refrigeration accounts for about 40% of total energy use and lighting for 25% (Pearson, 2014). Therefore, significant savings can be achieved by improving the efficiency of these systems.

Supermarkets operate in a way that is quite different from other building types and require unique HVAC strategies (Clark, 2015). In general, energy consumption for HVAC and lighting depends on several parameters such as size of sales area, construction, opening hours, products nature, occupancy levels and external weather conditions. Mavromatidis (2013) suggests that one of the main indicators for energy use by supermarkets is the ratio of food products against total products; the greater the ratio , the more energy is consumed.

This paper presents a study for the energy use reduction of the HVAC and lighting systems of a medium size frozen food supermarket with a high ratio of food product to other products (approximately 9:10) and a high percentage of chilled and refrigerated products (approximately 51% of total products). It will focus on the potential of night ventilation for energy savings through the existing HVAC system and will also examine the impact of external building envelope air-tightness. Few studies to date have considered ventilative cooling strategies for supermarkets (Li-Xia Wu et al., 2006) although the potential is high as will be discussed in this paper.

2 CASE-STUDY BUILDING

A newly built ground floor supermarket, opened in June 2013 is the case under study. The store (315m^2 sales floor area) belongs to a UK frozen food supermarket chain and is located in South London, in a commercial area. Its opening hours are from 8:30- 20:00 on weekdays and Saturdays and 10:00 to 16:00 on Sundays. The store includes the sales area (tills and display area), cold storage rooms and auxiliary spaces (storage, office, restrooms, staffroom) as well as a large roof void area. The building is constructed according to current building regulations (Part L) in the UK (Table 1) (Part L, 2014). The main entrance of the store (northwest) and the southwest side are glass while the northeast side is adjacent to another supermarket.

Table 1: Limiting fabric parameters

	U-value (W/m^{2}K)
External Wall	0.35
Ground Floor	0.25
Roof	0.25
Windows	5.7*
*Single Glazed Windows	

Table 2: Refrigeration Equipment

	RefrigerationEquipment No	**Power / unit (W)**
Open front multi-deck chilled food	7	1460
Lift up lid frozen food	58	200
Open top case frozen food	3	1345
Coldroom Chiller	1 (29.4m^2)	
Coldroom Freezer	1 (5.8m^2)	

The store is all electric from grid electricity. The annual energy consumption is 1102.8 kWh/m^2 sales area which is within the range identified for stores with sales area 180m^2-

$1400m^2$ in the UK (850 to 1500 kWh/m^2 per year) (Tassou et al., 2011). The refrigerated display fixtures consist of three different refrigeration cabinets; (a) chilled food open front multi-deck cabinets, (b) lift up lid and (c) open top case frozen food cabinets. One freezer ($29.4m^2$) and one chiller ($5.8m^2$) coldrooms are used for storage purposes. Lighting is provided by fluorescent lamps during opening hours and LED strips in the sales area during the night. The HVAC is a Variable Refrigerant Flow (VRF) system used for both heating and cooling. Two equally sized outdoor condensing units provide total heating output of 113kW and cooling output 101kW which are delivered to the tills and display areas only, through 7 ceiling cassettes and 1 door heater. The design cooling duty requirements of the store was estimated at 60kW sensible. The HVAC system is operated 24h with 20-21^oC set-point temperature for both cooling and heating. Ventilation rates have been set at 6ach for tills and display area, 10ach for restrooms and 1ach for storage area. The airtightness is designed at $15m^3/h\ m^2$ @50Pa according to the current UK building regulations.

3 BASELINE MODEL DEVELOPMENT

3.1 Software and operational data

Half hourly energy data is available since the opening of the store (June 2013). Moreover, additional sub metering of different systems has been acquired from similar stores of the same supermarket chain.

Environmental conditions (air temperature, relative humidity, light intensity and CO_2 levels) have been monitored at 15mins interval for one year at different locations and heights in the store using 21 HOBO data loggers (Tempcon Instrumentation Ltd.). In addition, data were collected of air temperature of the 7 diffusers using i-button data loggers (Measurement Systems Ltd). For the present paper these data were used to calibrate the Energy Plus model developed.

EnergyPlus (Version 8.1) (U.S. Department of Energy, Energy Efficiency & Renewable Energy) was used to model the energy and environmental performance of the store. EnergyPlus is based on the most popular features and capabilities of BLAST and DOE-2 (Crawley et al., 2001). It also incorporates many advanced features, such as multi zone airflow and extensive HVAC specification capabilities (Coakley et al., 2014) including refrigeration systems. The refrigeration system capability within EnergyPlus focus on properly accounting sensible and latent energy exchanges between the refrigerated cases and the building HVAC systems. It also includes a model for walk-in coolers (coldrooms) exchanging energy with multiple conditioned zones. Secondary loops, shared condensers and sub coolers are also included as well as a library of data for different refrigerants (Stovall et al., 2010)

External conditions were simulated by using an *.epw weather file custom created with data from Gatwick airport weather station and year specific data from Weather Underground (www.wundergound.com) to correspond to the period of available operational data.

3.2 Model parameters input

Mechanical and architectural design details were available through the energy manager of the building with descriptions of the fabric properties, electrical, refrigeration and HVAC services distribution, control strategies and customer occupancy. In addition, in-store observations were carried out by the authors for typical weekdays and Saturdays during July 2013 to determine customer flow numbers. This varies between 15-120/hour with an average of 90/hour. Figure 5 presents the daily schedules for lighting, electrical equipment, customers' activity created in combination with the in store observations and refrigeration schedules. A summary of the parameters imported in EnergyPlus is shown in Table 4. The 3D construction model of the store designed in Google Sketch Up and through Open Studio Plug-in was transferred to EnergyPlus. The store is separated in 9 thermal zones.

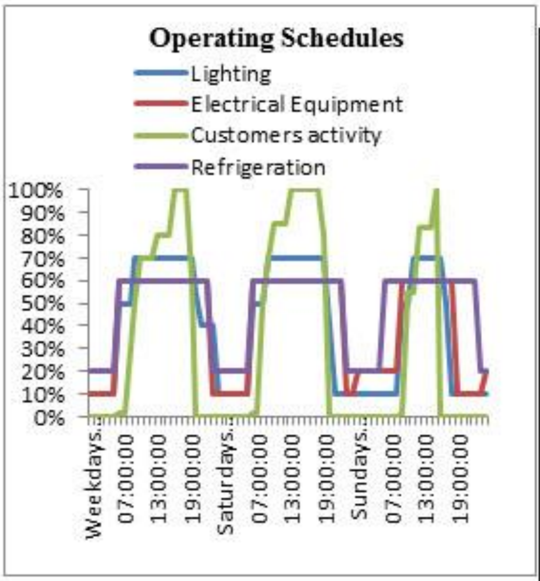


Figure 5: Operating Schedules (Lighting, Electrical Equipment, Customers activity, Refrigeration)

Table 4: Summary of parameters input

Input parameters	Value
HVAC System	VRF with 7 heat pump cassettes and door heater
Heating	
Heating Setpoint Temperature	20°C (non-operating hours) 21°C (opening hours)
Cooling	
Cooling Setpoint Temperature	20°C (non-operating hours) 21°C (opening hours)
Ventilation	6ach, Tills area, Sales area,
Infiltration	15m^3/hm^2 @50Pa
Lighting	16 W/m^2 Tills area 32.9 W/m^2 Display area
Electrical Equipment	16 W/m^2 Tills area 274.2 W/m^2 Kitchen 137.3 W/m^2 Restrooms 25.3 W/m^2 Office
Customers	7.6 m^2/person Tills area 16 m^2/person Sales area

3.3 Accuracy criteria

Due to complexity of the building and the dependency of independent interacting variables, it is difficult to achieve an accurate representation of the store. By calibrating the model to measured data, a more accurate and reliable representation of the building is achieved (Coakley et al., 2014). Kaplan et al. suggest calibrating models to short typical periods and not to annual data, for example to monthly data (Kaplan et al., 1990).
ASHRAE Guideline 14-2002 defines the evaluation criteria to calibrate a simulation model. Monthly and hourly data, as well as spot and short term measurements can be used for calibration. Mean Bias Error (MBE) and Coefficient of Variation of the Root Mean Squared Error (CVRMSE) are used to evaluate the model uncertainties (ASHRAE, 2002).

$$MBE = \frac{\sum_{i=1}^{N}(y_i - \hat{y}_1)}{\sum_{i=1}^{N} y_i} \tag{1}$$

$$CVRMSE = \frac{\sqrt{\sum_{i=1}^{N}(y_i - \hat{y}_1)^2 / N}}{\bar{Y_s}} \tag{2}$$

$$\bar{Y_s} = \frac{\sum_{i=1}^{N} y_i}{N} \tag{3}$$

where, y_i is the measured data; $\hat{y}_1$ is the simulated data; N is the sample size; and $\bar{Y_s}$ is the sample mean of measured data.

ASHRAE (2002) recommends an MBE of less than 5% and a CVRMSE of less than 15% relative to monthly calibration data. If hourly calibration data are used, these requirements could be 10% and 30% respectively

Monthly simulation results for the case study model have shown a MBE of 1% and CVRMSE of 3%. Table 5 presents the hourly simulation results evaluation. The estimated % error with negative values means that the results from modelling are higher than results from the measurements and vice versa for positive values, whereas CVRMSE values are always positive.

Table 5: Hourly Model calibration results

	MBE	CVRMSE		MBE	CVRMSE
June 2013	1%	14%	December 2013	1%	15%
July 2013	2%	15%	January 2014	0%	13%
August 2013	1%	13%	February 2014	0%	12%
September 2013	0%	14%	March 2014	-2%	14%
October 2013	2%	12%	April 2014	-3%	15%
November 2013	0%	13%	May 2014	-3%	14%

4 RESULTS AND DISCUSSION

4.1 Energy efficient retrofit analysis and ventilation features

The HVAC control strategy is changed to facilitate night ventilation as follows: operation between 6:00 to 23:00 for weekdays and Saturdays and 9:00 to18:00 for Sundays rather than 24h of the baseline model. This change alone would save 81 kWh/m^2 sales area per year. A previous study for night ventilation implementation to a supermarket has concluded that longer night cooling activation results to fewer hours of AC system operation and higher energy savings (Li-Xia Wu et al., 2006). However, studies for offices and other non-domestic building have indicated that three control aspects should be taken into consideration (Kolokotroni, 1998); duration, system initiation and system continuation in order to maximise energy savings. In this case study, the following rules were implemented:

(A) Initiation: $T_{out}<T_{in}$,

(B) Continuation: $T_{out}<T_{in}$ and $T_{out}-T_{in}<T_{offset}$,

(C) Termination: continuation rule and $T_{in}=T_{min,}$

The continuation rule ensures that the outside air that brought in is effective on cooling the building. When the temperature difference between inside and outside air is low, the air brought in will have little effect on cooling while the ventilation fan energy use will increase the total energy use. However, if the outside air temperature is significantly lower than the inside air temperature, T_{min} will be achieved fast and the duration of night ventilation is decreased (Aria & Akbari, 2007).

Figure 6 shows the total energy use after the implementation of night ventilation using the above rules. T_{offset} of 1, 2, 3, 4, 5 and 8 ^oC, T_{min} of 12^{o}C and 15^{o}C and air flow rates of 4, 6 and 10 ach were tested. 10ach results to the higher reduction in total energy use, increasing with higher T_{offset} until it reaches 5^{o}C. 6ach also results to the lowest energy use when T_{offset} is 5^{o}C; after that point cooling energy use increases and this leads to an increase of the total energy use (Figure 7). 4ach results to minor changes with T_{offset} changes 0^{o}C to 5^{o}C. As in the other cases total energy use increases with T_{offset}> 5^{o}C. These results indicate that the optimum energy savings can be achieved for this system when T_{min} is12^{o}C, T_{offset} is 5^{o}C and the air flow rate during night ventilation is 6ach.

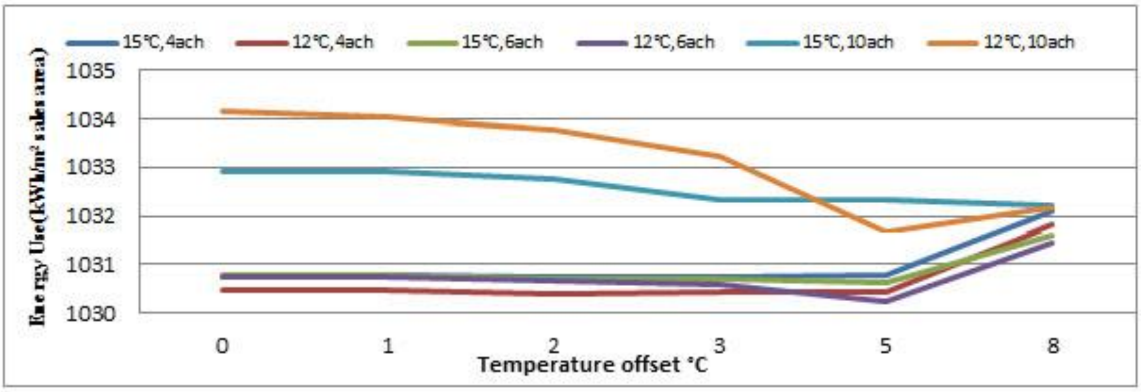


Figure 6: Annual Energy Use per sales area for different minimum indoor air temperature setpoints and different air flow rates of night ventilation in relation to the temperature offset

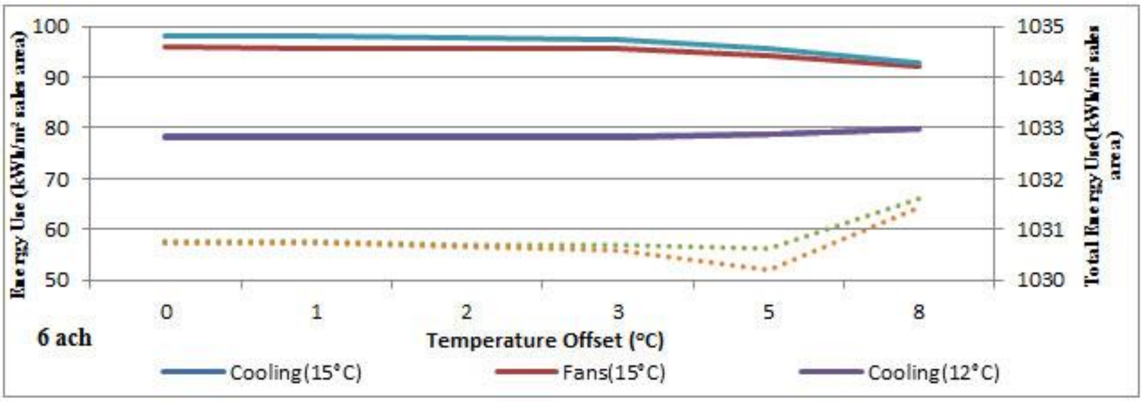


Figure 7: Annual Energy Use per sales area for cooling, fans and total for different minimum indoor temperature setpoints in relation to the temperature offset and 6ach air flow rate

Figure 8 presents the energy use reduction for each system due to different air flow rates of night ventilation. HVAC is mostly affected (around 40.1 %) because of the cooling demand reduction. Although night ventilation leads to added energy consumption of fans, this addition is smaller than the energy use reduction of the cooling demand.

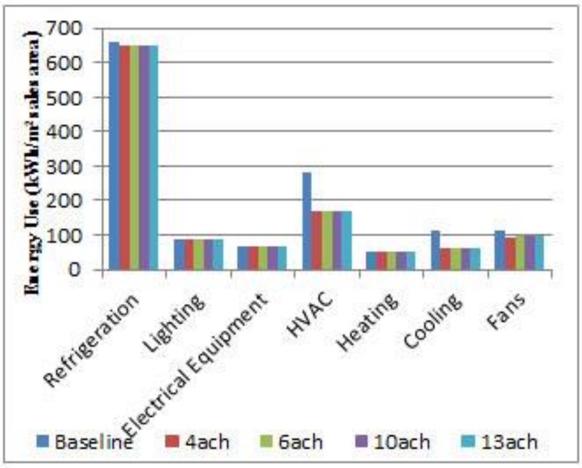


Figure 8: Annual Energy Use per sales area changes of the systems of the store due to different air flow rates of night ventilation

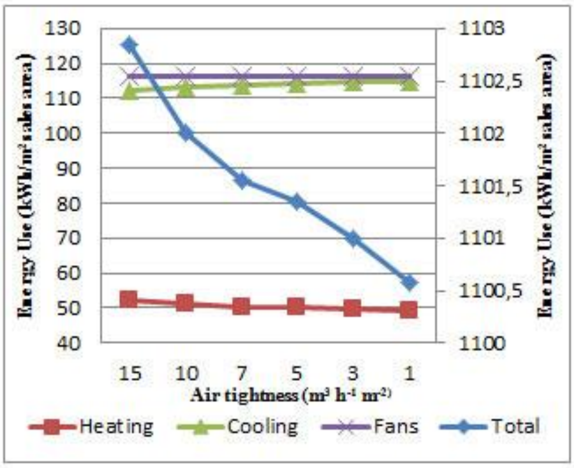


Figure 9: Annual Energy Use per sales area for heating, cooling, fans and total for different air tightness values

In the UK, air tightness tests are mandatory for buildings with floor are of more than $1000m^2$ and should be less than a maximum air permeability of 10 m^3 h^{-1} m^{-2} at a test pressure of 50Pa. However for buildings less than $500m^2$ total useful area, like the case study store, a test is not necessary; air permeability is taken as 15 m^3 h^{-1} m^{-2} at 50Pa (Part L, 2014). According to previous studies the desirable design value for low carbon supermarkets is 5 m^3 h^{-1} m^{-2} (Kolokotroni et al., 2014). Values of 15, 10, 7, 5, 3 and 1 m^3 h^{-1} m^{-2} were tested and presented in Figure 9. Increased air tightness results in a reduction in the total energy use but an increase in the cooling demand due to lower heat losses through the envelope.

Table 6 presents all changes implemented at the baseline model for optimum energy savings including high energy efficient lamps, daylight controls and envelope amendments.

Table 6: Summary of percentage energy savings of the total annual energy use per sales area

Input parameters	Value	Energy Use savings
HVAC System	6:00-13:00 weekdays and Saturdays	
	9:00-18:00 Sundays	
Heating Setpoint	19°C winter	
Cooling Setpoint	21°C summer	7.3%
Night Ventilation	Minimum indoor air temperature 12°C	
	Delta temperature offset 5°C	
	Air flow rate 6ach	6.6%
Lighting	Daylight control for 750 Lux	3%
	LED lighting in tills area and sales area	4.5%
	16 W/m^2 Tills area	
	9 W/m^2 Display area	**Total Lighting** 5.8%
Envelope	Double glazed windows	
	Insulation of the interior ceiling	2.9%
Total		13.1%

It was found that a 60% reduction is achieved in the lighting system and 29% in the HVAC; the cooling demand is reduced by 29%, the heating demand by 25% and the fans energy use by 17%. A 4% reduction was observed in the refrigeration system. Figure 10 presents the annual energy breakdown of the store after the implementation of the energy efficient amendments. Considerable amounts of energy are consumed by the refrigeration (66%), while the HVAC system consumes approximately 22% of the energy with the cooling energy use to be the most demanding of the HVAC system.

Figure 11 presents a daily energy profile to demonstrate the interactions between the systems. It shows a reduction of the baseline hourly energy use after the implementation of the changes (Table 6) to the baseline model. When the store is closed, it operates at base load which is 10kWh lower than the baseline. At 6:00 a peak is observed due to systems start up (HVAC and lighting). After 8:30 that the store is open, the energy use remains relatively constant during trading hours and starts reducing when the store is closed until 23:00 when HVAC and lighting system are turned off.

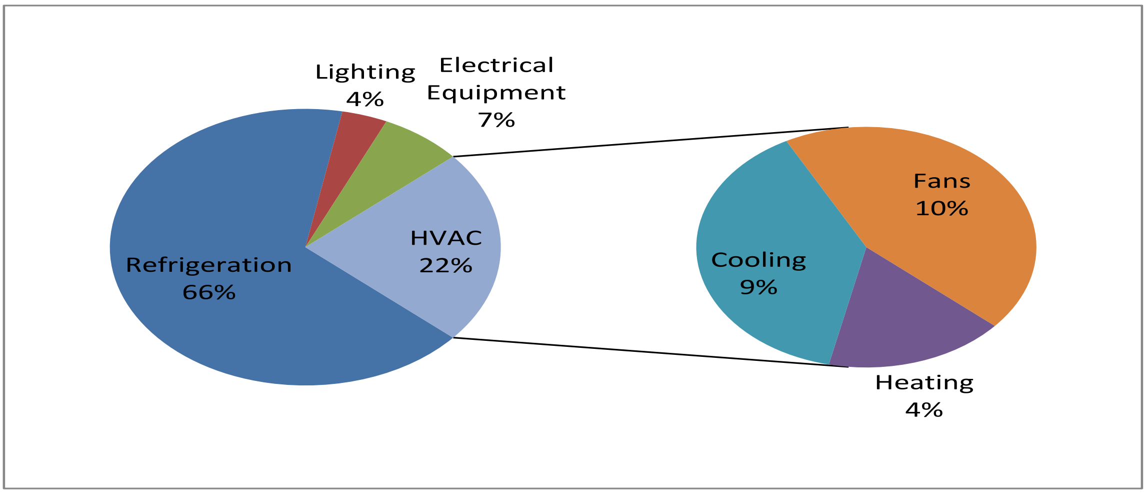


Figure 10: Percentage contribution of systems to the total annual energy use

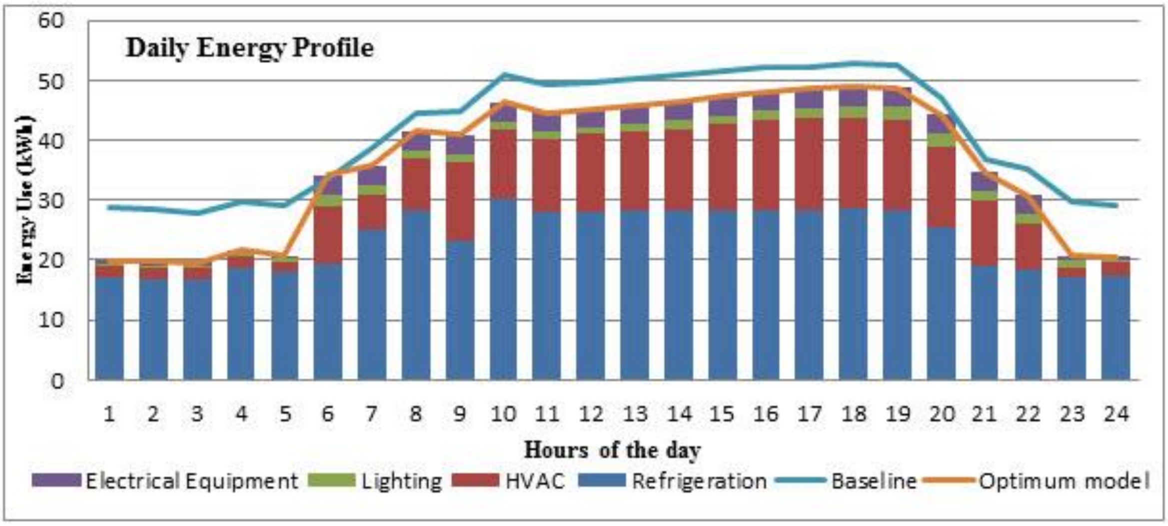


Figure 11: Typical summer weekday energy use profile of the store (Thursday 11/7)

4.2 Comparison of different HVAC systems

In general supermarkets present a challenge for air conditioning because of the interaction between the HVAC system and the refrigerated display cabinets which provide significant sensible cooling and increase the latent load fraction on the HVAC system (Tassou et al., 2011). According to the results presented in section 4.2, the HVAC system (VRF) energy use is between 22% and 24% of the total. However the most common system for supermarkets is the air constant volume (CAV). Apart from the CAV system, the variable air volume (VAV) HVAC system also simulated for the store under study.
Figure 12 presents a summary of the comparison between the different HVAC systems. The total annual energy use with a CAV HVAC system is higher by approximately 33% in comparison to the total annual energy use with a VRF system. A VAV differs at around 19% from the total annual energy use with VRF. However, if the optimal changes (Table 6) are implemented to the model with VAV, then the total annual energy use is approximately the same with the total annual energy use of the store as it is in operation today.

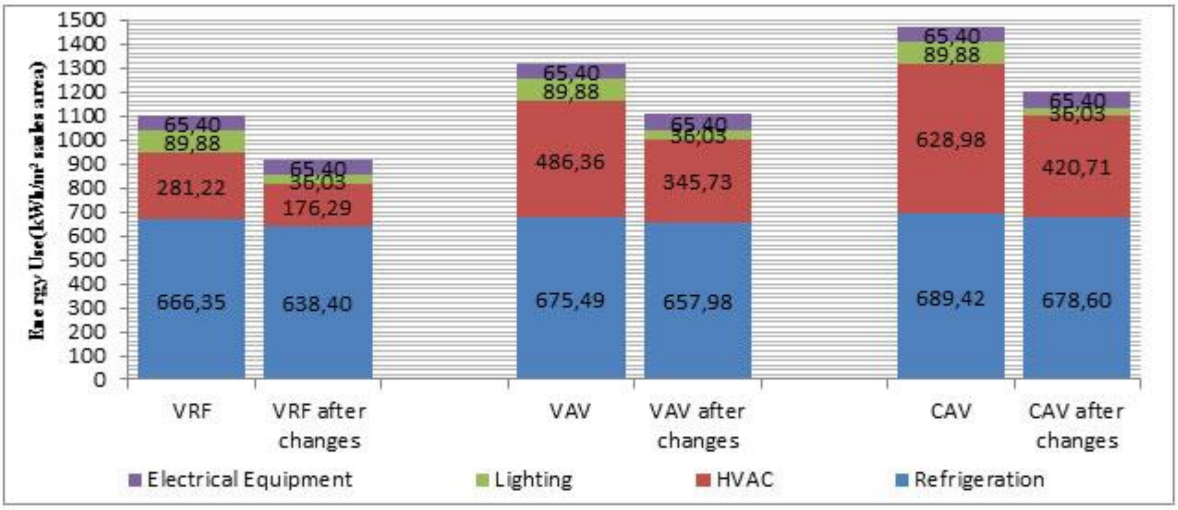


Figure 12: Annual Energy Use per sales area breakdown of electrical equipment, lighting, HVAC and refrigeration for total energy use of each HVAC system before and after the optimal changes of Table 6

Figure 13 presents the indoor air temperature annually for the baseline system models before the energy efficiency changes. It shows that indoor air temperature of the tills and display area is better maintained by a VRF system with the minimum total annual energy use per sales area.

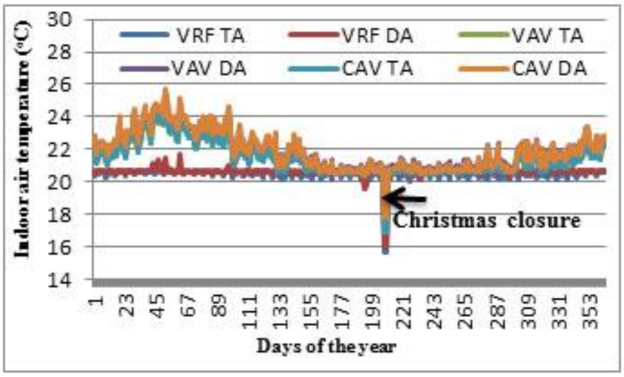


Figure 13: Indoor air temperature for Tills area (TA) and Display area (DA) during 1 year (1/6-31/5)

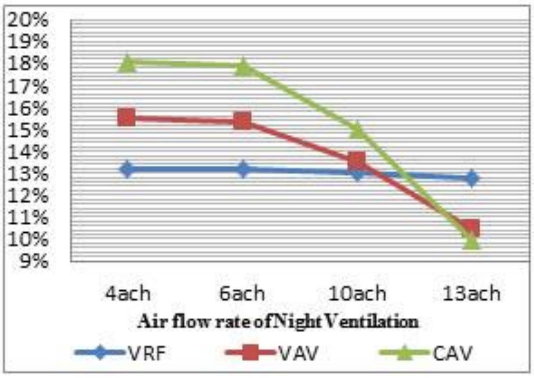


Figure 14: Percentage reduction of the total annual energy use per sales area of the different HVAC systems in relation to air flow rates of night ventilation.

Night ventilation was also investigated for CAV and VAV in comparison to VRF. Figure 15 presents the percentage reduction of the total annual energy use per sales area due to the night ventilation and its correlation to the air flow rate. The total annual energy use with the VRF system is observed to be approximately independent (13.2%-12.87%) of the air flow rate of night ventilation with 4 or 6 ach the best options, while the higher the air flow rate, the smaller the total energy use reduction is. 4 ach also achieve the higher energy savings for CAV and VAV; the higher energy use reduction is achieved with CAV (18.1%-10%). Figure 15 also indicated that CAV and VAV present a higher sensitivity to the increase of the air flow rates of night ventilation.

5 CONCLUSIONS

This paper presented results of energy use of a UK frozen food supermarket store which is used as a baseline for the investigation of energy efficiency measures for building fabric, lighting and ventilative cooling. EnergyPlus was used for the simulation of all scenarios with the baseline model calibrated using energy and environmental operational data from the case-study store. The papers focus in more detail on the HVAC system and night ventilation strategies. Parameters such as air flow rate, temperature offset and minimum indoor air temperature were investigated. The results for different air flow rates showed that total energy use is fairly flat in the range 0-6 ach with optimal flow rate to be around 6 ach in the case study building model with VRF system while cooling energy demand decreases with the increase of night ventilation air flow rate. Night ventilation can lead up to 6.6% total energy savings. Simulations also showed that the temperature offset for achieving highest energy savings is 5^{o}C.
Other energy efficient measures examined indicate that lighting system can achieve the biggest reduction (60%) when using LED lighting fixtures. Significant is the reduction of the HVAC system energy use (29%) due to night ventilation and the change of the indoor air temperature setpoints for summer and winter periods due to a cooling demand reduction.

The comparison of three HVAC systems (VRF, VAV and CAV) indicated that VRF results in the lowest total annual energy use per sales area with better maintenance of indoor air temperature. However, the VAV system could reach the same performance after the energy efficient measures discussed in the paper with the VRF system that is currently in operation. A further investigation of the influence of night ventilation on the performance of the different HVAC systems indicates that percentage highest savings can be achieved with the CAV system as it is the most inefficient of the three examined in this paper.

6 ACKNOWLEDGEMENTS

This work is carried out as part of the RCUK Centre for Sustainable Energy Use in Food Chains (EP/K011820/1) project. Thanks are due to Rick Jenkins for providing energy data and facilitating access to the case-study store.

7 REFERENCES

Aria H & Akbari H. (2007). A predictive Nighttime ventilation algorithm to reduce energy use and peak demand in an office building. Energy and Power Engineering,7,1821-1830.

ASHRAE. (2002). ASHRAE Guideline 14-2002, Measurement of Energy and Demand Savings.

Clark J. (2015). Energy-Efficient Supermarket Heating,Ventilation, and Air Conditioning in Humid Climates in the United States. NREL, National Renewable Energy Laboratory.

Coakley D, Raftery R, Keane M. (2014). A review of methods to match building energy simulation models to measured data. Renewable and Sustainable Energy Reviews. Elsevier,37, 123-141.

Crawley D B, Hand J W, Kummert M, Griffith B T. (2008). Contrasting the capabilites of building energy performance simulation programs. Building and Environment, Elsevier, 43, 661-673.

Crawley D B, Lawrieb L K,Winkelmannc F C, Buhlc W F, Huangc Y G, Pedersend C O, Strandd R K, Liesend C O, Fishere D E, Wittef M J, Glazerf J. (2001). EnergyPlus: creating a new-generation building energy simulation program. Energy and Buildings, Elsevier, 33, 319-331.

CREATIV, Industry Energy Efficiency. (2014, February 14). The future energy effective supermarket. Retrieved from SINTEF: http://www.sintef.no/projectweb/creativ/the-future-energy-effective-supermarket/

Geros V, Santamouris M, Tsangrasoulis A, Guarracino G (1998). Experimental evaluation of night ventilation phenomena. Energy and Buildings, Elsevier, 27, 141-154.

Kaplan M, Jones B, Jansen J. (1990). DOE-2.1 C model calibration with monitored end-use data. ACEEE 1990 Summer Study Energy Efficient Building , 115-25.

Kolokotroni M, Tassou S, Gowreesunker B L. (2014). Energy aspects and ventilation of food retail buildings. Advances in Building Energy Research, 9, 1-19.

Kolokotroni M, Webb B C , Hayes S D., (1998). Summer cooling with night ventilation for office buildings in moderate climates. Energy and buildings, Elsevier,27, 231-237.

Kolokotroni M. (1998, March 10). Night Ventilation for cooling: Field Tests and design Tools. Low-Energy Cooling Technologies for Buildings: Challenges and Opportunities for the Environmental Control of Buildings, Professional Engineering Publishing, London,UK,33-44.

Li-Xia Wu, Zhao J N, Wang Z J. (2006). Night ventilation and active cooling coupled operation for large supermarkets in cold climates. Energy and Buildings, Elsevier,38,1409-1416.

Mavromatidis G, Acha S, Shah N (2013). Diagnostic tools of energy performance for supermarkets using Artificial Neural Network algorithms. Energy and Buildings, Elsevier,62, 304-314.

Measurement Systems Ltd. Retrieved from https://www.measurementsystems.co.uk/data-logging/miniature_temperature_loggers/ds1922l__thermochron_data_logger_-40_to_85_c
Part L. (2014). Building Regulations Part L2A Conservation on Fuel and Power in new buildings other than dwellings.

Pearson A. (2014). CIBSE Case STudy: Advanced Energy Efficient Supermarket. CIBSE Journal.

Stovall T K, Baxter V D (2010). Modeling Supermarket Refrigeration with EnergyPlus. Retrieved from Oak Ridge National Laboratory:

Sullivan R, Gouldson A. (2013). Ten years of corporate action on climate change: What do we have to show for it? Energy Policy, Elsevier,60, 733-740.

Tassou S, Ge Y, Hadawey A, Marriott D. (2011). Energy consumption and conservation in food retailing. Applied Thermal Engineering, Elsevier,31, 147-156.

Tempcon Intrumentation Ltd. HOBO Temp/RH 3.5% data logger UX100-003, HOBO U12 Temperature/Relative Humidity/Light/External Data Logger - U12-012.

U.S. Department of Energy, Energy Efficiency & Renewable Energy. EnergyPlus Energy Simulation Software. Retrieved from http://apps1.eere.energy.gov/buildings/energyplus/

EFFECT OF BUILDING AND INSTALLATION DESIGN ON $PM_{2.5}$

Piet Jacobs*, Wouter Borsboom[1]

1 TNO
Leeghwaterstraat 44
2628 CA Delft

**Corresponding author: piet.jacobs@tno.nl*

ABSTRACT

People spend more than 80% of their time indoors. In contrast to ambient air, no (legal) limits for indoor particulate matter exist, although there are WHO guidelines. In the Netherlands a measurement protocol to determine the $PM_{2.5}$ in office buildings has been developed including 5 quality classes. However at the moment no simple guidelines or models are available which can support the design and in-use phases to predict the $PM_{2.5}$ concentration in office buildings and schools. This paper describes the results of a mass balance model which predicts the effect of building and installation design parameters on $PM_{2.5}$ concentration. Three experimental case studies are used to fit two parameters in the model and to validate the overall model. For Dutch buildings air tightness, ventilation flow and the filter quality prove to be critical factors. In the US, due to high recirculation rates, increasing the ventilation flow or reducing the infiltration flow has only a minor effect. There it is more effective to increase the filter quality. The results of our model further indicate that it is well possible for modern offices equipped with F7 filters to be below the WHO air quality guideline for annual mean $PM_{2.5}$ of less than 10 μg/m^3. It would be interesting to further validate the model with regard to deposition losses in offices and to include PM_{10} in the model for schools.

KEYWORDS

$PM_{2,5,}$ Office building, School, Air tightness, filter quality

1 INTRODUCTION

People spend more than 80% of their time indoors. However, unlike ambient air, no legal limits exist for particulate matter inside. Because there is no convincing evidence that the hazardous nature of particulate matter differs from indoor sources as compared to outdoor sources the steering group assisting in WHO indoor air quality guidelines concluded that the

2005 air quality guidelines for particulate matter are also applicable to indoor spaces (WHO 2010). In the Netherlands a measurement protocol to determine the $PM_{2.5}$ in office buildings has been developed [VLA 2014]. However at the moment no simple guidelines are available which can be referred to in the design and the use phase to predict the $PM_{2.5}$ in an office building. In EN 13779 (2007) different Outdoor Air (ODA) and Indoor Air (IDA) classes have been defined to facilitate filter selection for buildings. Firstly these classes are difficult to interpret as they are relative to the outdoor air quality. Secondly the effect of infiltration, by which untreated air enters the building, is not incorporated. Field studies (Jacobs, 2014) show that open windows, open doors and infiltration through cracks can have a major influence on the indoor $PM_{2.5}$ concentration. Thirdly the EN 13779 does not take into account internal sources of $PM_{2.5}$. These may have a significant effect on the indoor concentration at schools as reported by Blondeau et al. (2004) and Alves et al. (2014). Therefore this paper describes the results of a model to predict the effect of three building and installation design parameters (ventilation flow, airtightness and filter quality) on $PM_{2.5}$ concentrations in office buildings and schools.

2 METHODS

As a marker for the indoor fine dust, $PM_{2.5}$ has been chosen. The measurement protocol for $PM_{2.5}$ is published in Dutch (VLA 2014) and in English [Jacobs 2015]. In short the measurement protocol consists of the following steps. The indoor $PM_{2.5}$ concentration at the work place, see figure 1, is measured by a calibrated optical particle counter during a working week. The indoor concentration largely depends on the outdoor concentration and is corrected for low or high concentrations during the measurement week. The hourly outdoor concentration is derived from an outdoor station of the Dutch air quality monitoring network. Based on the one week measurement data an estimation of the yearly averaged $PM_{2.5}$ concentration is made. This estimate is then compared to the WHO (2006) annual advisory value of 10 $\mu g/m^3$ according to the VLA methodology class A , 15 $\mu g/m^3$ corresponding to class B and the Dutch limit value of 25 $\mu g/m^3$, in force since1-1- 2015, corresponding to class C. Buildings with $PM_{2.5}$ concentrations above 25 $\mu g/m^3$ are classified as D. To stimulate good building and installation designs an A+ class has also been defined: < 2.5 $\mu g/m^3$.

Figure 1: optical particle counter placed in suitcase in an office room.

A mass balance model has been established of a typical office building and a class room. The model contains three input parameters:

- Air exchange rate: the number of air changes per hour;
- Specific airtightness, this is the air leakage at 10 Pa pressure difference per m^2 floor area;
- Filter quality according to EN779:2012.

The indoor $PM_{2.5}$ is calculated by equation 1 similar to Chan (2015).

$$PM_{2.5} = \frac{PM_{2.5\ \text{ambient}}\,[I+V(1-\eta)]+s}{I+V+d} \qquad (1)$$

$PM_{2.5}$ and $PM_{2.5\ \text{ambient}}$ are the indoor and outdoor $PM_{2.5}$ concentrations ($\mu g/m^3$), I and V are the infiltration and ventilation air exchange rates (1/h).

The filter efficiencies η with regard to $PM_{2,5}$ have been estimated in consultation with filter suppliers and the results of the field studies, see table 1.

Table 1: assumed filter efficiencies.

Filter quality according to EN779	Estimated $PM_{2.5}$ removal fraction η
G4	0.50
M5	0.60
M6	0.90
F7	0.95
F9	0.99

In the model the effect of desorption on surfaces has been taken into account by a first order deposition loss d = 0.15 1/h as found by Zaatari (2014) for retail stores. The effect of internal sources has been taken into account. The internal source s ($\mu g/(h\ m^3)$) is based on the intercept b of the XY plots of the results of two office building case studies (Jacobs 2015). By assuming $PM_{2.5\ \text{ambient}}$ is zero, from equation (1) it follows:

$$S = (I + V + d) * PM_{2.5} = (I + V + d) * b \qquad (2)$$

The intercept for offices is typically 1,5 $\mu g/m^3$, the internal source S is then about 5 $\mu g/(h\ m^3)$. The internal source in class rooms has been estimated 4 times higher than offices due to the higher occupation.

To determine the infiltration rate a yearly average pressure difference of 3 Pa with the outside is assumed. Based on this average pressure difference and the airtightness levels used in the model the infiltration rate ranges from 0.1 to 0.9 air changes per hour. For reference, the Dutch Building decree specifies a minimum air tightness of 1.2 $dm^3/(s\ m^2)$ at 10 Pa pressure difference. This corresponds with an infiltration rate of about 0.7 air changes per hour.

3 RESULTS

3.1 Model study

The results of the mass balance model with regard to $PM_{2,5}$ in an office building and a class room are given in table 2 and 3. In the tables the $PM_{2,5}$ concentration is expressed according to the classification described under the method section. Offices equipped with F7 filters are well below the WHO air quality guideline for annual mean $PM_{2.5}$ of less than 10 µg/m^3 (class A). In schools, due to the higher internal sources, sufficient air flow is required to achieve class A.

Table 2: offices, indicative effect of filter efficiency, air tightness and ventilation rate on $PM_{2.5}$ in offices according to VLA classes

Air Exchange rate [ACH]	Airtightness [dm^3/(s m^2)] @10Pa	without filter	G4	M5	M6	F7	F9
< 1	> 1,2	B	B	B	B	B	B
1 – 2	> 1,2	B	B	B	A	A	A
2 – 3	< 1,2	B	B	A	A	A	A
> 3	< 0,8	B	A	A	A	A	A
> 3	< 0,4	B	A	A	A	A	A+
> 4	< 0,1	B	A	A	A	A+	A+

Table 3: indicative effect of filter efficiency, air tightness and ventilation rate on $PM_{2.5}$ in classrooms according to VLA classes

Air Exchange rate [ACH]	Airtightness [dm^3/(s m^2)] @10Pa	without filter	G4	M5	M6	F7	F9
1	> 1,2	C	C	C	B	B	B
2	> 1,2	C	B	B	B	B	B
3	< 1,2	C	B	B	A	A	A
4	< 0,8	C	B	B	A	A	A
5	< 0,4	C	B	A	A	A	A
6	< 0,4	C	B	A	A	A	A

3.2 Comparison model with case studies

Three case studies have been carried out, these have been described in detail by Jacobs (2015). The results are summarized in table 4 and compared with the model results. Note that the case study results are used for the prediction of the internal source S and the filter efficiency η. Two of the three classes have been predicted well. With regard to the third building class A is predicted while the measurement indicates a class A+ (<2.5 µg/m^3). The lay-out of the last building acts as a guard ring which ensures a very low infiltration of ambient air. With only an M6 filter the lowest $PM_{2.5}$ concentration is achieved. These results clearly show that not only the filter quality but also the infiltration flow relative to the ventilation flow is an important parameter.

Table 4: comparison case studies with model prediction $PM_{2.5}$ [μg/m³]

Case study	**Measured (μg/m³) / VLA class**	**Model (μg/m³) / VLA class**
1. Office building Rotterdam (F7)	4.4 / class A	3.5 / class A
2. TNO Office building (F7)	5,8 / class A	5.3 / class A
3. Office with guard ring (M6)	1 / class A+	2,6 / class A

3.3 Comparison with literature

Chan et al (2015) has performed a similar model study for the United States. The predicted $PM_{2.5}$ concentrations of Chan and by this study are listed in table 5. Especially in the situation without filters, large deviations between the two models can be seen. With regard to buildings equipped with F7/MERV 13 filters the models are quite close. Explanations are:

- Ambient air quality: for the US an average outdoor $PM_{2.5}$ of 10.8 and for the Netherlands 15 μg/m³ is assumed;
- The filter efficiency used by Chan is much lower than assumed in this paper;
- The air infiltration rate used by Chan for schools and offices is 0.2 and 0.1 /h respectively which is much lower as assumed in this paper.
- Chan assumes a relative low ventilation rate of 0.5 – 1.0 and a recirculation rate of 3 /h while in the Netherlands the ventilation rates are in the order of 1 – 4 /h and no recirculation is applied.

Table 5: predicted mean indoor concentration $PM_{2.5}$ [μg/m³]

	Chan (2015)	TNO model
Office without filter	6.6	15.8
Office G4/MERV 8	5.9	10.3
Office F7/MERV 13	3.5	5.3
School without filter	8.3	19.3
School G4/MERV 8	8.3	13.3
School F7/MERV 13	5.2	8.0

It can thus be concluded that there exists large differences in air handling between the US and the Netherlands. This also has an effect on the strategy for $PM_{2,5}$ reduction. In the US, due to the high recirculation rates used, it is most effective to increase the filter quality. Increasing the ventilation flow or reducing the infiltration flow has only a minor effect. In the Netherlands, recirculation is not applied, therefore a higher ventilation flow and a better air tightness have a relatively large effect.

3.4 Future research

The estimate for the deposition d is derived from research in retail stores. This loss may be too low for offices. A better estimate may be derived from research in naturally ventilated offices where no filters are applied. Further, more research in schools is required to validate the model calculations with regard to schools and to also include PM_{10}.

4 CONCLUSIONS

A mass balance model has been set up to calculate the $PM_{2.5}$ concentration in office buildings and schools. The model provides a straightforward method to classify buildings with regard to the indoor $PM_{2.5}$ concentration and enables estimation of the effect of measures to improve the air quality. For the Netherlands air tightness in combination with the ventilation flow and the filter quality are critical factors. In the US, due to the use of recirculation, increasing the ventilation flow or reducing the infiltration flow has only a minor effect. More research in naturally ventilated offices could improve the estimate for the deposition rate. Also more research in schools is required to include PM_{10} in the model.

5 REFERENCES

Alves C. et al (2014) Indoor/Outdoor relationships between PM10 and associated organic compounds in a primary school, *Aerosol and Air Quality Research*, 14: 86 – 98.

Blondeau P., Lordache V., Poupard O., Genin D, Allard F. (2004) Relationship between outdoor and indoor air quality in eight French schools, *Indoor Air*, 15: 2 – 12.

Chan W.R.et al. (2015) Estimated effect of ventilation and filtration on chronic health risks in U.S. offices, schools, and retail stores, *Indoor Air*, 2015.

EN 13779 (2007) Ventilation for non-residential buildings – Performance requirements for ventilation and room conditioning

Jacobs P, Phaff H., Voogt M. (2014) Energy efficient reduction of fine and ultra-fine dust in a nursery, *Proceedings Indoor Air conference Hong Kong*.

Jacobs P., Voogt M. (2015), $PM_{2,5}$ measurement protocol for offices, *Proceedings of Healthy Buildings Europe congress 2015*.

VLA (2014): Meetprotocol fijn stof bepaling in kantoren (in Dutch: http://binnenmilieuadvies.info/fijnstof-in-binnenlucht/)

WHO (2006) Air quality guidelines global update 2005, ISBN 92 890 2192 6, (http://www.euro.who.int/__data/assets/pdf_file/0005/78638/E90038.pdf)

WHO (2010) WHO guidelines for indoor air quality: selected pollutants, ISBN 978 92 890 0213 4 (http://www.euro.who.int/__data/assets/pdf_file/0009/128169/e94535.pdf)

Zaatari M. et al. (2014) Particle characterisation in retail environments: concentrations, sources, and removal mechanisms, *Indoor Air*, 24, 350-361.

EXPERIMENTAL CHARACTERIZATION OF THE EFFICIENCY AND ENERGY CONSUMPTION OF VARIOUS CENTRAL VENTILATION AIR CLEANING SYSTEMS

Patrice Blondeau[1], Marc Olivier Abadie[1], Alexandra Durand[2], Carole Deléglise[2], Pascal Kaluzny[2], Sylvie Parat[3], Alain Ginestet[4], Dominique Pugnet[4], Céline Tourreilles[5] and Thierry Duforestel[5]

1 LaSIE / University of La Rochelle,
Avenue Michel Crépeau
17042 La Rochelle cédex 1, France
Corresponding author : patrice.blondeau@univ-lr.fr

2 Tera Environnement
628 rue Charles de Gaulle,
38920 Crolles, France

3 Air & Bio
13 rue Jules Ferry
73000 Chambéry, France

4 CETIAT
25 avenue des arts
BP 52042, 69603 Villeurbanne, France

5 EDF R&D
Avenue des Renardières
77818 Moret-sur-Loing cédex, France

ABSTRACT

The present study aimed at assessing six commercially-available in-duct air cleaning devices which are designed to be mounted in the central ventilation system of offices or commercial buildings. The selected devices use different air cleaning technologies: mechanical filtration, electrostatic precipitation, gas filtration, ionisation / cold plasma, photocatalytic oxidation (PCO) and catalysis under UV light. They were tested against particles, a mixture of volatile organic compounds containing acetone, acetaldehyde, toluene, heptane and formaldehyde, and two bio-contaminants: *Aspergillus brasiliensis* (fungus) and *Staphylococcus epidermidis* (bacteria).

Two different test rigs were used for the tests. The single pass efficiency of each device was determined for 3 airflow rates, ranging from 1200 m^3/h to 3600 m^3/h, and two sets of temperature and humidity that are representative of wintertime and summertime indoor air conditions. The concentrations of the challenge VOC were also varied in the range from 30 to 100 $\mu g/m^3$ as a way to characterize their influence upon efficiency at realistic concentration levels for non-industrial buildings. Ozone and formaldehyde concentration measurements downstream of the air cleaners were achieved to determine the rate of harmful by-products that are released. Finally, the energy issue was addressed by measuring the electric consumption (if any) and pressure loss of the devices.

The results show that single pass efficiencies can vary in a wide range from one system to another. For a same device, it can also vary a lot from one challenge contaminant to another, which is somewhat a more intuitive conclusion. Two systems have no efficiency at all, or negligible impact on the concentrations of the challenge pollutants. The air handling unit containing a F8 class mechanical filter, a PCO reactor and a gas filter proves to be quite efficient in removing pollutants. However, photocatalytic oxidation isn't effective while tremendously adding to the energy consumption. Finally, two devices show from moderate to high efficiency for a wide range of contaminants and acceptable energy consumption. In a general way, the study provides a methodology to assess the benefits of using central ventilation in-duct air cleaners, and then to determine which system is most suited for a building, based on indoor air quality, cost and energy criteria.

KEYWORDS

Air cleaning, in-duct, efficiency, energy, by-products

1 INTRODUCTION

During the past decade, various indoor air cleaning solutions have come to the market, including stand-alone devices, photoactive or adsorbent materials, and in-duct devices. In-duct systems put into play different air cleaning technologies. They are intended to be mounted on the recycled air as a way to promote efficiency by multiple passes of the indoor air through the device. So far, this type of systems has mainly been assessed in the context of house applications, and tested against airflow rates of few hundreds of cubic meter per hour. Experimental studies dealt with the efficiency of either commercially-available devices (Sidheswaran et al, 2012) or prototypes (Hodgson et al, 2007; Destaillats et al, 2012). Others aimed at characterizing the possible adverse health effects resulting from the emission of ozone or by-products (Gunther et al, 2011; Saughnessy et al, 2014). Very few researches nevertheless investigated the relevance of in-duct air cleaning in the context of office or commercial buildings where air handling units are designed to circulate and condition large airflow rates. Moreover, all available studies focussed on the capacity of adsorbent filters to remove volatile organic compounds (VOC) or ozone (Haghighat et al, 2008; Bastani et al, 2010; Stanley et al, 2011), while many other technologies are now available on the market. The present study aimed at filling the gap in the knowledge. Six commercially available in-duct air cleaners having a nominal airflow rate about 3600 m^3/h were fully assessed from laboratory experiments. For each device, the single pass efficiency was measured for a wide range of air contaminants in realistic conditions of operation. Ozone and formaldehyde production rates were also measured, when relevant. Finally, the energy consumption was evaluated by measuring both the device energy and pressure drop.
After presenting the main features of the selected devices, the paper describes the two tests rigs and associated protocols that were used to carry out the tests. Then, the devices are compared, and the potential of in-duct air cleaning for office buildings is discussed based on single-pass efficiency, yield of ozone and by-products, and energy cost.

2 MATERIAL AND METHODS

2.1 Selected devices

Six in-duct air cleaning devices that are commercially available in Europe were selected to undergo the tests. All of them have a nominal airflow rate around 3600 m^3/h, but use different air cleaning processes (Figure 1).

S1 is an air filtering solution which is made of a particle filter and an activated-carbon gas filter. Therefore, there’s no device energy in this case. The particle filter has a V shape and is of class F7 according to EN 13779. It is sold as a low energy consumption filter. The adsorbent (activated-carbon) filter also has a V-shape. It was designed to adsorb low-weight volatile organic compounds (VOC); therefore it is particularly well suited for indoor air applications. The two filters were tested one by one against airborne particles but as an assembly against gases and bio-contaminants.

S3 is a stand-alone and ready to use air handling unit (AHU) having external dimensions of 1.95 x 0.71 x 0.85 m. The internal cross-sectional area is 0.61 x 0.61 m. The AHU has a fan that can circulate airflow rates up to 5600 m^3/h, but it was turned off during all tests. The device was supposed to contain a F8 class filter (EN 13779), a photocatalytic oxidation (PCO) reactor, and an activated-carbon gas filter, which corresponds to the manufacturer's recommendation for office building applications. However, only the PCO reactor was tested. The efficiency and energy parameters of the whole device was computed afterwards from typical fractional efficiency and pressure loss data for F8 class filters, efficiency and pressure loss of the activated carbon filter, which was assumed to be the one implemented in S1, and finally measured efficiency and pressure loss of the PCO reactor. The PCO reactor is made of a series of 3 flat TiO_2-coated filters and 2 series of 8 UVC lamps having a unit power of 95 W between them. The tests were repeated for two geometries of the filters, honeycomb and knitted metal.

S4 is an air cleaning device which is to be inserted into a duct or an AHU. It utilizes the so-called radio catalytic ionization (RCI) to generate a plasma from a 'UVX' lamp that irradiates honeycomb cells. The latter are made of a rhodium, titanium, silver and copper alloy. The manufacturer claims destruction of airborne and deposit fungi and bacteria, as well as removal of airborne particles and VOC due to the production of ions, free radicals, and subsequent oxidation processes.

S5 is an electrostatic precipitator (also called electronic particle filter) and S6 a plasma ionizer which are distributed by the same company. Both have standard filter dimensions of 0.592 x 0.592 m. The two devices were tested separately but also as an assembly (S2 = S5 + S6) since the distributor claims higher efficiency of the plasma ionizer when the electrostatic precipitator operates upstream. The explanation is that the electrostatic precipitator releases ozone which promotes ion generation.

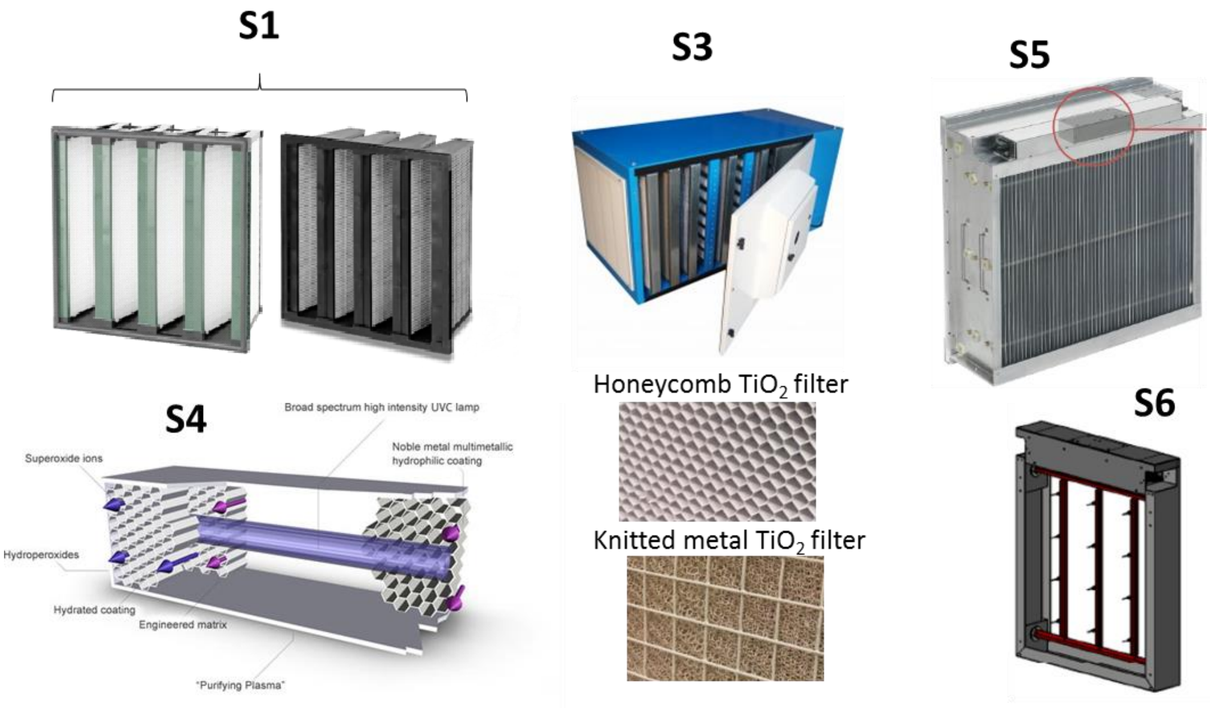


Figure 1: Air cleaning devices tested (NB: S_2 is the combination of S_5 and S_6)

2.2 Challenge contaminants and energy parameters

The selected systems were first assessed based on their efficiency against three groups of indoor air contaminants:

- VOCs, including acetaldehyde (C_2H_4O), acetone (C_3H_6O), heptane (C_7H_{16}), toluene (C_7H_8), and formaldehyde (CH_2O). These are the challenge contaminants that were selected to determine the efficiency of photocatalytic devices in the frame of the European standard project prEN 16846-1;

- Airborne particles. The fractional efficiency of the air cleaners was determined using either latex or DEHS particles. Latex particles exhibit a wider range of particle sizes (0.2 – 5.0 μm) than DEHS particles;

- Bio-contaminants. *Staphylococcus epidermidis* and *Aspergillus brasiliensis* (previously called *Aspergillus niger*) are representative of indoor air bio-contaminants. *Staphylococcus epidermidis* is a Gram-positive bacterium. It is part of the normal human flora, typically the skin flora, and less commonly the mucosal flora. *Staphylococcus epidermidis* has a spherical shape (cocci) of about 1 μm diameter. *Aspergillus brasiliensis* is a fungus and one of the most common species of the genus *Aspergillus*. Its spores have a spherical shape of diameter lying in the range from 3.5 to 5.0 μm.

Possible adverse health effects resulting from the yield of ozone when the devices use either UV lamps or electric fields (S2, S4, S5, S6), and/or production of by-products when the air cleaning process is based on the principle of VOC mineralization (S2, S3, S4, S5, S6), were assessed by measuring ozone and formladehyde concentrations upstream and downstream of the tested devices (C_{up} and C_{down} in μg/m^3, respectively). Then, this data was converted into ozone or formaldehyde production rate, τ (μg/h), from the equation:

$$\tau = Q\left(C_{up} - C_{down}\right), \qquad (1)$$

Where Q (m^3/h) is the air flow rate.

Finally, the energy consumption related to the implementation of the air cleaning devices in the HVAC system of a building was assessed by measuring the electric power and pressure loss.

2.3 Test conditions

When mounted on the recycled air of the HVAC system of buildings, air cleaning devices operate under time varying contaminant concentrations, air flow rate, temperature and humidity. Therefore, for a full characterization of the selected devices, these parameters were varied in a range which corresponds to realistic values for office-like buildings. The maximum concentration to be tested was set to 100 μg/m^3 for each single VOC. Then the tests were repeated for set point concentrations of 50 and 20 μg/m^3 (concentration was assumed to have negligible influence upon efficiency for airborne particles and bio-contaminants). Similarly, the devices were tested at a maximum and minimum airflow rate of 3600 m^3/h and 1200 m^3/h (particles) or 1600 m^3/h (gas and bio-contaminants), respectively. When the measured efficiencies were significantly different, the tests were repeated for an intermediate airflow rate of 2400 m^3/h for particle measurements, and 2600 m^3/h for gas and bio-contaminant measurements. Finally, two sets of temperature and relative humidity were considered: 19°C / 45% RH and 24°C / 70% RH. These are commonly used set points for wintertime and summertime, respectively, in the air conditioned buildings of European countries.

2.4 Test rigs and measurement methods

Two different test rigs were used to carry out the tests: rig 1 was used for particle efficiency and pressure loss measurements, while rig 2 was used for gas, bio-contaminants and device energy measurements.

Test rig 1 is made of horizontal square ducts with a cross sectional area of 0.61 m x 0.61 m. Any filter having standard dimensions can be inserted in the open-loop and tested according to the EN 13779:2012 standard. Starting from the air inlet, the test apparatus includes a diaphragm to monitor the circulating airflow rate, a heating coil to prevent from excessive moisture content of the air in wintertime, and a speed-regulated fan. A high efficiency filter (HEPA type) is mounted downstream of the fan to remove as many particles as possible from the air. The pressure loss of filters, or any other air cleaning device, is measured from four pipes which are connected to each of the four sides of the duct upstream and downstream of the device on the one hand, and to a differential pressure sensor (Rosemount) on the other hand.

Particles were generated and injected upstream of the tested device using a Collison-type (latex) or a Laskin-type (DEHS) generator. Air samples were then collected in an isokinetic way upstream and downstream of the device, and analysed by an optical particle counter (Lasair PMS 210). The fractional efficiency was determined as the average of 13 alternate particle counts upstream and downstream of the device. Each count lasted for one minute. The sampling line was purged with clean air for one minute each time the system was changing sampling port.

Test rig 2 was designed to test different sizes and types of air cleaning devices in a once-through (open-loop) mode (Figure 2). It is made of insulated galvanized steel round or rectangular ducts. Airflow rates ranging from 1200 to 3600 m^3/h can be circulated through the apparatus by a centrifugal fan having a speed control which is mounted prior to exhausting. The outdoor air is introduced to the system through an inlet damper and then passes through an AHU containing a G4 filter and a F7 filter, heating coils, a humidifier, a cooling coil and finally a high efficiency particle filter (H10). The components were designed to achieve setup air temperature between 19°C and 25°C, and relative humidity between 45% and 75%, whatever the outdoor air conditions. These parameters were monitored upstream of the tested device using a unique probe (HOBO U14 LCD data Logger, point 4 in Figure 2a). The airflow rate was also monitored using a hot-wire air speed probe (960 Probe, TSI, point 3 in Figure 2a).

The test apparatus had two parallel lines: Line 1 is made of square ducts and was designed to test devices with nominal cross sectional area of 0.61 m x 0. 61 m; the devices are mounted inside an AHU which is part of the rig (see between points 5 and 6 in Figure 2a). Line 2 allows to test air cleaning devices of any size and shape by connecting flexible ducts to the device on one side, and the test rig on the other side. This line was used to test device S3 (Figure 2b). Low leakage dampers were mounted in each end of the two lines to avoid air leakage through the unused line.

The challenge contaminants were injected far enough upstream of the air cleaning device, and upstream the duct bend (point 2 in Figure 2), as a way to promote mixing of the contaminants with the air. VOCs were injected from two programmable syringe pumps (Harvard apparatus 70-4500): one contained toluene, heptane, acetone and acetaldehyde while the other one contained formaldehyde as an water solution. *A. brasiliensis* and *S. epidermidis* were generated from a liquid suspension using a Collison CN25 particle generator and a stainless steel pipe going through the duct wall. This pipe was bended inside the duct so that the contaminants are injected in the direction of the airflow. The suspensions were calibrated to concentrations of 10^6 ufc/ml of *A. brasiliensis*, and 5.10^6 ufc/ml of *S. epidermidis*. The generator was cleaned with water and a biocide solution after each test, and the pipe was disinfected using alcohol 70%.

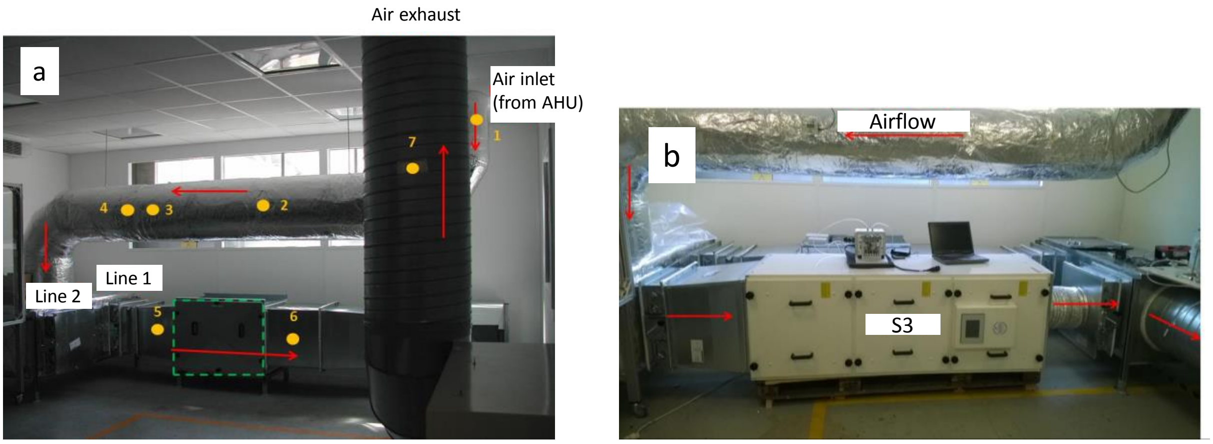


Figure 2: General view of test rig 2, used for gas and bio-contaminant measurements

The position of sampling ports upstream and downstream of the air cleaning device is represented by points 5 and 6 in Figure 2. Preliminary tests showed that challenge contaminant concentrations were uniform throughout the whole cross section of the duct in the upstream and downstream zones of the tested device, thus allowing a single point sampling procedure from each zone.

Bio-contaminant concentrations were determined using a single-stage Andersen impactor. Isokinetic air sampling was achieved at a rate of 28.3 l/min for 4 to 8 minutes. Samples upstream and downstream of the tested device were taken at the same time, each with 3 replicates. Trypticase soy broth incubated at 37°C for 48 h, and malt agar incubated at 27°C for 72 h were used as culture media for *S. epidermidis* and *A. brasiliensis*, respectively. The count uncertainty was estimated to be of ±10% for *S. epidermidis* and ±15% for *A. brasiliensis*. It can be noted that for all tests, sample blanks were achieved by measuring the bacteria and fungus concentrations upstream and downstream of the device before they were generated. Additional measurements also included situations when the device is turned off. Finally, S3 was tested with only the UVC lamps operating (TiO_2 filters were removed from the AHU), and then with the filters but the UVC lights turned off, as a way to isolate the effects of mechanical filtration and direct germicidal irradiation from the global efficiency of the PCO reactor.

Toluene, heptane, acetone and acetaldehyde concentrations were measured online using a Ion Molecule Reaction Mass Spectrometry (IMR-MS) analyzer (Airsense, V and F). The limit of quantification for acetaldehyde, acetone, heptane and toluene are 13, 17, 25 and 25 µg/m^3, respectively. Formaldehyde concentrations were measured according to ASTM Method D5197 which consists of trapping of the analytes on a silica gel adsorbent coated with 2,4-dinitrophenylhydrazine (DNPH), followed by analysis by HPLC/UV. The limit of quantification is 5 µg/m^3. Finally, ozone concentrations were measured downstream of the tested devices using an online analyzer (03 41M, Environnement SA). Continuous measurements were made over a period of 10 to 30 minutes, with a time resolution of 1 measurement per minute. The ozone concentration that was measured with the device turned off was taken as the concentration upstream of the device to compute the ozone production rate from Eq. 1.

3 RESULTS AND DISCUSSION

3.1 Efficiency

Figure 3 provides an overview of the efficiency of the selected air cleaning devices. S2, S4 and S6 are not displayed because all efficiencies were close to zero for S4 and S6, and any significant increase in efficiency could be observed when comparing S2 with S5 (which means that there’s actually very few influence of ozone generated by S5 upon efficiency of S6). These devices are no longer considered hereafter.

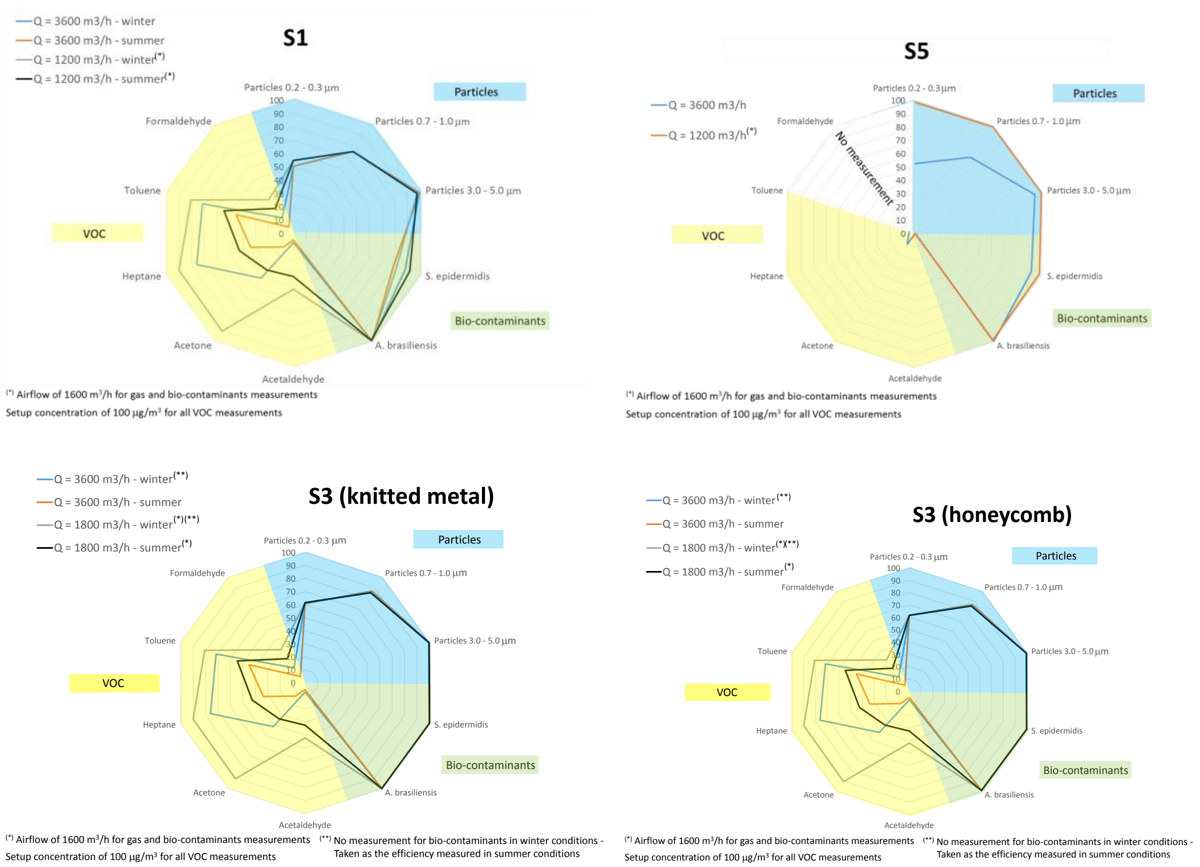


Figure 3: Overview of measured efficiencies for devices S1, S3 and S5 – Summer = 24°C/70% RH, Winter = 19°C/25% RH

At first, it can be noted from Fig. 3 that the electrostatic filter S5 exhibits high efficiencies for all kind of particles, including bacteria and fungus, but it is inefficient against VOC, which was expected. The measured efficiency is close to 100% for all particle sizes at the lowest air flow rate tested (1200 m^3/h), but it falls to 55% for the smallest particles when the air flow rate is increased to 3600 m^3/h. This observation is consistent with the principles of electrostatic precipitation. S1 and S3 are a bit less efficient in removing particles and bio-contaminants from the air than S5, especially when the air flow rate is high. On the other hand, Fig. 3 shows that these devices have from moderate to pretty high efficiencies for VOCs depending on species and airflow rate. The fact that S1 and S3 efficiency profiles are very similar whatever the type of photocatalytic filter implemented demonstrates that the PCO reactor doesn't really add to the air cleaning performance of S3. Actually, the tests which were performed with this reactor showed 100% efficiency for *S. epidermidis*, and a very low efficiency, or no efficiency at all, for all other challenge contaminants. By comparing the results when the lights were turned on and off, and when TiO_2 filters were inside or taken out of the AHU, it could also be clearly demonstrated that bacteria were killed as a result of the germicidal effect of UVC lamps rather than photocatalytic oxidation processes. In the end, the particle and bio-contaminant efficiencies of device S3 are mainly contributed by the mechanical filter (F8 class) while the VOC efficiency is only contributed by the adsorbent filter.

The strong influence of air flow rate (air speed) upon VOC efficiency of S1 is consistent with the fundamentals of adsorption dynamics. The results are also consistent with theory regarding 1) the observed correlation between boiling point of VOCs and sorption efficiency (Figures 3 and 4): for a same airflow rate, the gas filter efficiency is the highest for toluene (T_b = 110.6 °C) and then heptane (T_b = 98.4 °C), acetone (T_b = 56.1 °C), acetaldehyde (T_b = 20.2 °C) and formaldehyde (T_b = -19.3 °C). Such correlation had previously been emphasized by Haghighat and al (2008), Bastani and al (2010) and Popescu and al (2013); 2) the influence of temperature and relative humidity upon sorption efficiency: for a same airflow rate and a same VOC concentration upstream of the filter, the measured efficiency is significantly higher at a lower temperature and a lower relative humidity. Only formaldehyde shows similar efficiency at 24°C/70% RH and 19°C/45% RH, which can be explained by a very high solubility in water. In this case, higher amounts of condensed water in the pores of the activated carbon when relative humidity is increased contribute to promote formaldehyde absorption. It can balance and even overtake the decrease of formaldehyde adsorption at the pore surfaces due to higher temperature and competition with water vapour; 3) the influence of concentration: except in the case of toluene, for which the efficiency is close to 100% whatever the concentration in the conditions presented on Figure 4, higher efficiencies are observed at higher concentrations upstream of the filter. The increase in efficiency is most particularly visible when the setup concentration is increased from 33 to 66 μg/m^3.

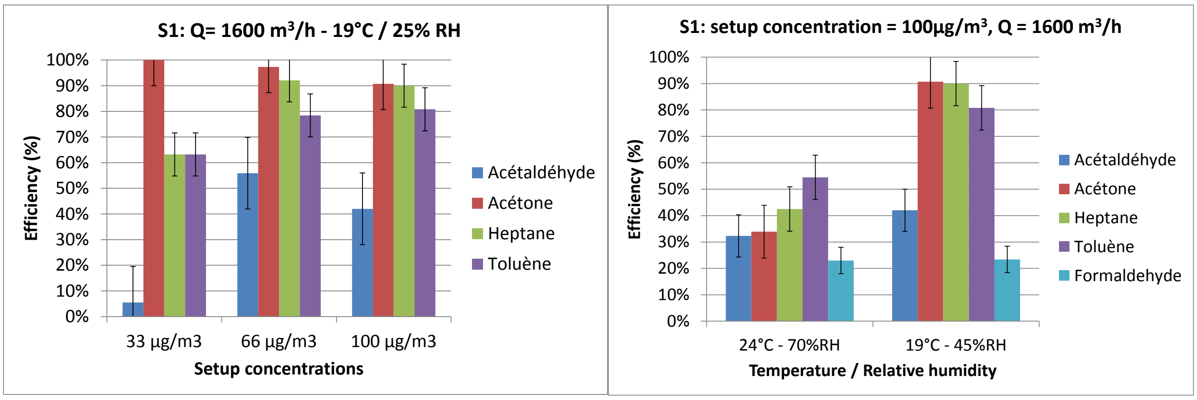


Figure 4: Efficiency of S1 as a function of VOC, concentration and temperature / humidity air conditions

3.2 Production of ozone and by-products

No formaldehyde could be detected downstream of devices S2, S3, S4, S5 and S6 when they were tested against the mixture of toluene, heptane, acetone and hexaldehyde. This was expected since no significant removal efficiency, and therefore no significant oxidation of any of the challenge VOCs, could be observed. S3 and S4, which use UV lamps, were found to emit no ozone in the circulating air. On the other hand, measurements of ozone concentrations downstream of S5 and S2 led to surprising results. Figure 5 shows that the ozone emission rate is both influenced by the air flow rate and the hygrothermal conditions of the air upstream of the device. In indoor air summertime conditions, the ozone production rate at 3600 m^3/h is twice the one measured at 1600 m^3/h. No linearity with airflow rate can be observed from Fig. 4 but the production rates at 2400 m^3/h are most probably underestimated due to technical problems that led to achieve a relative humidity of the air which was far below the set point (41% vs 70%). Relative humidity obviously has a strong influence upon ozone production since for the maximum airflow rate, the production rates of S5 and S2 in summertime air conditions are about 3 times higher than in wintertime air conditions.

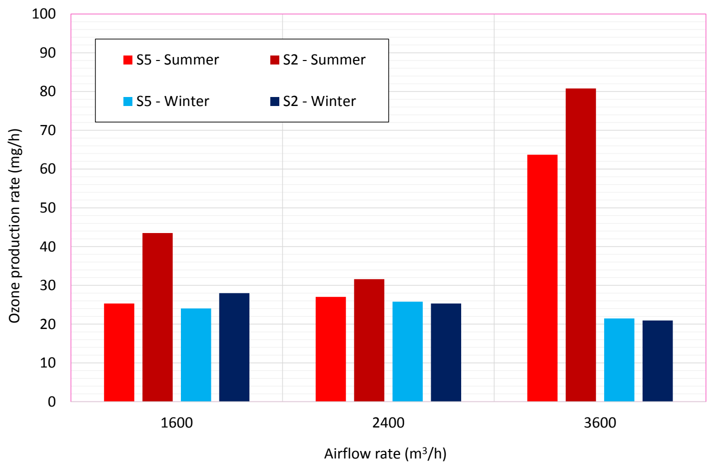


Figure 5: Ozone production rate of devices S5 and S2 - Summer = 24°C/70% RH, Winter = 19°C/25% RH

3.3 Energy consumption

The fan energy resulting from the implementation of any device in the HVAC system of a building, P_v (W), can be computed from the equation:

$$P_v = \frac{Q\Delta P}{\eta_v}, \qquad (2)$$

Where ΔP (Pa) is the pressure loss of the device, and η_v is the fan efficiency. The latter was here assumed to be 50% as a way to compare P_v with the device energy, P_a (W), on one hand, and to compare the selected air cleaning devices based on their total energy consumption, on the other hand. The results presented in Table 1 show that for all systems except S3, the device energy is pretty low, and far less than the fan energy when the air flow rate is greater than 2400 m^3/h. For S3, the fan energy is also quite important at high air flow rates when knitted metal filters are used (3 filters are connected in series). The device energy is overall so high here that it can hardly fit the standards of low energy buildings.

Table 1: Device energy and fan energy of the selected air cleaning devices

Energy (W)	**S1**	**S2**	**S3** Honeyc.	**S3** M. stitch	**S4**	**S5**	**S6**
Device energy	0	24	1520	1520	20	15	9
Fan energy, $\eta_v = 0.5$							
Q = 1200 m^3/h	17	5	1	12	negligible	5	negligible
Q = 1800 m^3/h	49	15	5	40		15	
Q = 2400 m^3/h	142	46	13	131		46	
Q = 3600 m^3/h	310	104	29	272		104	

4 CONCLUSIONS

The results presented in this paper are a first insight into the potential of in-duct air cleaners to improve indoor air quality while preserving energy in office-like or commercial buildings. Six devices that are available on the market were tested against a set of challenge contaminants, in realistic operating conditions. In the end, only two of the six devices proved to be relevant. Two of them showed no efficiency, or a very low efficiency, for all challenge contaminants. One has significant efficiency for a wide range of contaminant but it was demonstrated from additional tests that the main cleaning process is partly inefficient in removing contaminants while tremendously adding to the energy consumption of the building. Finally, the association between the electrostatic precipitator and the plasma ionizer wasn't found to be relevant though it was recommended by the seller. The reader's attention is drawn to the fact that the selected devices utilize different air cleaning techniques, but in no way the conclusions of this study should be extrapolated to all devices that utilize one of the tested technologies. It's also worth mentioning that all devices were assessed as brand new devices. Consequently, the efficiency and pressure loss of devices that contain filters are the initial efficiency and pressure loss. Although considering the mean efficiency and pressure loss over a given period of time would probably be more suitable, the methodology presented here may serve as a basis to develop a new standard to assess in-duct air cleaners.

5 ACKNOWLEDGEMENTS

The authors would like to express their gratitude to the French National Agency for Energy Conservation and Environmental Protection (ADEME) for the financial support of the study.

6 REFERENCES

Bastani A., Lee C.S., Haghighat F., Flaherty C., Lakdawala N., 2010, Assessing the performance of air cleaning devices – A full-scale test method, *Building and Environment*, 45, 143-149

Destaillats H., Sleiman M., Sullivan D.P., Jacquiod C., Jean Sablayrolles J., Molins L. Key parameters influencing the performance of photocatalytic oxidation (PCO) air purification under realistic indoor conditions. Applied Catalysis B: Environmental, 128 (2012) : 159– 170

Gunther M., Rhodes J., Morrison G., Siegel J. (2011). A Laboratory Method For Measuring The Ozone Emission From In-duct Air Cleaners. *Proceedings of Indoor Air 2011, the 12th International Conference of Indoor Air Quality and Climate,* Austin, USA, June 5-10, 2011

Haghighat F., Lee C.S., Pant B., Bolourani G., Lakdawala N., Bastani A. (2008) Evaluation of various activated carbons for air cleaning – Towards design of immune and sustainable buildings, *Atmospheric Environment*, 42, 8176-8184

Hodgson A.T., Destaillats H., Sullivan D.P., Fisk W.J. (2007) Performance of ultraviolet photocatalytic oxidation for indoor air cleaning applications. *Indoor Air*, 17(4): 305-316

Popescu R.S.**,** Blondeau P., Jouandon E., Coste J.C., Fanlo J.L. Elemental modeling of adsorption filter efficiency for indoor air quality applications. *Building and Environment*, 66, 2013, 11-22

Shaughnessy R., Reisdorph D. and Glenn Morrison (2014) Field testing to estimate ozone emission rates of in-duct air cleaners in occupied homes. *Proceedings of Indoor air 2014, the 13th International Conference on Indoor Air Quality and Climate, Indoor Air 2014*, Hong-Kong, China, July 7-12, 2014

Sidheswaran M., Destaillats H., Sullivan D., Cohn S., Fisk W.W. (2012) Energy efficient indoor VOC air cleaning with activated carbon fiber (ACF) filters. *Building and Environment (47)* , 357-367.

Stanley W., Ligman B., Rahmathullah M., Nance N. (2011) Life Cycle Valuation of Gas-Phase Filters for Ozone Removal. *Proceedings of Indoor Air 2011, the 12th Int. Conference of Indoor Air Quality and Climate*, Austin, USA, June 5-10, 2011.

CO_2 AND VOLATILE ORGANIC COMPOUNDS AS INDICATORS OF IAQ

Andrzej Szczurek, Monika Maciejewska * and Tomasz Pietrucha

Faculty of Environmental Engineering
Wroclaw University of Technology
Wybrzeże Wyspiańskiego 27
50-370, Wroclaw, Poland
** monika.maciejewska@pwr.edu.pl*

ABSTRACT

The trend toward minimizing ventilation of houses in order to reduce energy consumption for heating and cooling leads to an increase in indoor air pollution. The deterioration of indoor air quality (IAQ) negatively affects human health, safety, productivity and comfort. In order to evaluate the scale of this influence IAQ assessment has to be performed. However, the IAQ itself is not well defined and a number of parameters are considered as its indicators. In this work we compared carbon dioxide and volatile organic compounds as indicators of indoor air quality. In order to examine the problem, time series of CO_2 concentrations were considered as the source of information about IAQ. The data were obtained from continuous measurements of CO_2 and total volatile organic compounds (TVOCs). The following analytical instruments were applied: the non-dispersive infrared (NDIR) sensor – for CO_2 measurements, the photoionization detector (PID) and semiconductor gas sensors – for TVOCs determination. The correlation and regression analysis were applied to examine the relationship between measured quantities in two time scales, namely one day and 30 minutes. They reflect different time scales of CO_2 and volatile organic compounds (VOCs) concentration variation. Based on the analysis, we concluded that CO_2 and TVOCs measurements conveyed different information about IAQ, as a function of time. The analytical method had strong influence on the information obtained. In particular, the discrepancy was observed when comparing NDIR and PID methods. Also techniques applied for VOCs measurements provided different information about these substances. The results of our work entitle to conclude that the total concentration of VOCs should be taken into account as the indicator of IAQ in addition to CO_2.

KEYWORDS

Indoor air quality, correlation, sensor, time series, carbon dioxide, VOCs

1 INTRODUCTION

The fundamental requirement imposed on a heat, ventilation and air conditioning system is to assure appropriate indoor air quality (IAQ), thermal comfort and acoustic environment. This goal should be achieved cost-effectively (ASHRAE 62.1:2007; ISO 16814:2008). The attention is particularly focused on energy savings (Burman et al., 2014). The recent trend toward minimizing ventilation within houses in order to reduce energy consumption for heating and cooling leads to an increase in indoor air pollution level (Daisey et al. 2003; Crump, 2011; Yu et.al., 2009; Chung et al., 2001). The deterioration of indoor air quality negatively affects human health, safety, productivity, and wellness (Turune et al., 2014; Haverinen-Shaughnessy et al., 2015). One of main roles in IAQ assessment is played by indicators. In practice, it is difficult to determine the reference parameters describing chemical properties of air inside a building, because the "quality" of indoor air is not well defined. In case of thermal comfort, the situation is relatively simple. The main indicator for thermal comfort is room temperature or sometimes, a combination of temperature and humidity (specific enthalpy). Chemical indoor air quality may be characterized by a set of values and parameters extracted from qualitative and quantitative analysis of air. However, in many cases the link between the perception of air quality and the chemical composition as well as concentration levels of various substances is still unclear. The problem is also related to the possibilities to measure the indicators. The qualitative sensory analysis of indoor air is complex and subjective. The quantitative and qualitative analysis of indoor air composition is difficult to perform since this gas is a mixture of many substances. Usually, their concentrations are low. In addition, indoor air constituents are strongly affected by many factors. Due to it they display considerable temporal changes. For that reason, the applicability of traditional analytical methods is limited. Even if there are available measuring techniques for determination of the required parameter, the measuring device must fulfil certain requirements in order to be applicable for the ventilation control. For example, it should provide fast, stable and reliable output signals. They have to correspond to the value of the specified reference quantity which is determined. Incorrect measurements can lead to uncomfortable indoor microclimate or excessive use of energy as an effect of under- or over-ventilation of rooms.

The aim of work is to compare carbon dioxide (CO_2) and volatile organic compounds (VOCs) as indicators of IAQ.

2 CO_2 AND VOC$_S$

Carbon dioxide is one of the most important constituents of indoor air. In many buildings, a major source of this gas is human occupancy. Occupants emit CO_2 through exhalation. The emission rate depends on the number of people and their level of activity. Carbon dioxide at very high concentrations (greater than 5000 ppm) can pose a health risk. At concentrations commonly found in buildings, direct influence of this gas on human health is minimal, although symptoms such as drowsiness and decrease of perception are observed. Current technology enables easy and relatively inexpensive measurement of CO_2 (Mahyuddin etal., 2012). As the indicator, the carbon dioxide concentration level inside a building is considered from two points of view. Firstly, this quantity is closely related to the ventilation rate and therefore it can be useful to estimate building air exchange rates and the percentage of outdoor air intake at an air handler. CO_2 measurements are often used in demand controled ventilation (Fisk et al., 2010). Generally speaking, this system adapts the airflow rate continuously to the actual pollutant emissions from activities and processes in the room. In practice, it allows to adjust ventilation rates according to the actual CO_2 concentration, rather than using pre-determined rates e.g. based on maximum occupancy. Currently, CO_2 concentrations remain a rough and easily measurable surrogate for ventilation rate. Secondly, under some circumstances CO_2 is proposed as an indicator of general indoor air quality. This approach is justifiable only in buildings where metabolic or combustion sources of CO_2 are predominating. It means that this gas may be an indicator only for air quality related to human occupancy. Therefore the proposition to use CO_2 concentration as an indicator of occupant odors (odorous bioeffluents) and the acceptability of a space in terms of human body odor is controversial. Indoor CO_2 concentrations are particularly poor indicators of health risks in rooms with strong pollutant emissions from the building or building furnishings, particularly when occupant densities are low. For that reason, indicators which allows to adjust ventilation rates based on the measurement of other parameters are needed. This requirement can be fulfilled by volatile organic compounds.

VOCs include a variety of chemicals, some of which may have short- and long-term adverse health effects (Tucker, 2000; Wolkoff et al., 2006). They are emitted by a wide array of products numbering in thousands, e.g. paints and lacquers, paint strippers, cleaning agents, pesticides, building materials and furnishings, office equipment such as copiers and printers, correction fluids and carbonless copy paper, graphics and craft materials including glues and adhesives, permanent markers, and photographic solutions. Lack of standards for non-industrial buildings describing acceptable concentrations of many common volatile organic compounds is the limitation for the application of these substances as indicators of IAQ. On the other hand, the choice of an indicator based on VOCs is to a great extent dependent of the possibilities to measure this parameter. It is relatively simple to measure just one substance e.g. CO_2. However, regarding VOCs, indoor air is a combination of many different organic compounds at different concentrations. The commercially available gas sensors measure non-selectively a wide range of volatile substances and they do not provide selective information. Therefore indicator such a total concentration of VOCs is proposed. The serious disadvantage of this approach is the interpretation of measuring results.

3 EXPERIMENTAL

The analysis presented in this work was based on the experimental data collected during an indoor air quality study, performed in a university classroom. Nine days in the winter season 2014/15 were taken under consideration.

The classroom is located on the third floor of the building which was erected in 70-ties. Recently, it has been renovated, including thermo-isolation. Airtight windows were mounted. The room size is 9.6 x 7.2 x 3.2 m. It may host up to 40 students. Air exchange is realized by a natural ventilation.

The study focused on teaching hours. Measurements were performed from early morning till the evening in a continuous manner. They consisted in monitoring air parameters using following sensors: temperature sensor, relative humidity sensor, CO_2 sensor, photoionization detector (PID) sensor and semiconductor gas sensors (SGS). The last two techniques are considered as applicable for determining VOCs in indoor air. The measurement characteristics of the applied instruments are presented in Table 1. The temporal resolution of data collection was 1 min.

Table 1: Measuring characteristics of sensor devices applied in IAQ study

Sensor	Measuring principle	Measuring range	Accuracy	Resolution
Temperature sensor	thermistor NTC 10 Ω	-20 – 60 °C	±0.2 °C ± 0.15 % display reading	0.1 °C
Relative humidity sensor	capacitive sensor	5 – 100 %RH	±2 % (10 – 90 %RH); ±2.5 % outside	0.1 %
CO2 sensor	non dispersive infrared (NDIR)	0 – 5000 ppm	±50 ppm + 3 % display reading	1 ppm
Semiconductor gas sensor, TGS822	semiconductor gas sensor	5-10000 ppm organic solvents	N/A	N/A
Photoionization detector	Photoionization detection, 10.6 eV Krypton PID lamp	0.1 - 5000 ppm	± 5% display reading ± one digit	0.1 ppm

All measuring devices were located in one place, at the back of the classroom 1m over the floor, out of the direct influence of classroom occupants. In parallel with measurements, the number of people in the classroom was counted. In addition, there were noted down the times of windows/ doors opening or setting windows ajar.

4 METHODS

4.1 Pearson correlation

Sample correlation coefficient for two data vectors **p**=(x_1, x_2, …, x_n) and **q**=(x_1, x_2, …, x_n) in *n*-dimensional vector space is given by (Navidi, 2011):

$$R = \frac{\sum_{i=1}^{n}(y_i-\bar{y})\sum_{i=1}^{n}(x-\bar{x})}{\left(\sum_{i=1}^{n}(y_i-\bar{y})^2\sum_{i=1}^{n}(x-\bar{x})^2\right)^{1/2}} \tag{1}$$

where, $\bar{x}$ and $\bar{y}$ denote the sample mean of vector **p** and **q** respectively. In statistics and other sciences, R it is widely used as a measure of the degree of linear dependence between two variables. Correlation coefficient belongs to an interval <-1,1>. It is commonly accepted that values close to 1 indicate strong positive correlation, values close to -1 indicate strong negative correlation and values close to zero announce lack of correlation. A precise determination of the statistical significance/insignificance of the correlation coefficient is based on *t*-test (for $n \geq 3$).
If the null hypothesis H_0: R=0 is true, the test statistic (Navidi, 2011):

$$t = \frac{R}{(1-R^2)^{1/2}}(n-1)^{1/2} \tag{2}$$

has t distribution with $\nu = n-2$ degrees of freedom. In case of testing the null hypothesis against the alternative H_a: $R \neq 0$, at the significance level α, the critical interval is $I = \left(-\infty, -t_{\alpha/2,\nu}\right\rangle \cup \left\langle t_{\alpha/2,\nu}, \infty\right)$. If sample test statistic belongs to critical interval, correlation coefficient may be considered statistically significant at the significance level α.

4.2 Simple linear regression

The formula (Navidi, 2011):

$$y = ax + b + \varepsilon \tag{3}$$

describes a simple linear regression of variable *y* with respect to variable *x*. This model is widely applied in science to quantitatively determine a linear relationship between two variables. The parameters *a* and *b* are the slope, and offset, respectively. Only the deterministic part of the dependent variable variability may be explained by linear transformation of *x* (i.e. the term *ax*+*b*). Random component, ε represents the part which is not explained. The parameters of simple regression are typically calculated using least squares method.
We examined the quality of fit for regression models developed in this work using coefficient of variation of the RMSE. It is given by (IDRE, 2015)

$$CV = \frac{RMSE}{\bar{y}} \tag{4}$$

where RMSE is the root mean square error and $\bar{y}$ is the mean of the dependent variable. *CV* compares the size of RMSE in relation to the average value of the variable described by a model. RMSE is computed from

$$RMSE = \left(\frac{\sum_{i=1}^{n}(y_i - \hat{y}_i)^2}{n}\right)^{1/2} \tag{5}$$

where y and $\hat{y}$ are the dependent variable and the output of regression model, respectively. All calculations utilized in this work were done in MATLAB environment.

5 RESULTS AND DISCUSSION

The postulate that CO_2 and VOCs could be considered as alternative indicators of indoor air quality is based on the assumption that both have common source - building occupants. Hence, it could be expected that the temporal variation of the two indicators follows a similar pattern. In consequence, they may be highly correlated. In this paper, we have analyzed this issue in detail.

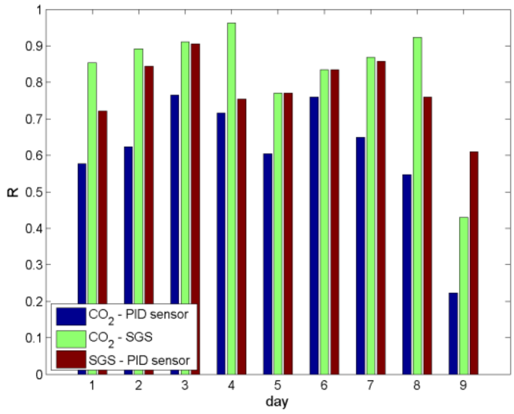


Figure 1: Pearson correlation between CO_2 concentration (NDIR sensor), PID sensor and semiconductor gas sensor responses. Correlation coefficients refer to the time scale of one day.

In Fig. 1 we compare Person correlation between the time series of CO_2 concentration, PID sensor and semiconductor gas sensor responses, which were recorded during nine days. The presented correlation coefficients correspond to the time scale of one day (strictly - working hours). Based on the bar plot we see that, in this time scale, correlations between the considered variables were usually relatively high; in particular for pairs: CO_2 – SGS and SGS – PID sensor. In spite of day to day differences, such results could be interpreted as indicating an overlap in information provided by CO_2 measurement and VOCs indication based on semiconductor gas sensor (green bars), as well as the interchangeability of VOCs assessment based on two techniques – semiconductor gas sensing and photoionization detection (brown bars). Contrarily, the information shared by CO_2 data and PID sensor responses was quite small (blue bars).

However, the assumption that observations based on correlation analysis which refers to the time scale of one day are true in any time scale may not be valid; In particular for shorter time scales. In fact, such scales are more important from the point of view of providing a useful information about air quality to the ventilation system control. Lack of validity may be caused by the fact that indoor air parameters usually display a substantial temporal variation during the day, especially during working (teaching) hours. In our study, we also noted this phenomenon, as illustrated in Fig. 2 for an exemplary day 9.

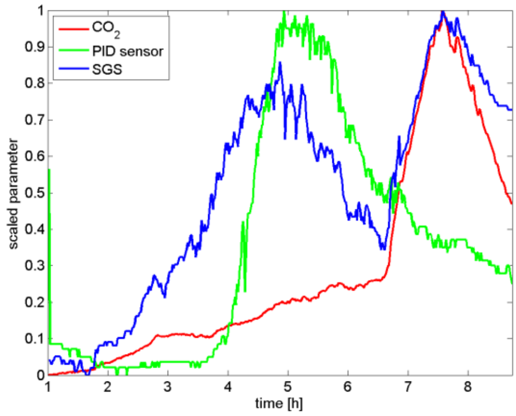


Figure 2: Temporal variation of CO_2 concertation, PID sensor and SGS responses in the time scale of one day (working hours at the university, day 9). Due to fact that the orders of magnitude of parameters are different measurement data was scaled to the range <0,1> for displaying it in one plot.

While displaying temporal variability, CO_2 concentration, PID sensor and SGS responses generally did not behave in the same way, as shown in Fig. 2. Namely, their increase, decrease or lack of change were not synchronized. Hence, it couldn't be expected that the correlation between the parameters is maintained at the same level over entire period of several hours.

In further analysis, there were considered correlations referring to shorter time scale. We chose 30 minutes. This scale was a compromise between the minimum duration of two major events which take place in the studied object, the break (15 minutes) and classes (45 minutes). The time window of this size was moved along the time series of data collected over the period of one day in order to cut out 30 min long time subseries. Correlation coefficients were calculated for the set of subseries associated with each position of the window. The results of the analysis for an exemplary day 9 are shown in Fig. 3. We distinguished statistically insignificant correlation coefficients (α=0.01) using red markers.

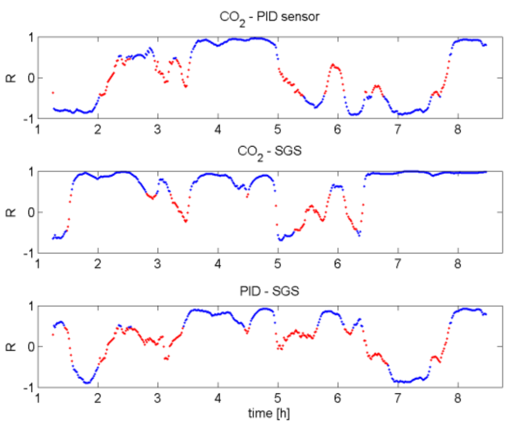


Figure 3: One day evolution of Pearson correlation between CO_2 concentration, PID sensor and SGS responses, computed in the time scale of 30 min (day 9). The results were obtained using moving time window technique.

The results shown in Fig. 3, confirmed that correlation between the examined indoor air parameters was neither constant nor high during the day. Correlation coefficients calculated using time scale of 30 min exhibited high temporal variation and they took values from a full available range of <-1,1>. One shall draw special attention to the fact that, in some periods of time, the coefficients were not statistically significant (red markers in Fig. 3). Insignificant correlation means that the co-variation of the two variables is negligible. This in turn implies that unknown changes of one variable may not be deduced based on the known behavior of the other variable. In other words, the parameters carry mutually irrelevant information. In Fig. 4 we summarized the fractions of daytime, when the 30 min time scale correlations were statistically insignificant. The results obtained for 9 days were considered jointly.

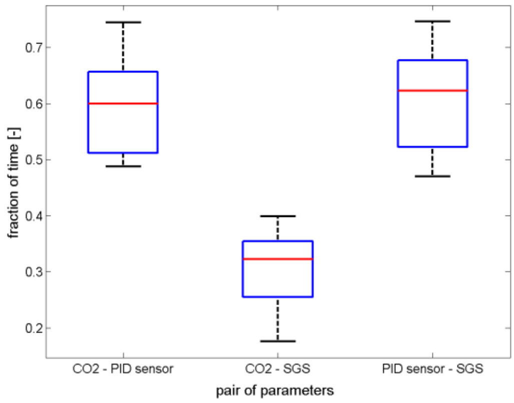


Figure 4: Fraction of daytime when Person correlation between CO_2 concentration, PID sensor and SGS responses was statistically insignificant. The plot summarizes nine days of IAQ study.

As shown in Fig. 4, on average, short time scale correlations CO_2 – PID sensor as well as PID sensor – SGS were statistically insignificant during as much as 60 % of daytime. For the CO_2 – semiconductor gas sensor pair the respective time fraction was more than 30 %. Based on these results, we may infer that in short term perspective, time series of CO_2 concentration conveyed different information than the corresponding time series of PID sensor responses. In addition, the PID sensor responses were not a credible source of information provided by gas sensors.

Further, we estimated the error of inference about one IAQ indicator based on other parameter, using a univariate linear regression model. Two approaches were compared. In the first approach, one regression model was parameterized based on the data collected during one day. This model was later applied to infer about all values of the dependent variable on that day. The corresponding errors are shown in Fig. 5a. In the second approach a set of regression models was applied. An individual model was parameterized, based on the data associated with one 30 min time interval, and it was applied to infer about the value of the dependent variable in the middle of this interval. A set of models was obtained using moving time window technique. The corresponding errors are shown in Fig. 5b.

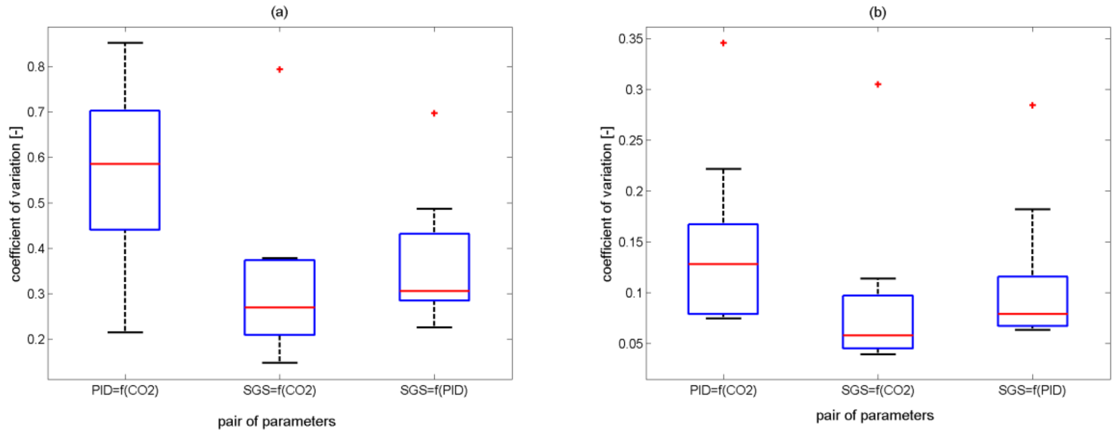


Figure 5: Error of inference about the measured values of CO_2 concentration, PID sensor and SGS responses using: a) simple linear regression model – parameterized for the time scale of one day, b) set of simple linear regression models – parameterized for the time scale of 30 min, using moving time window technique. The plot summarizes nine days of IAQ study.

In Fig. 5 we see that the application of model developed for the period of one day resulted in high errors. The average coefficient of variation when inferring about PID responses based on CO_2 concertation was 60 %. In case of modeling semiconductor gas sensor responses using CO_2 concentration the error was about 25 %. Reconstruction of SGS responses based on PID sensor responses was loaded with *CV*=30 %. These unacceptably hing errors were a consequence of the fact that the models developed for the period of 1 day were not able to cope with explaining the short term variability of the dependent quantities. They flattened their variation, as shown in Fig. 6.

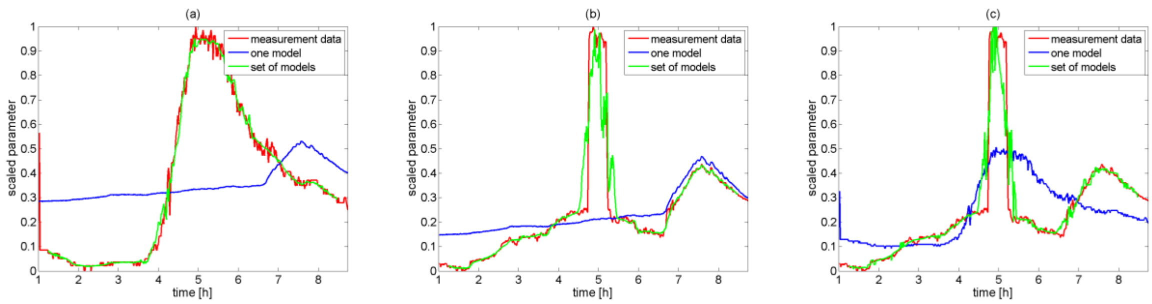


Figure 6: Results of modeling: a) PID=f(CO_2), b) SGS=f(CO_2) and c) SGS=f(PID) using simple linear regression model –parameterized for the time scale of one day and set of models – parameterized for the time scale of 30 min, using moving time window technique.

When applying short time scale models, the results of mutual inference about IAQ indicators were very good. In all considered cases coefficients of variation were smaller than 15 %. However, we observed that models coefficients were highly dependent on the position of moving time window. Therefore, if one wanted to deduce accurately changes of one variable, based on the other indicator, the model would have to be reparametrized every 30 min. Hence, the instruments providing data on both IAQ indicators would have to be available permanently. It is against the logics of the alternative use of indicators.

6 CONCLUSIONS

Several conclusions were drawn based on the results obtained in this work. (1) The co-variation of measured quantities showed high temporal variability. It means that the results of

CO_2 and TVOCs measurements conveyed different information about IAQ, as a function of time. (2) The analytical method had strong influence on the information obtained. In particular, the discrepancy was observed when the NDIR and PID method were compared. (3) Techniques applied for VOCs measurements provided different information about these substances. This fact was reflected in the behaviour of correlation between PID and semiconductor gas sensor measurements. VOCs contained in indoor air have different properties and variously affect IAQ. Therefore, a wide spectrum of information about air composition is required to control the quality of this gas. (4) Room occupancy was not clearly related to the direction or magnitude of correlation between the considered quantities.
The results of our work entitle to conclude that the total concentration of VOCs should be taken into account as indicator of IAQ in addition to CO_2. It is already possible from the technological point of view. Current sensor technology allows to provide information about these substances in air, easily and relatively inexpensively.

7 ACKNOWLEDGEMENTS

This contribution was supported by the project: "The variability of physical and chemical parameters in time as the source of comprehensive information about indoor air quality". The project is financially supported by the National Science Center, Poland, under the contract No. UMO-2012/07/B/ST8/03031.

8 REFERENCES

ASHRAE 62.1:2007, *Ventilation for acceptable indoor air quality*.

Burman E., Mumovic D., Kimpian J. (2014). Towards measurement and verification of energy performance under the framework of the European directive for energy performance of buildings. *Energy*, 77, 153–163.

Chung K.C., Hsu S.P. (2001). Effect of ventilation pattern on room air and contaminant distribution. *Building and Environment*, 36, 989–998.

Crump D. (2011). Nature and Sources of Indoor Chemical Contaminants in *44. Indoor Air Quality*, Guardino S. X., Editor, *Encyclopedia of Occupational Health and Safety*, Stellman J.M.,. Editor-in-Chief, International Labor Organization, Geneva.

Daisey J.M., Angell W.J., Apte M.G. (2003). Indoor air quality, ventilation and health symptoms in schools: an analysis of existing information. *Indoor Air*, 13(1), 53–64.

Fisk W.J., Sullivan D.P., Faulkner D., Eliseeva E. (2010), *CO_2 monitoring for demand controlled ventilation in commercial buildings*, Report to the California Energy Commission Public Interest Energy Research Program.

Haverinen-Shaughnessy U., Shaughnessy R.J., Cole E.C., Toyinbo O., Moschandreas D.J. (2015), An assessment of indoor environmental quality in schools and its association with health and performance. *Building and Environment*, in press.

IDRE (2015) http://www.ats.ucla.edu/stat/mult_pkg/faq/general/coefficient_of_variation.htm

ISO 16814:2008, Building environment design -- Indoor air quality -- Methods of expressing the quality of indoor air for human occupancy.

Mahyuddin N., Awbi H.B. (2012), A review of CO2 measurement procedures in ventilation research. *International Journal of Ventilation*, 10(4), 353–370.

Navidi, W (2011) *Statistics for Engineers and Scientists*, McGraw Hill.

Tucker, W.G., Chapter 31. *Volatile organic compounds*, in Indoor Air Quality Handbook (2000). Ed. by. Spengler J.D., Samet, J.M. and McCarthy J.F., McGraw Hill: New York City, 31.1–31.20.

Turune M., Toyinbo O., Putus T., Nevalainen A., Shaughnessy R., Haverinen-Shaughnessy U. (2014). Indoor environmental quality in school buildings, and the health and wellbeing of students. *International Journal of Hygiene and Environmental Health*, 217, 733–739.

Wolkoff P., Wilkins C.K., Clausen P.A., Nielsen G.D. (2006). Organic compounds in office environments - sensory irritation, odor, measurements and the role of reactive chemistry. *Indoor Air*,. 16(1), 7–19.

Yu B.F., Hu Z.B., Liu M., Yang H.L., Kong Q.X., Liu Y.H. (2009). Review of research on air-conditioning systems and indoor air quality control for human health, *International Journal of Refrigeration*, 32(1), 3–20.

IMPACT OF VENTILATION AND RECIRCULATION RATES ON EXPOSURE TO AND INTAKE OF OZONE AND ITS INITIATED CHEMISTRY PRODUCTS: MASS BALANCE MODEL EVALUATION

Moshood Olawale Fadeyi[1,2] and Kwok Wai Tham[*3]

[1] Berkeley Alliance for Research in Singapore, 1 Create Way, #11-01, Singapore 138602, Singapore
[2]School of Civil and Environmental Engineering, Nanyang Technological University, N1-01a-29, 50 Nanyang Avenue, Singapore 639798, Singapore

[3]Department of Building, School of Design and Environment, National University of Singapore SDE 1, 4 Architecture Drive, Singapore 117566, Singapore
**Corresponding author: bdgtkw@nus.edu.sg*

ABSTRACT

A mass balance model is used to examine the impact of two ventilation (1 /h and 2 /h) and recirculation (7 /h and 14 /h) rates on concentrations, exposure to and intake of ozone (of outdoor origin) and secondary organic aerosols (SOA) derived from the ozone initiated chemistry in indoor environment. Measured data from several experimental studies conducted by the authors in a 236m^3 field environmental chamber (FEC) configured to simulate an office are used for the mass balance model evaluations. At steady state, increase in ventilation rate increases exposure to and intake of indoor ozone, but reduces exposure to and intake of SOA. Increase in recirculation rate reduces exposure to and intake of ozone and SOA. Increase in outdoor ozone concentration increases exposure to and intake of ozone and SOA. As expected, indoor ozone and SOA concentrations are lower for human occupancy scenario than non-human occupancy scenario. Interestingly, human sink of ozone and SOA is much lower at higher recirculation rate than at lower recirculation rate - this is a new finding that has not been reported in the literature.
.

KEYWORDS

Ventilation rate; Recirculation rate; Ozone; Indoor chemistry; Exposure and intake

1 INTRODUCTION

A foremost energy challenge in the 21st century is influencing how buildings are being designed to reduce exchange between outdoor and indoor air (Mudarri, 2010). The rate at which outdoor and indoor air are deliberately exchanged (ventilation) could influence building occupants' exposure to and intake of pollutants of outdoor and indoor origins. In this paper, pollutant of interest is ozone. Ozone is chosen for three reasons.

First, the outdoors is a major source of ozone in indoor environment (Weschler, 2000). Second, increase in exposure to and intake of ozone has been associated with increase in hospital visits, morbidity and mortality rates (Bell et al. 2004). Third, ozone is a powerful oxidizing agent and products of its initiated chemistry could be more harmful than ozone alone (Rohr, 2013).

Due to energy concerns, recirculation of a larger proportion of indoor air has become a normal practice in air-conditioned buildings located in regions with warm and humid outdoor air. The effects of ventilation and recirculation rates on ozone and its initiated chemistry products have been reported in the literature (Zuraimi et al. 2007; Fadeyi et al. 2009; Fadeyi, 2015). These studies were done with no humans present in the indoor environment. However, in reality humans will be present in buildings – and this should be the primary reason why these studies were conducted in the first place. Furthermore, evidence have shown that when human are present in the indoor environment, they change the dynamics and concentrations of ozone and its initiated chemistry products (see references in Weschler, 2015).

Fadeyi et al (2013) examined the impact of human presence on ozone and secondary organic aerosols (SOA) derived from ozone initiated chemistry products, when the air-conditioned system recirculates a large proportion of indoor air, and ventilation transports outdoor air, containing ozone, to indoor environment. The key finding from their study is that human presence would reduce concentrations of indoor ozone and SOA. Fadeyi et al. (2013) and previous studies did not specifically address exposure to and intake of these pollutants – ozone and SOA. There is also no evidence on how changes in ventilation and recirculation rates would influence the dynamics and concentrations of ozone and SOA, exposure and intake when human are present in indoor environment for considerable amount of time.

To bridge this gap in knowledge, a mass-balance model evaluation is used to examine how ventilation and recirculation rates would influence concentrations, exposure to and intake of ozone and SOA derived from ozone initiated chemistry. Additionally, impact of increased outdoor ozone concentration on exposure to and intake of ozone and SOA is examined. This study gives indication of how changes in ventilation and recirculation rates could influence concentrations, exposure to and intake of pollutants of outdoor and indoor origins.

2 METHODS

The mass balance model adopted in this study used measured data from experimental studies conducted by the authors in a 236m^3 field environmental chamber (FEC) configured to simulate an office. The air system of the FEC operates under recirculation mode with a recirculation loop of ~30m^3. Details of the experiments and FEC can be found in Fadeyi et al. (2009, 2013) and Zuraimi et al (2007). The mass balance model (see Equations 1and 2) used in this study has been validated with experiments conducted by Fadeyi et al. (2009).

In this study, the mass balance model evaluation is used to evaluate a hypothetical 3-h exposure and intake period – time from when steady state has been achieved in the FEC. We did not account for exposure and intake during transient reactions – period leading to the steady state – that occurred when occupants just entered the FEC. Specifically, the mass balance model is used to examine the impact of two ventilation (1 /h and 2 /h) and two recirculation (7 /h and 14 /h) rates on concentrations, exposure to and intake of ozone (of outdoor origin) and SOA derived from ozone reacting with limonene emitted in the FEC at a constant emission rate of 180mg/h throughout the steady state 3-h occupancy of the FEC. The impact of increase in ventilation rate from 1 /h to 2 /h is calculated for doubling the "emission rate of ozone" – with assumed outdoor concentration of 157µg/m^3 (80ppb) – transported from outdoor to indoor. Ozone emission "E_{O3}" rates into the FEC are calculated to be 41.8 and 83.5mg/h at ventilation rates of 1 /h and 2 /h, respectively.

$$[C_{O3}]_{ss} = \frac{(\eta_v\lambda_v + \rho\lambda_L)\, C_{out} + (E_{O3}/V)}{(\lambda_v + \lambda_L + k_{sr,O3} + k_{O3,lim}[lim]_{ss})} \quad (1)$$

$$[C_{SOA}]_{ss} = \frac{\{ý_{SOA}\%\ k_{O3,lim}[lim]_{ss}\ [C_{O3}]_{ss})/V\}}{\{\lambda_v + \lambda_L + f\,(\lambda_v + \lambda_{recirc}) + k_{sr,SOA}\}} \quad (2)$$

"$[C_{O3}]_{ss}$" is the concentration of ozone in the FEC at steady state. "$(\eta_v\lambda_v)\ C_{out}$" is the contribution of outdoor environment to ozone in the FEC through fraction (η_v=1, i.e. assuming 100% penetration of outdoor ozone to indoor as it is introduced together with ventilation) of outdoor ozone (C_{out} – assumed to be 157µg/m) that enters the FEC through dedicated outdoor inlet meant for outdoor air.

266m^3 is the volume "V" of the system – summation of FEC and recirculation loop volumes. The values, based on experimental data from Fadeyi et al. (2009), used for "$k_{sr,O3}$" – the rate at which ozone is removed by indoor surfaces – are 2.96 /h and 6.73 /h for ventilation rates of 1 /h and 2 /h, respectively after adjusting for assumed no leakages in the system – i.e. λ_L = 0, thus penetration factor "ρ" is also "0".

$k_{O3,lim}[lim]_{ss}$" is the rate at which ozone is removed from the FEC by limonene through chemical reactions; 0.018 ppb^{-1}h^{-1} is used for "$k_{O3,lim}$" – the second rate constant for the reactions between ozone and limonene (Atkinson et al. 1990);"$[lim]_{ss}$" is the concentration of limonene at steady state.

"$[C_{SOA}]_{ss}$" is the concentration of SOA formed in the FEC at steady state. "$\{ý_{SOA}\%\ k_{O3,lim}[lim]_{ss}\ [C_{O3}]_{ss})$" is the rate at which SOA is generated in the FEC, while $ý_{SOA}$ is the yield of formation of SOA. 0.16 and 0.14 are used as "$ý_{SOA}$" values for ventilation rate (λ_v) of 1 /h and 2 /h, respectively (Youssefi and Waring, 2014). 0.39 is used for new filter filtration efficiency "f" (Fadeyi et al. 2009). "$k_{sr,SOA}$" is the rate at which SOA is removed by indoor surfaces. 0.5 /h and 1/h are "$k_{sr,SOA}$" values at recirculation rate "λ_{recirc}" of 7 /h and 14 /h (Zuraimi et al. 2007).

Equations 3 and 4 show parameters that would determine concentrations of ozone and SOA in the FEC, respectively for a scenario when the FEC is occupied by 18 occupants – typical capacity of the FEC. "$k_{sr,O3\_human}$", rate at which ozone is removed by human surface is calculated, from experimental data, to be ~2 /h for both recirculation rates of 7/h (Fadeyi et al. 2013). The same value – ~2 /h – is hypothetically used for recirculation rate of 14 /h. "k_{O3_inhal}" – the rate at which ozone is removed by human through inhalation – is assumed to be negligible.

"k_{sr,SOA_human}" is the rate at which SOA is removed by human – 0.1 /h is used for recirculation rate of 7 /h (Fadeyi et al. 2013). 0.2 /h (value for "k_{sr,SOA_human}") is hypothetically used for recirculation rate of 14 /h. 0.04 /h is used for "k_{SOA_inhal}" – the rate at which SOA is removed by human through inhalation (Fadeyi et al. 2013). Other parameters have been defined earlier. 3-h ozone and SOA exposure are calculated by multiplying calculated ozone and SOA concentrations, using Equation 3 and 4, values by 3 – hypothetical 3-h exposure at steady state and intake period.

$$[C_{O3}]_{ss} = \frac{(\eta_v\lambda_v + \rho\lambda_L)\, C_{out} + (E_{O3}/V)}{(\lambda_v + \lambda_L + k_{sr,O3} + k_{sr,O3\_human} + k_{inhal} + k_{O3,lim}[lim]_{ss})} \quad (3)$$

$$[C_{SOA}]_{ss} = \frac{\{ý_{SOA}\%\ k_{O3,lim}[lim]_{ss}\ [C_{O3}]_{ss})/V\}}{\{\lambda_v + \lambda_L + f\,(\lambda_v + \lambda_{recirc}) + k_{sr,SOA} + k_{sr,\ SOA\_human} + k_{SOA\_inhal}\}} \quad (4)$$

The 3-h ozone and SOA intakes by the occupants are calculated using Equations 5 and 6 – written based on knowledge gained from Weschler (2006). Scenario where adults are the exposed occupants was examined. Assuming occupants performed sedentary activities throughout their occupancy of the FEC, their breathing rate (BR_{indoor}) was assumed to be $0.54m^3/h$ (Weschler, 2006).

$$\text{3-h } [C_{O3}]_{ss} \text{ intake} = 3x\ [C_{O3}]_{ss} \text{ x } BR_{indoor} \quad (5)$$

$$\text{3-h } [C_{SOA}]_{ss} \text{ intake} = 3x\ [C_{SOA}]_{ss} \text{ x } BR_{indoor} \quad (6)$$

3 RESULTS

Increase in ventilation rate – from 1 /h to 2 /h, when source of ozone is from outdoors, increases indoor ozone concentration (see Figure 1). This impact of increased ventilation rate on indoor ozone is reduced by increasing recirculation rate from 7 /h to 14 /h. At 7 /h, the differences in ozone concentrations are calculated to be 6.7 and 5.3µg/m^3 for "human" and "no human" status, respectively. At 14 /h, the differences in ozone concentrations are calculated to be 4.5µg/m^3 for both occupancy statuses.

Increase in recirculation rate – from 7 /h to 14 /h – reduces indoor ozone concentrations. This effect is more evident when the FEC is not occupied. When FEC is occupied, increased recirculation rate reduces ozone concentrations by 10.3 and 12.5µg/m^3 for ventilation rates of 1 /h and 2 /h, respectively. When FEC is not occupied, increased recirculation rate reduces ozone concentrations by 15.9 and 16.7µg/m^3 for ventilation rates of 1 /h and 2 /h, respectively.

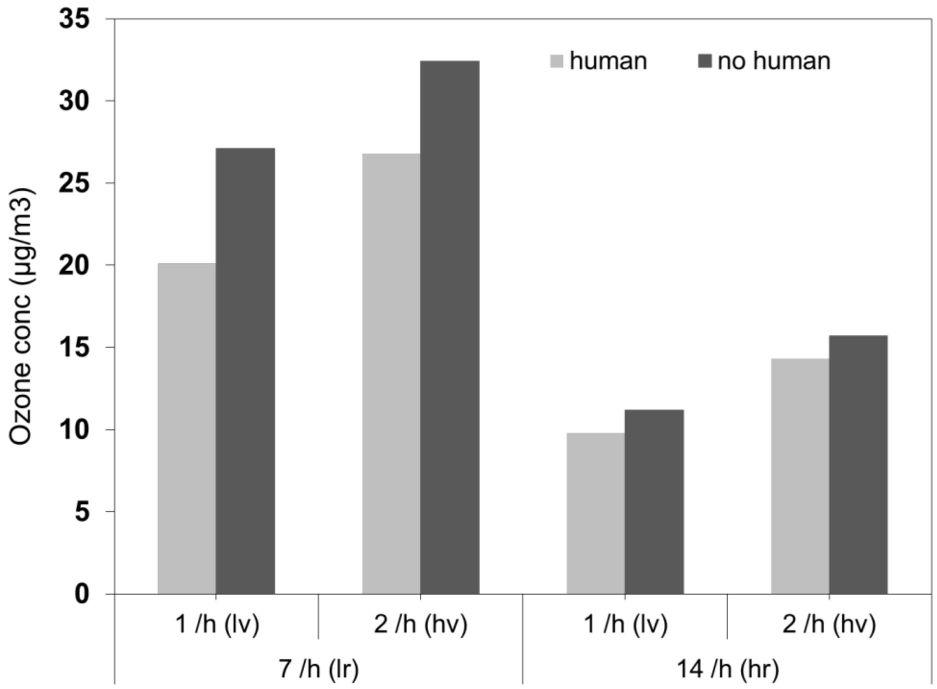


Figure 1: Impact of ventilation and recirculation rates on ozone concentration during human and non-human presence in the FEC.

As evident in Figure 1, additional sink provided by human presence causes indoor ozone concentrations to be lower than when it is not occupied (“no human”), irrespective of ventilation and recirculation rates. Interestingly, the sink effect diminishes when recirculation rate is increased from 7 /h to 14 /h. At recirculation rate of 7 /h, the decrease is calculated to be 7 and 5.6μg/m^3, for ventilation rates of 1 /h and 2 /h, respectively. At recirculation rate of 14 /h, the decrease is calculated to be 1.4μg/m^3 for both ventilation rates.

Figure 2 shows impact of ventilation and recirculation rates on SOA concentrations during human and non-human presence in the FEC. Increase in ventilation rate – from 1 /h to 2 /h – reduces SOA concentrations. This impact of increased ventilation rate on indoor SOA concentrations is reduced by increasing recirculation rate from 7 /h to 14 /h. At 7 /h, the differences in SOA concentrations are calculated to be 2.5 and 3.6μg/m^3 for “human” and “no human” status, respectively. At 14 /h, the differences in ozone concentrations are calculated to be 0.6 and 0.8 μg/m^3 for “human” and “no human” status, respectively.

Increase in recirculation rate – from 7 /h to 14 /h – reduces SOA concentrations (see Figure 2). This effect is more evident when the FEC is not occupied for each of the ventilation rate. At 1 /h, increased recirculation rate reduces SOA concentrations by 3.4 and 4.7μg/m^3 for “human” and “no human”, respectively. At 2 /h, increased recirculation rate reduces SOA concentrations by 1.5 and 1.9μg/m^3 for “human” and “no human”, respectively.

Lower SOA concentrations occur when the FEC is occupied than when it is not occupied, irrespective of ventilation and recirculation rates (see Figure 2). Like in the case of ozone, sink effect due to human presence diminishes when recirculation rate is increased from 7 /h to 14 /h. At recirculation rate of 7 /h, the decrease is calculated to be 1.6 and 0.5μg/m^3 for ventilation rates of 1 /h and 2 /h, respectively. At recirculation rate of 14 /h, the decrease is calculated to be 0.3 and 0.1μg/m^3 for ventilation rates of 1 /h and 2 /h, respectively.

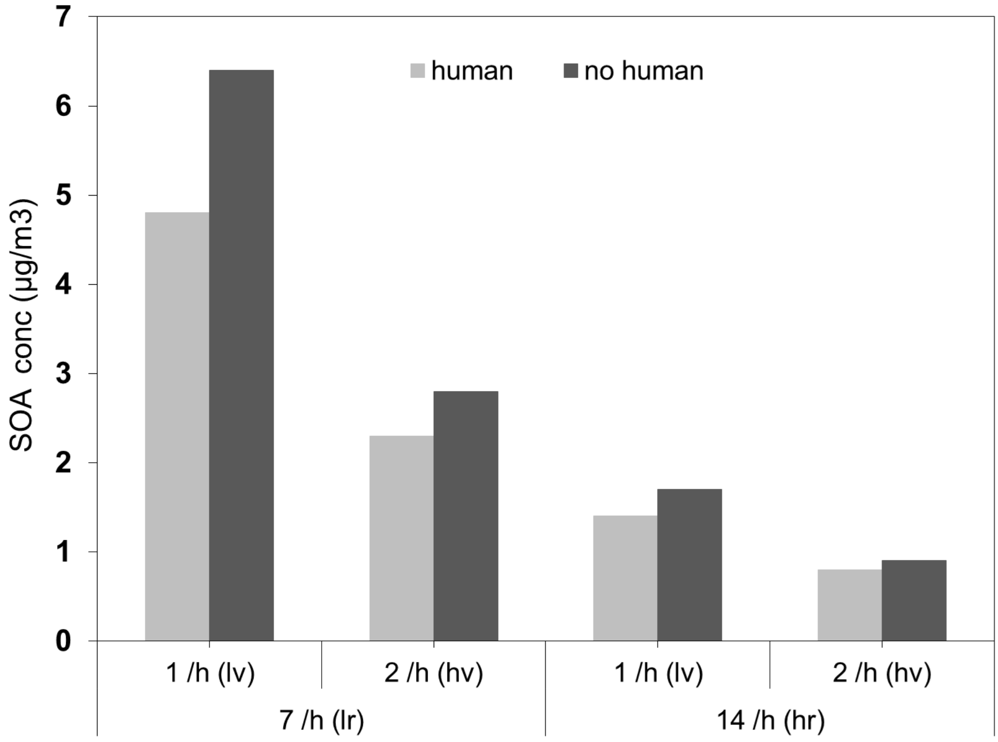


Figure 2: Impact of ventilation and recirculation rates on SOA concentrations during human and non-human presence in the FEC.

Figure 3 shows 3-h exposure to and intake of ozone at different ventilation and recirculation rates. Increase in ventilation rate – from 1 /h to 2 /h – increases building occupants' 3-h ozone exposures and intakes. The effect of increased ventilation rate causing higher exposure and intake is more evident at lower recirculation rate. Increases in ozone exposures are calculated to be 20.1 and 13.4µg/m^{3}h at recirculation rates of 7 /h and 14 /h, respectively. Increases in ozone intakes are calculated to be 10.9 and 7.2µg at recirculation rates of 7 /h and 14 /h, respectively.

Increase in recirculation rate – from 7 /h to 14 /h – reduces occupants' 3-h ozone exposures and intakes. The effect of increased recirculation rate causing lower ozone exposure and intake is more evident at higher ventilation rate when more outdoor ozone is transported indoor. Reductions in ozone exposures, caused by increasing recirculation rate, are calculated to be 31 and 37.7µg/m^{3}h at ventilation rates of 1 /h and 2 /h, respectively. The reductions in ozone intakes, caused by increasing recirculation rate, are calculated to be 16.7 and 20.4µg at ventilation rates of 1 /h and 2 /h, respectively.

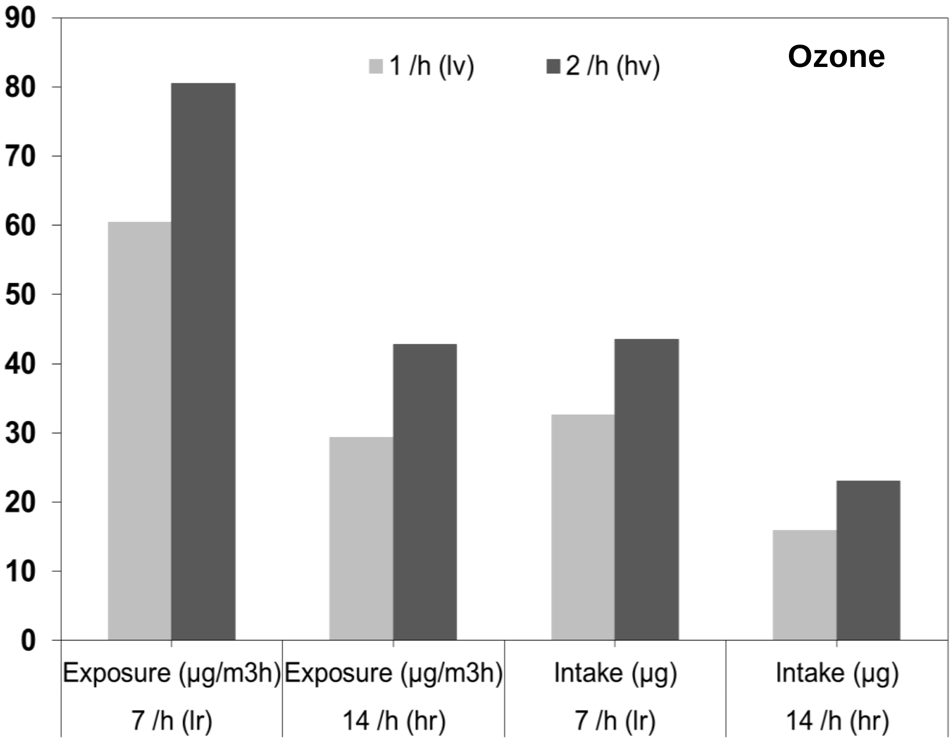


Figure 3: 3-h exposure to and intake of ozone at different ventilation and recirculation rates

Figure 4 shows 3-h exposure to and intake of SOA at different ventilation and recirculation rates. Increase in ventilation rate – from 1 /h to 2 /h – reduces occupants' 3-h SOA exposures and intakes. The effect of increased ventilation rate causing lower SOA exposure and intake is more evident at lower recirculation rate. Reductions in SOA exposures, caused by increasing ventilation rate, are calculated to be 7.6 and 1.9µg/m^{3}h at recirculation rates of 7 /h and 14 /h, respectively. Reductions in SOA intakes, caused by increasing ventilation rate, are calculated to be 4.1 and 1µg at recirculation rates of 7 /h and 14 /h, respectively.

Increase in recirculation rate – from 7 /h to 14 /h – reduces building occupants' 3-h SOA exposures and intakes. The effect of increased recirculation rate causing lower SOA exposure and intake is more evident at lower ventilation rate. Reductions in SOA exposures, caused by increasing recirculation rate, are calculated to be 10.2 and 4.5μg/m^{3}h at ventilation rates of 1 /h and 2 /h, respectively. Reductions in SOA intakes, caused by increasing recirculation rate, are calculated to be 5.5 and 2.4μg at recirculation rates of 7 /h and 14 /h, respectively.

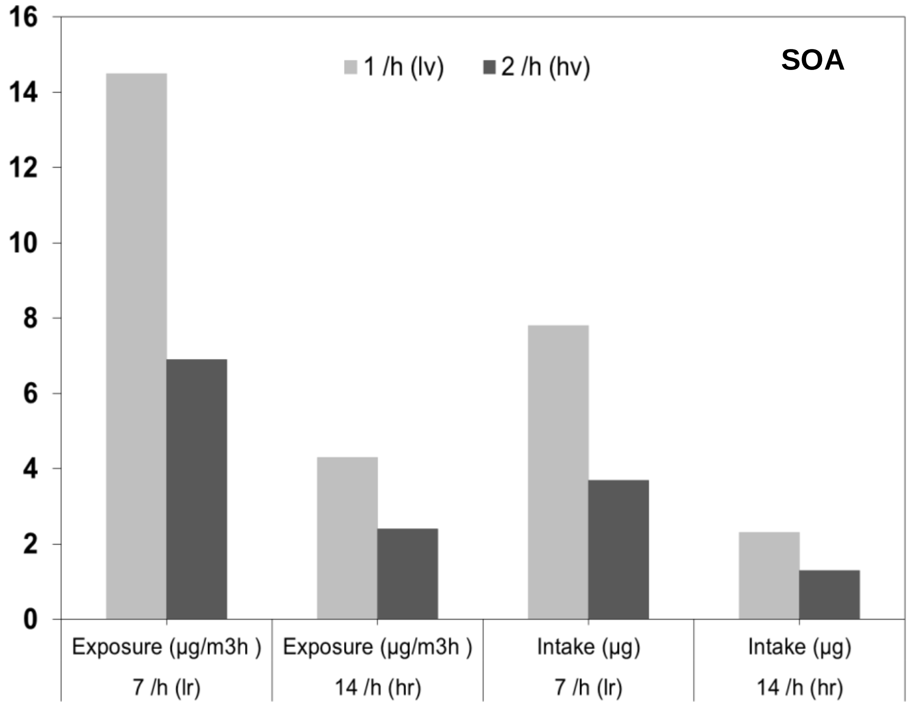


Figure 4: 3-h exposure to and intake of SOA at different ventilation and recirculation rates

Figure 5 shows exposure to and intake of ozone and SOA at different outdoor ozone concentrations. Increase in outdoor ozone concentration increases exposure to and intake of ozone and SOA in the FEC.

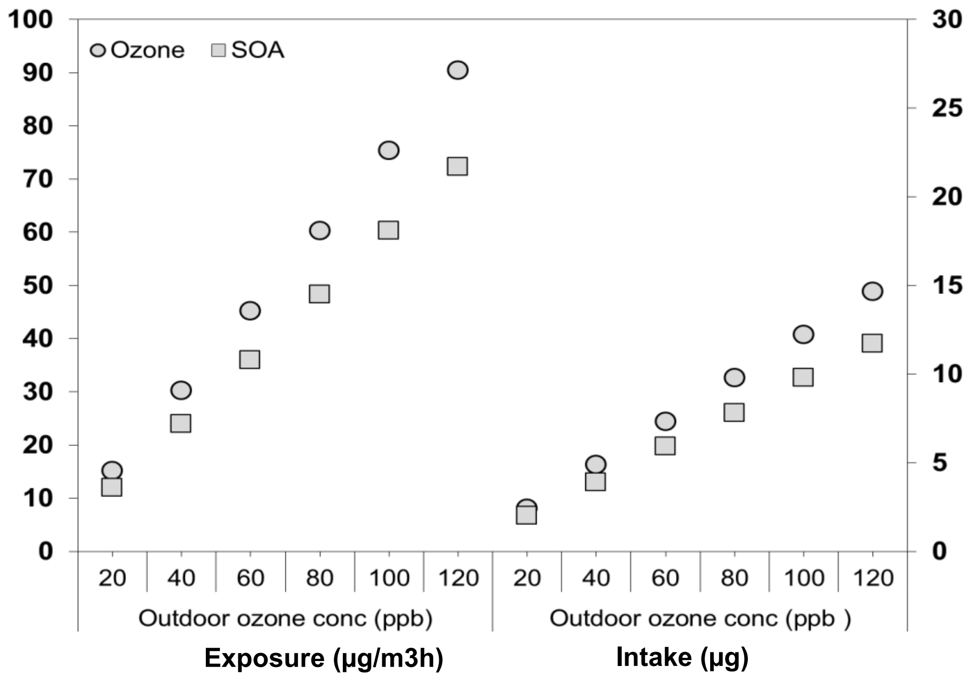


Figure 5: Exposure to and intake of ozone and SOA at different outdoor ozone concentrations

4 DISCUSSION

4.1 Impact of ventilation and recirculation rates and human presence

With ozone coming from outdoors, increase in ventilation rate will increase indoor ozone concentration because of increased outdoor to indoor transport of ozone. However, increase in ventilation rate did not increase SOA concentration because SOA is a pollutant of indoor source. In this case, for every increase in ventilation rate, dilution of SOA will increase. Additionally, the resident time – available time – for ozone to react with limonene to generate SOA is lower at higher ventilation rate (Weschler and Shield 2013). This phenomenon leads to lesser SOA formation at higher ventilation rate than at lower ventilation rate which has higher resident time. Boundary layer of surfaces in the FEC and air system become thinner as air moves at a much faster rate when recirculation rate is set higher (Zuraimi et al. 2007). The thinner nature of the surfaces enhance surface removal rate of ozone and SOA from the gas phase. This explains why higher recirculation rate causes lower indoor ozone and SOA concentrations.

When humans are present in the FEC, it means additional sink for ozone and SOA are introduced (Fadeyi et al. 2013). This explains why ozone and SOA concentrations are lower when FEC is occupied. At higher recirculation, the amount of ozone and SOA available at the gas phase is much lower, due to much higher surface removal of ozone and SOA by material surfaces in the FEC and air system. Thus, the ozone and SOA sink effect due to human presence, although still evident, diminishes – this observation is more evident in the case of ozone which is more prone to human sink effect than SOA. To the authors´ knowledge this is a new finding and it is being reported for the first time.

Decrease in ventilation rate increases SOA exposure and intake. However, by moving indoor air in the FEC and air system at a much faster rate – i.e. at higher recirculation rate – the effect of lower resident time causing more SOA will be diminished. This is due to increased surface removal of ozone caused by higher recirculation rate. In this study, ozone is assumed to be introduced from the outdoors. Thus, increased ventilation rate increases ozone exposure and intake. However, increased ozone exposure and intake caused by increased ventilation rate diminishes by moving air at a much faster rate. It is important to note that increased ventilation rate still increases outdoor to indoor transport of ozone, but the effect of this transport on increased indoor ozone concentration at steady state reduces because of higher surface removal of ozone at the gas phase by higher recirculation rate. These findings have implication on energy savings. Setting ventilation rate low especially when outdoor air poses threat of introducing pollutant(s) to indoor environment can be used to conserve energy, while using higher recirculation rate which has relatively lower energy consumption when compared to increasing ventilation rate to reduce indoor pollutants (Fadeyi and Tham 2008).

The difference between SOA exposure and intake when air is moved at a faster rate (14 /h) – and lower rate (7 /h) – is much higher than the difference experienced in the case of ozone. This is because higher recirculation rate and ventilation reduces SOA concentration; however, ventilation introduces more ozone while higher recirculation reduces ozone. Increase in outdoor ozone concentration, when there is air exchange between outdoor and indoor air, increases the vulnerability of indoor occupants to higher ozone and SOA exposure and intake. This finding poses a concern for cities and regions experiencing high outdoor pollutants, not necessary ozone alone, caused by natural and human factors. Such occurrence also poses threats to gaining benefits inherent in the usage of ventilation to reduce indoor pollutants.

4.2 Uncertainly in mass-balance modeling

Only few ozone data points were used for calculations of ozone surface removal rate by the system "$k_{sr}O_3$" at the higher recirculation rate with filter placed in the AHU. This is because ozone decayed at a faster rate. This limitation caused calculated "$ksrO_3$" values to be less accurate. "$k_{sr}O_3$" values are more accurate for lower recirculation rate because ozone decayed at a slower rate and more data were used for the calculation.

For simplification purpose, the mass-balance modelling of SOA particles was based on a single particle size (~100 nm) to represent all SOA particles. Computation of SOA particles by properly accounting for all SOA particles measured during the experiments would produce more accurate results. Changes in outdoor particle concentrations during experiments reduced the accuracy of measured data used for this mass balance modelling.

Due to instrumentation error, limonene concentration was computed and not measured directly by Fadeyi et al. (2009). All hypothetically assumed values may be less accurate. In this study, the system is assumed to have no leakage. However, leakage is a common occurrence in buildings. When leakage occurs, it may change the impact of outdoor air change rate on steady state indoor ozone concentration (Fadeyi et al. 2009). Finally, exposure and intake analysis reported in this paper are based on steady state period. The findings from this analysis may not be applicable for non-steady state period.

5 CONCLUSIONS

How would changes in ventilation and recirculation rates influence building occupants' exposure to and intake of ozone and by-products of ozone initiated chemistry? A mass balance model evaluation, based on several experiments conducted by the authors, was used in an attempt to answer this question. The following are the main conclusions from this exercise.

- Increase in ventilation rate, with ozone originating from outdoor, increases indoor ozone concentration, exposure and intake, but reduces SOA concentration, exposure and intake.
- Increase in recirculation rate reduces ozone and SOA concentrations, exposure and intake.
- Indoor ozone and SOA concentrations are lower for human occupancy scenario than non-human occupancy scenario.
- Human sink of ozone and SOA is much lower at higher recirculation rate than at lower recirculation rate. This finding has never been reported before in the literature.
- Increase in outdoor ozone concentration increases outdoor to indoor transport of ozone, thereby causing higher exposure to and intake of ozone and SOA.

6 ACKNOWLEDGEMENTS

The financial assistance from the National University of Singapore is gratefully acknowledged through grants R296-000-117-112. Dr. Moshood Olawale Fadeyi participation in this research is partly funded by the Republic of Singapore's National Research Foundation through a grant to the Berkeley Education Alliance for Research in Singapore (BEARS) for the Singapore-Berkeley Building Efficiency and Sustainability in the Tropics (SinBerBEST) Program. The contributions of Prof. Charles J. Weschler (CJW), Dr. Zuraimi M. Sultan, and Miss Wu W. Yi to experimental studies used for the mass balance model evaluation are gratefully acknowledged. We thank CJW for his advice in the construction of the mass balance model.

7 REFERENCES

Atkinson, R., Hasegawa, D,, Aschmann, S.M. (1990). Rate constants for the gas-phase reactions of ozone with a series of monoterpenes and related compounds at 296 2 K, *International Journal of Chemical Kinetics* 22, 871–887.

Bell, M.L., McDermott, A., Zeger, S.I., Samet, J.M., Dominici, F. (2004). Ozone and short-term mortality in 95 US urban communities, 1987-2000, *JAMA* 292, 2372-2378.

Fadeyi, M.O. (2015). Ozone in indoor environments: Research progress in the past 15 years, *Sustainable Cities and Society* 18, 78-94.

Fadeyi, M.O, Tham, K.W (2008). Impact of varying ventilation and recirculation rates in the presence of different filters scenarios on energy consumption, *Proceedings of Indoor Air 2008*, 17-22 August 2008, Copenhagen, Denmark- Paper ID:1064

Fadeyi, M.O., Weschler, C.J., Tham, K.W. (2009). The impact of recirculation, ventilation and filters on secondary organic aerosols generated by indoor chemistry, *Atmospheric Environment* 43, 3538-3547.

Fadeyi, M.O., Weschler, C.J., Tham, K.W., Wu, W.Y, Zuraimi, M.S. (2013). Impact of human presence on secondary organic aerosols derived from ozone-initiated chemistry in a simulated office environment, *Environmental Science & Technology* 47, 3933-3941.

Mudarri, D.H. (2010). Building codes and indoor air quality, Prepared for Office of Radiation and Indoor Air, Indoor Environments Division, U.S. Environmental Protection Agency.

Rohr, A.C. (2013). The health significance of gas- and particle-phase terpene oxidation products: A review, *Environment International* 60, 145-162.

Weschler, C.J. (2000). Ozone in indoor environments: Concentration and chemistry, *Indoor Air* 10, 269-288.

Weschler, C.J. (2006). Ozone's impact on public health: Contributions from indoor exposures to ozone and products of ozone-initiated chemistry, *Environmental Health Perspectives* 114, 1489-1496.

Weschler, .CJ. (2015). Roles of the human occupant in indoor chemistry, *Indoor Air*, doi:10.1111/ina.12185.

Weschler CJ, Shields HC (2003) Experiments probing the influence of air exchange rates on secondary organic aerosols derived from indoor chemistry, *Atmospheric Environment* **37**, 5621-5631.

Youssefi, S., Waring, M.S. (2014). Transient secondary organic aerosol formation from limonene ozonolysis in indoor environments: Impacts of air exchange rates and initial concentration ratios, *Environmental Science & Technology* 48, 7899-7908.

Zuraimi, M.S., Weschler, C.J., Tham, K.W., Fadeyi, M.O. (2007). The impact of building recirculation rates on secondary organic aerosols generated by indoor chemistry, *Atmospheric Environment* 41, 5213-5223.

ASSESSMENT OF VENTILATION SYSTEMS EFFICIENCY IN OFFICE BUILDINGS, RELATED TO INDOOR VOLATILE ORGANIC COMPOUND CONCENTRATIONS

Cătălin Teodosiu*, Viorel Ilie, and Raluca Teodosiu

Technical University of Civil Engineering Bucharest
Lacul Tei Bvd., no. 122-124
Bucharest 020396, Romania
**Corresponding author: cteodosiu@yahoo.com*

ABSTRACT

The objective of this study is to develop an approach concerning the integration of volatile organic compounds (VOCs) emissions due to office equipment in computational fluid dynamics (CFD) simulations, in order to assess the indoor air quality (IAQ) in offices. The transport and diffusion phenomena of VOCs are taking into account in the CFD model by means of conservation equations of the mass fraction, written for each VOC that is intended to be considered in the simulation. These equations are added to the basic equations describing turbulent confined non-isothermal flows (conservation of mass, momentum, energy, and turbulent quantities) in CFD modelling. On the other hand, these equations should include source terms of mass for each VOC that is planned to be taken into account in the model. The values of these source terms are specified in the model according to experimental data available in the literature. The numerical model is applied in this study for a small office, taking into account two ventilation systems: conventional mixing ventilation system and displacement ventilation system. The simulations are carried out for low air flow rates with different air supply temperatures. Concerning the sources of VOCs in terms of electronic devices, the office is supposed to be equipped with 4 computers, 4 monitors, and 4 laser printers. The assessment of IAQ in the office is accomplished by taking into account in the CFD model the indoor levels of the following five VOCs: benzaldehyde, ethylbenzene, o-xylene, styrene, and toluene. The CFD model proposed in this study allows achieving values of VOCs concentrations throughout the entire indoor environment in a particularly comprehensive manner. Consequently, results are presented in terms of benzaldehyde, ethylbenzene, o-xylene, styrene, and toluene concentration contours in the office. The results show that the estimated VOCs concentration levels due to office equipment are far below the set threshold limit values, both in the case of mixing ventilation and displacement ventilation, despite low ventilation rates taken into account. However, it should be remembered that the results are based only on VOCs emissions due to electronic devices, VOCs generated from other indoor sources have not been taken into consideration during the simulations. On the other hand, the numerical description of VOCs sources for CFD modelling developed in this work may be easily extended for other indoor VOCs sources. Finally, the numerical approach proposed in this study can lead to relevant IAQ analyses, being an appropriate alternative to experimental investigations, challenging to carry out in situ.

KEYWORDS

IAQ - indoor air quality; VOCs – volatile organic compounds; ventilation efficiency; CFD – Computational Fluid Dynamics modelling

1 INTRODUCTION

Indoor air quality (IAQ) is more and more one of the most important issues facing human civilisation nowadays. This is due to the following main factors: people in developed countries spend approximately 90% of the time indoors, 5% in different means of transportation (cars, trains, airplanes, etc.), and only 5% outdoors (Hancock, 2002); building ventilation rates reach often values below nominal ventilation rate conditions established according to standards for healthy indoor environments (Mendell et al., 2013); there are more and more sources of indoor air pollution due to our modern style of life (Zhang and Smith, 2003).
The best example related to the last point mentioned above is the dramatic growth of volatile organic compounds (VOCs) in classic indoor air samples because of new sources each year: from hundreds species in 1990s to thousands individual VOCs nowadays (Manivanan, 2006).
In this context, office devices are becoming major sources of indoor pollution due to their widespread and increasing use in modern, well-equipped, office buildings. Studies have shown that the emissions from these devices (computers, monitors, printers, copiers, etc.) generate about 35 substances and most of these are VOCs (Maddalena et al., 2011).
Unfortunately, numerous recent studies (McGee, 2015) have been undoubtedly revealed that human exposure to VOCs can lead to acute effects (e.g. inflammation of mucous membranes, irritation, headache, asthma exacerbation) and even chronic effects (e.g. respiratory system, circulatory system, and central nervous system diseases, organs dysfunction, cancer).
Consequently, the number of studies dealing with IAQ associated to VOCs emissions and indoor concentrations is continuously increasing, particularly for office work environment, as many people in developed countries spend large part of the day in office buildings, frequently improperly ventilated. Most of these investigations are based on experimental methodologies (Wang et al., 2011; Kowalska et al., 2015). However, experimental studies in this field are challenging to fulfil in situ. As a result, there are also a number of numerical approaches used to evaluate the IAQ, including the Computational Fluid Dynamics (CFD) technique. Indeed, thanks to amazing hardware development, the CFD approach can be employed nowadays as a pertinent simulation tool to assess the IAQ for different applications (Helmis et al., 2007; Stathopoulou and Assimakopoulos, 2008; Corgnati and Perino, 2013; Zhuang et al., 2014; Yang et al., 2014).
Consequently, the objective of this work is to numerically predict the IAQ within small ventilated rooms in office buildings (based on CFD technique), taking into account VOCs emissions from office equipment.
Results are presented in terms of VOCs concentrations contours in the office, for different air supply temperatures in the case of mixing ventilation and displacement ventilation. This allows to compare the performance (efficiency) of different ventilation strategies for the enclosure taken into consideration.

2 MODELLING OF VOLATILE ORGANIC COMPOUND CONCENTRATIONS

The development of the numerical model is based on VOCs transport and diffusion in the indoor air in order to assess the efficiency of the ventilation systems. As a result, the main hypotheses of the CFD model are the following:

- fluid, air-VOCs mixture;
- mixture air-VOCs, ideal gas of perfect gases (air and VOCs);

- mixture air-VOCs, incompressible Newtonian fluid;
- no chemical reaction between the species of the mixture;
- insignificant heat and mass transfer interactions within the mixture;
- density of mixture air-VOCs, ideal gas law formulation (based on the mixture temperature and the mass fraction of each species (air and VOCs);
- specific heat capacity of mixture air-VOCs, mixing law formulation (depending on the mass fraction average of the species, air and VOCs, heat capacities;
- thermal conductivity and viscosity of mixture air-VOCs, expressed through kinetic theory;
- diffusion coefficient of VOCs in air, constant values, based on data extracted from the literature.

Based on the above assumptions, the transport and diffusion phenomena of VOCs are taking into account in the CFD model by means of conservation equations of the mass fraction, written for each VOC that is intended to be considered in the simulation. These equations can be written as classic convection-diffusion equations. In addition, these equations are added to the equations describing turbulent confined non-isothermal flows (conservation of mass, momentum, energy, and turbulent quantities) in CFD modelling. As a result, VOCs conservation equations and the other equations, governing in the CFD model the conservation of variables in the computational domain, have similar expression:

$$0 = -\nabla \cdot \left(\rho \phi \vec{v} \right) + \nabla \cdot \left(\Gamma \nabla \phi \right) + S \tag{1}$$

where ρ – mixture density; Φ - variable of interest (it can be: mass fraction for each VOC considered, 1 for the continuity, velocity components, temperature, or turbulent parameters); v – mixture velocity; Γ - diffusion coefficient; S – source term.

The first term on the right-hand side of the Eq. (1) stands for the convection mechanisms and, in the case of VOCs, it is representing the change of the VOC concentration due to the flow. On the other hand, the second term on the right-hand side of the Eq. (1) signifies the diffusion term. It is worthwhile to mention that this term integrates in the numerical model both aspects of diffusion, molecular and turbulent. The molecular diffusion is based on Fick's first law: diffusion flux is proportional to the concentration gradient, using constant diffusion coefficient in the case of each VOC, as mentioned above. On the other hand, the turbulent diffusion for each VOC is taking into account in a similar way to that of the Reynolds analogy. Consequently, mixture mass turbulent diffusivity is related to the eddy viscosity through the CFD turbulence model used to describe the flow in the computational domain. Finally, the third term on the right-hand side of the Eq. (1) refers to source terms. The values of these source terms for each VOC are based on VOC generation rates of different sources which are intended to be considered in the numerical model. Consequently, these source terms for each VOC will be specified in the model according to experimental data available in the literature (see the next section).

3 CASE STUDY

The numerical model briefly presented above is applied in this study for a small office (6.2 x 3.1 x 2.5 m^3), taking into account two ventilation systems:

- classic mixing ventilation system, air supply occurs at the upper part of the room, while air exhaust is located at the lower part of the room (Figure 1);
- displacement ventilation system, in this case the air supply is located in the occupied zone, while air exhaust is placed at the upper part of the room (Figure 2).

The simulations are carried out for low air flow rates with different air supply temperatures, both in the case of mixing ventilation and displacement ventilation, see Table 1.

Table 1: Case Study Conditions (Air Inlet Characteristics)

Test	Air flow rate (m^3/h)	Air changes per hour (h^{-1})	Temperature (°C)
Mixing ventilation	96.1	2.0	15.0
Mixing ventilation	96.1	2.0	30.0
Displacement ventilation	96.1	2.0	15.0
Displacement ventilation	96.1	2.0	30.0

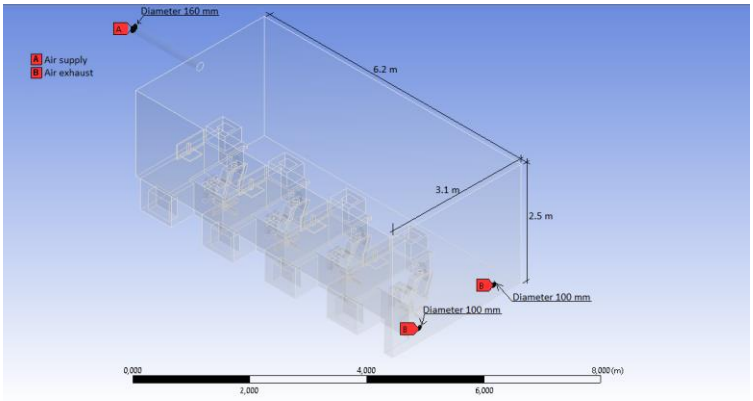


Figure 1: Mixing ventilation system

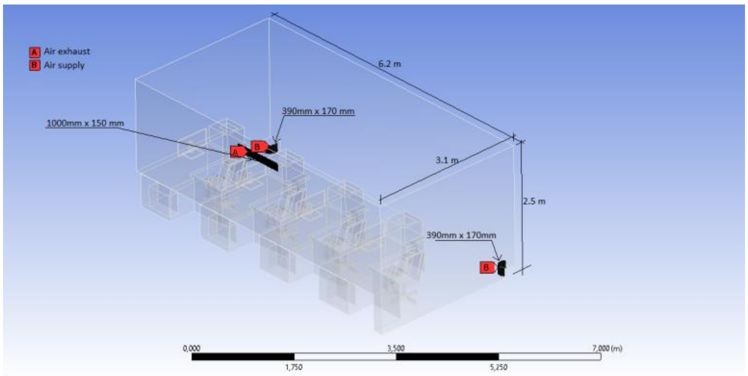


Figure 2: Displacement ventilation system

Concerning the sources of VOCs in terms of electronic devices, the office is supposed to be equipped with 4 computers, 4 monitors, and 4 laser printers (see Figure 3).

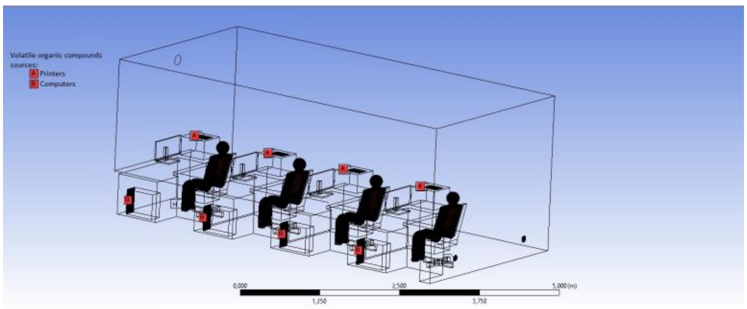


Figure 3: VOCs sources in the office

The assessment of IAQ in the office is accomplished by taking into account in the CFD model the indoor levels of the following five VOCs: benzaldehyde, ethylbenzene, o-xylene, styrene, and toluene. The choice of these VOCs is justified by the fact that these VOCs are characterised by significant emissions from office equipment, according to data available in the literature (Maddalena et al., 2011; Kowalska et al., 2015). In addition, it is worthwhile to mention that these five VOCs are classified as possibly carcinogenic to humans.

Consequently, the implementation of the CFD model requires in this case five supplementary equations expressing the conservation of mass fraction for each VOC taken into consideration (benzaldehyde, ethylbenzene, o-xylene, styrene, and toluene). As indicated before, these equations have the classic formulation of advection-diffusion equations, see Eq. (1), and they are added to the basic equations expressing turbulent confined non-isothermal flows in CFD modelling. The values of the diffusion coefficient of each VOC, required in these conservation equations for each VOC, are specified in Table 2 (New Jersey, 2015).

Table 2: VOC, Diffusion Coefficients in Air

VOC	**benzaldehyde**	**ethylbenzene**	**o-xylene**	**styrene**	**toluene**
diffusion coefficient x 10^{-6} (m^2/s)	7.3	7.5	7.5	7.1	8.7

On the other hand, the emissions of VOCs from office equipment are taken into account by mass source terms added in each of these five equations describing the conservation of mass fraction, for each VOC considered (benzaldehyde, ethylbenzene, o-xylene, styrene, and toluene). The values of these source terms for each of the five VOCs considered are specified in the model according to experimental data existing in the literature (Maddalena et al., 2011), see Table 3.

Table 3: VOC, Source Terms for Office Equipment

VOC	**benzaldehyde**	**ethylbenzene**	**o-xylene**	**styrene**	**toluene**
computer + monitor source term x 10^{-11} (kg/s)	0.209	1.414	0.930	0.921	2.056
laser printer source term x 10^{-11} (kg/s)	2.367	1.956	1.625	2.220	1.550

The numerical model developed as specified above was assembled using a finite-volume, Navier-Stokes solver (Fluent, version 15.0.0). The main elements of this model are shown in Table 4.

Table 4: CFD Model

Feature	**Description**
Fluid	air-VOCs mixture
Flow	three-dimensional, steady, non-isothermal, turbulent
Computational domain discretisation	finite volumes, unstructured mesh (tetrahedral elements): 5,744,884 cells for mixing ventilation; 6,007,026 cells for displacement ventilation
Turbulence model	Shear Stress Transport (SST) turbulent kinetic energy-specific turbulent dissipation rate (k-ω), with low-Reynolds corrections
Numerical resolution	second-order upwind scheme velocity-pressure coupling: SIMPLE algorithm convergence acceleration: algebraic multigrid

4 RESULTS

The numerical model allows achieving VOCs concentration field all over the computational domain. Consequently, results are presented below in terms of benzaldehyde, ethylbenzene, o-xylene, styrene, and toluene concentration contours in the office, for each configuration taken into account (see Table 1). These results are reported in a vertical plane passing nearby the occupants. Figures 4 and 5 show the results for the cold air supply test and hot air supply test in the case of mixing ventilation system. The same results are presented in Figures 6 and 7 for the displacement ventilation system.
It should be first emphasized that, despite the numerous indoor VOCs sources taken into account, limited dilution volume and low ventilation rates, for all configurations taken into consideration, VOCs concentration levels in the office are well below the occupational

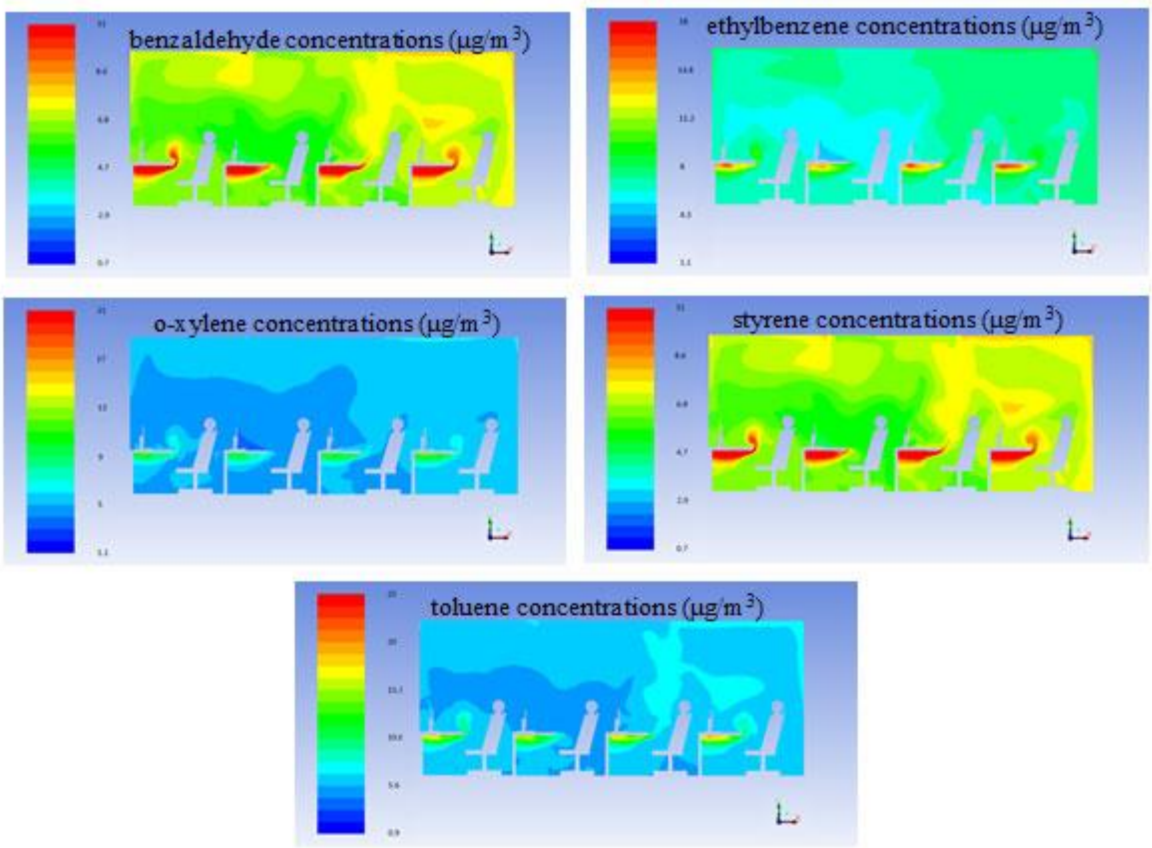


Figure 4: VOCs concentrations (mixing ventilation system, cold air supply, 2 h^{-1} air changes per hour)

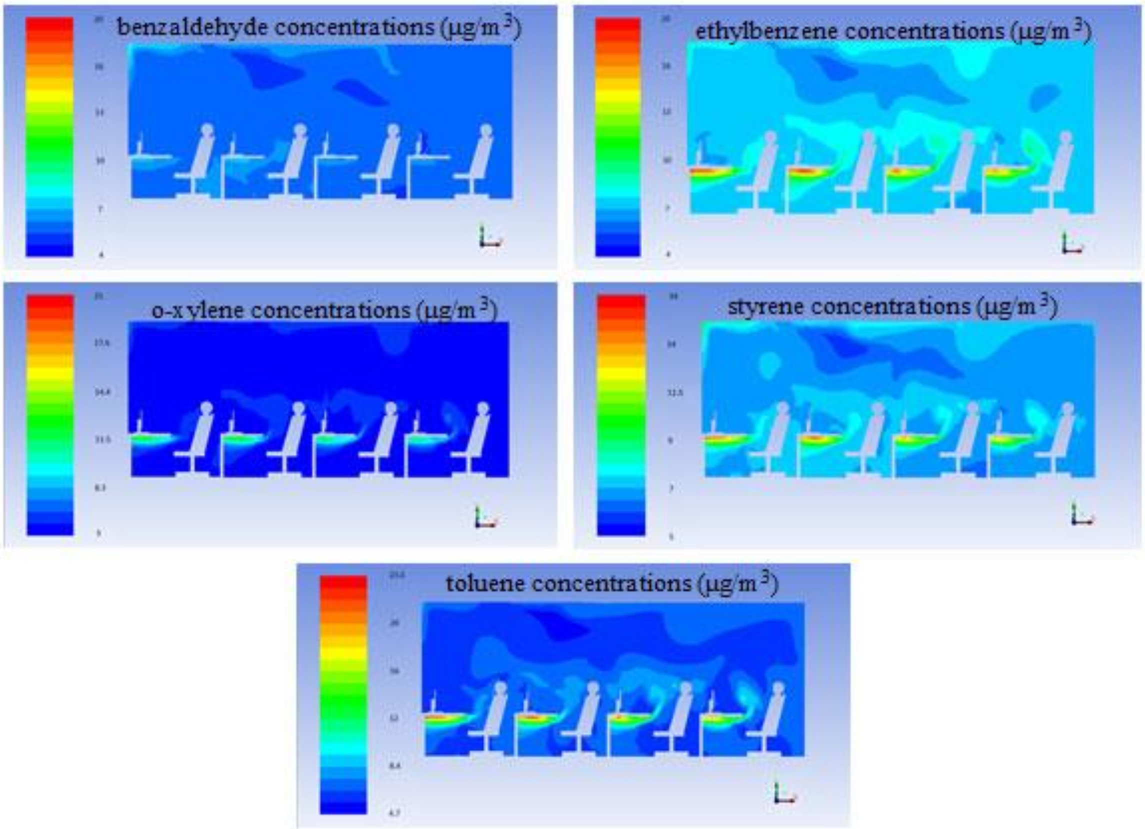


Figure 5: VOCs concentrations (mixing ventilation system, hot air supply, 2 h^{-1} air changes per hour)

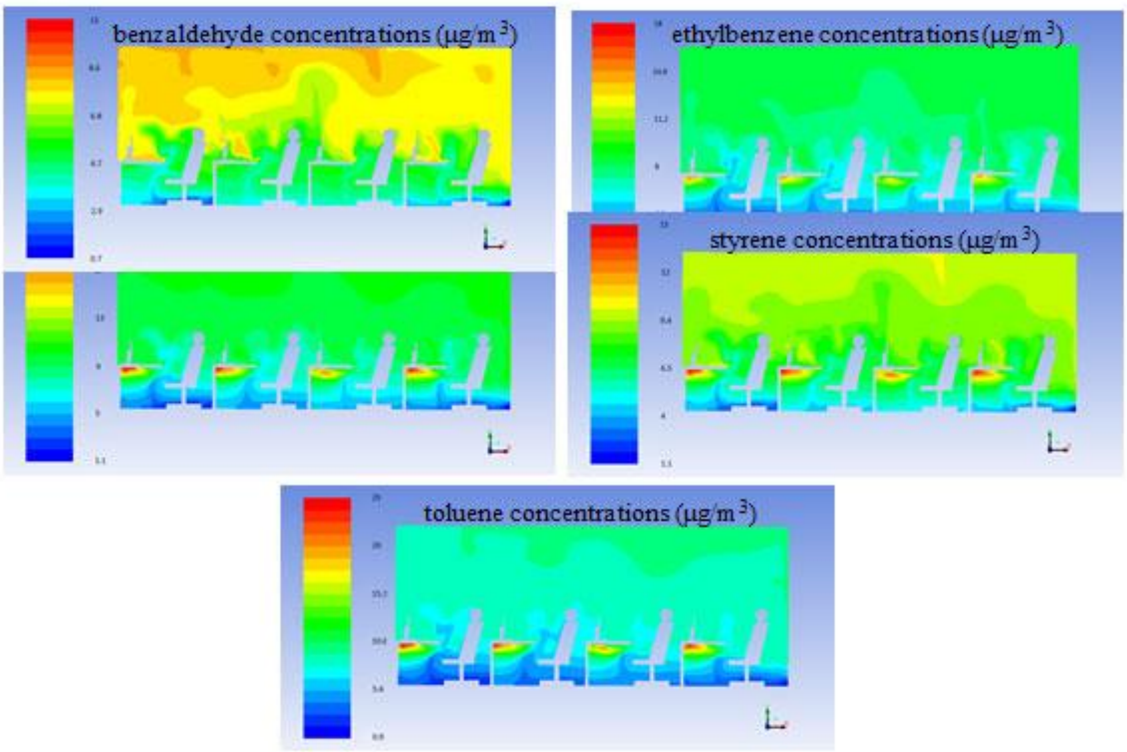


Figure 6: VOCs concentrations (displacement ventilation system, cold air supply, 2 h^{-1} air changes per hour)

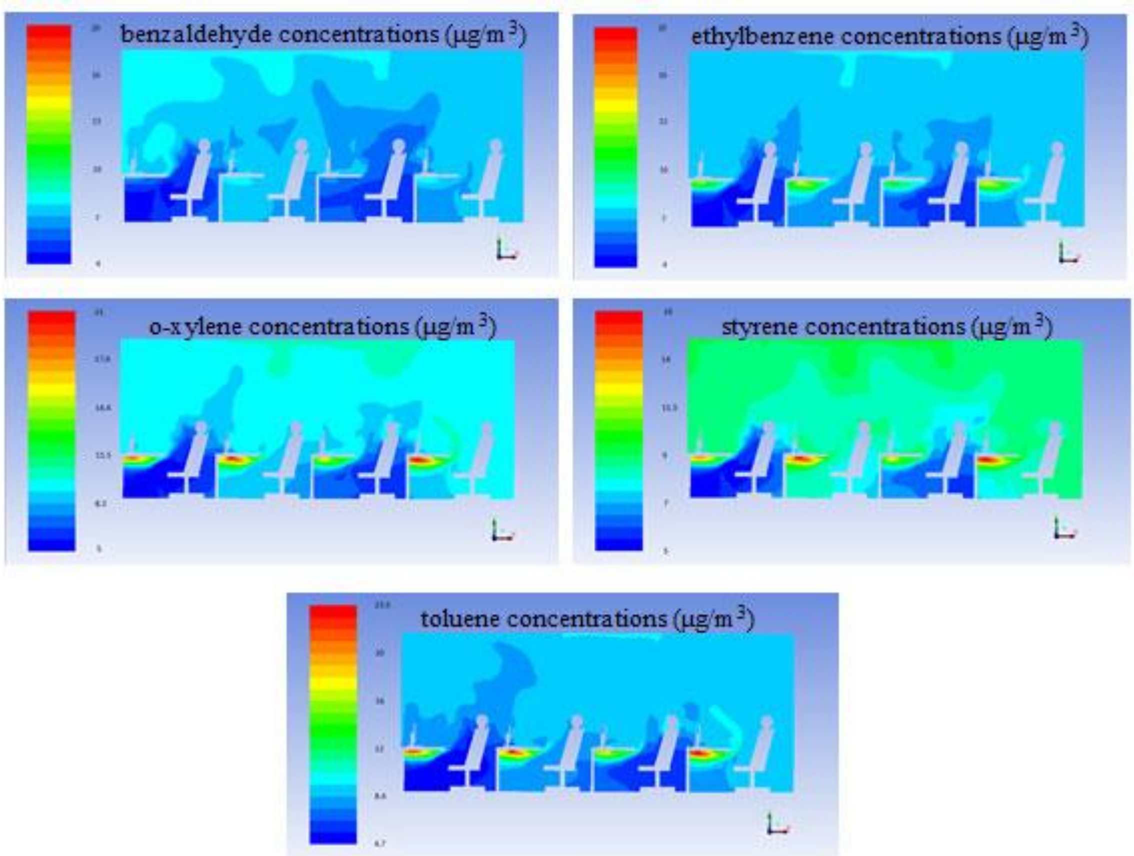


Figure 7: VOCs concentrations (displacement ventilation system, hot air supply, 2 h^{-1} air changes per hour)

exposure limits established by different organisms: Permissible Exposure Limits – PELs, set by Occupational Safety and Health Administration – OSHA (California, 2015) or Threshold Limit Values – TLVs set by American Conference of Government Industrial Hygienists – ACGIH (Abdul-Wahab, 2011). These results confirm the experimental data summarised in (Levin and Hodgson, 2006).

On the other hand, there are differences in effectiveness between the two ventilation systems, as expected. Accordingly, the numerical results highlight the well-known behaviour of mixing ventilation systems and displacement ventilation systems. In the case of mixing ventilation, efficiency is different depending on the temperature of the air supply and room area: the efficiency can be decent in the case of cold air supply in the region where the fresh air jet falls in the occupied zone, while the VOCs concentrations can reach values twice as high in other regions of the occupied zone (Figure 4). In addition, the mixing ventilation efficiency is affected in the entire occupied zone in the case of hot air supply due to buoyant forces and Coanda effect even if the indoor levels of VOCs remain low (Figure 5).

Concerning the displacement ventilation, the overall effectiveness is constant in the occupied zone, regardless of the supply air temperature. Basically, in this case, the diffusion of the pollutants follows the air flow established in the enclosure.

5 CONCLUSIONS

The implementation of the CFD model proposed in this study allows achieving values of VOCs concentrations throughout the entire indoor environment. As a result, this methodology can lead to pertinent IAQ investigations, being an appropriate option to experimental analyses, difficult to perform in situ. In addition, the numerical model allows forecasting indoor VOCs values in a particularly comprehensive way. This makes possible to assess the

levels of VOCs near the occupants, allowing more accurate estimations regarding the IAQ, contrary to analyses based on average values or on a small number of measuring points in the occupied zone.
On the other hand, the numerical description of VOCs sources for CFD modelling developed in this work may be extended for other indoor VOCs sources (e.g. carpets, furniture, paints, building materials, etc.).
The results of the case study taken into account show that the estimated VOCs concentration levels due to office equipment are far below the established threshold limit values, despite low ventilation rates taken into account. On the other hand, it should be remembered that the results are based only on VOCs emissions due to electronic devices, VOCs generated from other indoor sources have not been taken into consideration during the simulations. Furthermore, in spite of low concentrations of indoor VOCs from electronic devices numerically assessed in this study, long term exposure could be more harmful to the health than short term exposure to higher concentrations. Besides, cumulative chemical effects of multiple VOCs may lead to more detrimental impact on occupants' health.
Consequently, investigations based on numerical model presented in this work should be continued to make available new data concerning IAQ related to VOCs emissions from indoor sources.

6 ACKNOWLEDGEMENTS

This work was supported by a grant of the Romanian Ministry of Education and Research, CNCS-UEFISCDI, project number PN-II-ID-JRP-RO-FR-2012-0071.

7 REFERENCES

Abdul-Wahab, S.A. (2011). *Sick Building Syndrome in Public Buildings and Workplaces*. London: Springer.

California (2015). *Permissible Exposure Limits for Chemical Contaminants (State of California – Department of Industrial Relations).* Retrieved June 15, 2015 from https://www.dir.ca.gov/title8/5155table_ac1.html

Corgnati, S.P., and Perino, M. (2013). CFD application to optimise the ventilation strategy of Senate Room at Palazzo Madama in Turin (Italy). *Journal of Cultural Heritage*, 14(1), 62-69.

Hancock, T. (2002). *Built Environment (Encyclopedia of Public Health).* Retrieved June 4, 2015 from http://www.encyclopedia.com/doc/1G2-3404000130.html

Helmis, C.G., Adam, E., Tzoutzas, J., Flocas, H.A., Halios, C.H., Stathopoulou, O.I., Assimakopoulos, V.D., Panis, V., Apostolatou, M., and Sgouros, G. (2007). Indoor air quality in a dentistry clinic. *Science of the Total Environment*, 377(2), 349-365.

Kowalska, J., Szewczyńska, M., and Pośniak, M. (2015). Measurements of chlorinated volatile organic compounds emitted from office printers and photocopiers. *Environmental Science and Pollution Research*, 22(7), 5241-5252.

Levin, H., and Hodgson, A.T. (2006). VOC Concentrations of Interest in North American Offices and Homes. In: Proceedings of Healthy Buildings 2006, Lisbon, Portugal, vol. 3, pp. 233-238.

Maddalena, R., McKone, T., Destaillats, H., Rusell, M., Hodgson, A., and Perino, C. (2011). *Quantifying Pollutant Emissions from Office Equipment: A Concern in Energy-Efficient Buildings*. California Energy Commission, PIER Energy-Related Environmental Research. CEC-500-2011-046.

Manivanan, R. (2011). *Recycling of Industrial Effluents*. New Delhi: New India Publishing Agency.

McGee, A. (2015). *Potential Health Effects from Exposure to VOC Emissions (Institute of Neurotoxicology and Neurological Disorders)*. Retrieved June 4, 2015 from http://www.toxipedia.org/display/FOC/Potential+Health+Effects+from+Exposure+to+VOC+Emissions

Mendell, M.J., Eliseeva, E.A., Davies, M.M., Spears, M., Lobscheid, A., Fisk, W.J., and Apte, M.G. (2013). Association of classroom ventilation with reduced illness absence: a prospective study in California elementary schools. *Indoor Air*, 23(6), 515-528.

New Jersey (2015). *Chemical Properties for Calculation of Impact to Ground Water Soil Remediation Standards (State of New Jersey – Department of Environmental Protection).* Retrieved June 10, 2015 from http://www.nj.gov/dep/srp/guidance/rs/chemproperties.pdf

Stathopoulou, O.I., and Assimakopoulos, V.D. (2008). Numerical Study of the Indoor Environmental Conditions of a Large Athletic Hall Using the CFD Code PHOENICS. *Environmental Modeling & Assessment*, 13(3), 449-458.
Wang, Z.M., Wagner, J., and Wall, S. (2011). Characterization of Laser Printer Nanoparticle and VOC Emissions, Formation Mechanisms, and Strategies to Reduce Airborne Exposures. *Aerosol Science and Technology*, 45(9), 1060-1068.

Yang, L., Ye, M. and He, B.J. (2014). CFD simulation research on residential indoor air quality. *Science of the Total Environment*, 472, 1137-1144.

Zhang, J., and Smith, K.R. (2003). Indoor air pollution: a global health concern. *British Medical Bulletin*, 68(1), 209-225.

Zhuang, R., Li, X. and Tu, J. (2014). CFD study of the effects of furniture layout on indoor air quality under typical office ventilation schemes. *Building Simulation*, 7(3), 263-275.

INFLUENCE OF MOVING OBJECTS ON VENTILATION PLAN FOR SMOKING ROOM

Sihwan LEE*[1], Takashi KURABUCHI[1]

1 Department of Architecture, Tokyo University of Science
6-3-1, Niijuku, Katsushika-ku
125-8585, Tokyo, Japan
Corresponding author: shany@rs.tus.ac.jp

ABSTRACT

Ventilation plan for smoking room must deal with pollutants since they affect the air quality of adjacent rooms. Although ventilation plan typically maintains a negative room pressure to remedy this problem, the transport of indoor air pollutants between rooms is affected by moving objects, such as human movement and door opening. The purposes of this study were to evaluate the effects of moving objects on the rate of transport of indoor air pollutants and to propose a method of controlling contamination for smoking room.
First, we measure the inter-zonal air exchange rate by uniformly dispersing sulfur hexafluoride (SF_6) as a tracer gas with swinging of the door between an air-contaminated room and corridor. We also measure the direction and velocity of air flow to ascertain variations in air flow around the door due to door opening and closing. Results of these measurements were compared to results of Computational Fluid Dynamics (CFD) analysis with overset mesh technics to verify that CFD analysis was consistent with actual findings. Then, we evaluated the inter-zonal air exchange rate by influence of moving objects with isothermal/non-isothermal conditions. In addition, we evaluate the several ventilation planning to decrease pollutants leakage from the smoking room, such as installation of sliding door or air curtain system.
The measured absolute rate of inter-zonal air exchange as a result of opening and closing the door once differed little from the result of CFD analysis, so the results roughly coincided. Although the air curtain operation was ineffective to decrease the inter-zonal air exchange rate when the temperature differential between two adjacent spaces is little, the air curtain operation was predictably effective when the temperature differential between two adjacent spaces is large. Moreover, use of a sliding door in smoking room with exhaust ventilation system limited the inter-zonal air exchange rate more than use of a swinging door. However, the inter-zonal air exchange rate increased when human movement was combined with use of a sliding door.

KEYWORDS

Ventilation, Air exchange rate, Moving object, Smoking room, Computational fluid dynamics

1 INTRODUCTION

Ventilation plans for rooms with contaminated air, such as laboratories, smoking rooms, and bathrooms, must deal with pollutants since they affect the air quality of adjacent rooms. A ventilation plan typically maintains a negative room pressure to remedy this problem. However, the transport of indoor air pollutants between rooms is affected by moving objects, such as human movement and door opening, despite planned ventilation. Although there have been many studies evaluating the performance of such rooms with negative room pressure [N. Rice et al. (2001)], there have been relatively fewer studies assessing how door opening motions and human passage through the door [Adams NJ et al. (2011), Petri Kalliomäki et al. (2013), Julian W. Tang et al. (2013)], and the effect of air curtain system [J.C. Goncalves et al. (2012), D. Frank et al. (2015)]. The purposes of this study were to evaluate the effects of moving objects on the rate of transport of indoor air pollutants and to propose a method of controlling contamination in an indoor environment.

2 METHODOLOGY

2.1 Measurement and verification of CFD analysis

The space studied is shown in Fig. 1. Air flow in this space was visualized, the direction and velocity of air flow were measured, and the inter-zonal air exchange rate was measured. Results of these measurements were compared to results of CFD analysis of the same space to verify that CFD analysis was consistent with actual findings. Air flow was visualized by generating smoke in the room with a water and glycol-based smoke generator, and then visualizing air flow around the door during door opening and closing (height of Nd:YV04 laser beam : 1.2 from the floor). The direction and velocity of air flow were measured to ascertain variations in air flow around the door due to door opening and closing using 3D ultrasonic anemometer. Measurement was done 0.1 m (h = 1.2 m) from the edge of the door (projecting into the corridor) when the door was fully open. The inter-zonal air exchange rate was measured by uniformly dispersing sulfur hexafluoride (SF_6) as a tracer gas in the room. The door was closed 0 times/h (natural ventilation), 6 times/h, 12 times/h, and 60 times/h. The inter-zonal air exchange rate (Eq. 1) was calculated based on the concentration decay as a result of the door being closed a certain number of times. All of the measurement units are shown in Table 1.

$$Q = \frac{V}{\Delta t} \cdot \ln \frac{C_S}{C} \tag{1}$$

Here, Q [m^3/h] is the inter-zonal air exchange rate, C_s [kg/m^3] is the initial concentration of contaminants, C [kg/m^3] is the concentration of those pollutants in the room after time t [h], and V [m^3] is the room volume.

Boundary conditions for CFD analysis are shown in Table 2. The turbulence model used was a high Reynolds number k-epsilon model, and the differencing scheme used was a MUSCL scheme. To realize door opening motion, CFD analysis was considered the calculation of time-dependent and overset mesh technique. The initial concentration of contaminant was also set to 1 in indoor and 0 in corridor to calculate the inter-zonal air exchange rates by CFD analysis. The interval for door opening and closing was set to 0~4 sec (door opening) → 4~6 sec (door opened) → 6~10 sec (door closing) → 10~12 sec (door closed).

2.2 Evaluation of ways to decrease leakage of pollutants from a smoking room

Smoke from smoking area is affect non-smoking area by including human breathing, attached clothes, induced airflow by the human moving, and door opening/closing. Therefore there is need to install exhaust fan or air cleaner system, and there is need to maintain the acceptable concentration limit of carbon monoxide, carbon dioxide and particulate matter. In Japan, the design guideline for smoking room has proposed by Ministry of Health, Labour and Welfare, and it is recommended to maintain the air velocity more than 0.2m/s at the door of smoking area by installing an exhaust fan. However, the design guideline is non-binding standards and non-engineered recommendation. In this study, we evaluated the effects of air curtain and exhaust fan to decrease leakage of pollutants from a smoking room, and conducted quantitative analysis about the transport rates of pollutants.

(1) Air curtain effects under different thermal conditions

Air curtains are commonly used as virtual barriers in doorways separating two different thermal or contaminant environments. At the smoking room, an air curtain can considerably reduce the spreading of cigarette smoke, heat loss, and draughts between two adjacent spaces without impeding human traffic through the doorway. To evaluate the effects of air curtain at the smoking room, the extent of pollutant leakage due to the air flow from an air curtain was studied by CFD analysis.
The schematic of smoking room model is shown in Fig. 5. The volume of a smoking room is 15.625 cubic meters and the adjacent space is a corridor of 12.500 cubic meters. Details of the door and door louvers are also shown in Fig. 5, and the interval for door opening and closing was set to 0~2 sec (door opening) → 2~4 sec (door opened) → 4~6 sec (door closing). For this investigation, we also used a 6m/s vertically downwards blowing air curtain model. Boundary conditions and calculation cases for CFD analysis are shown in Tables 3, 4. The CFD analysis was calculated the pollutant leakage in air curtain operation and non-operation with temperature difference between smoking room and corridor. The temperature of smoking room was fixed to 20°C, and the temperature of corridor was set to 40°C from 0°C at 5°C intervals for investigation. Pollutant leakage was also calculated when the smoking room had a sliding door and a swinging door.

(2) Exhaust fan effects under isothermal conditions

The extent of pollutant leakage due to the air flow from an exhaust fan was studied in a smoking room in which negative room pressure was maintained (Fig. 5). Pollutant leakage was studied when the room had a sliding door and a swinging door. The calculation cases for CFD analysis are shown in Table 5. The steady state was calculated and then the unsteady state resulting from door movement was calculated. The inter-zonal air exchange rate was assessed for a smoking room with an air exchange rate of 5 times/h, 12 times/h, 30 times/h, and 77 times/h. The air exchange rate of 77 times/h is a value of the design guideline proposed as the air velocity 0.2m/s at the door of smoking room. In addition, human movement was also added to instances with an air exchange rate of all cases.

3 RESULTS

3.1 Verification of the results of CFD analysis

The visualized air flow at different times during door opening and closing and the results of CFD analysis are shown in Fig. 2. The standard k-ε turbulence model used in CFD analysis used the Reynolds average, so precise changes in turbulence characteristics over time could not be reproduced. However, flow characteristics around the door were similar to that actually visualized. Measurements of the flow rate variation in the x and y directions (the ensemble average determined based on 50 measurements) at the measurement site are shown in Fig. 3. These measurements were compared to the results of CFD analysis. Measurements and results of CFD analysis differed by 0.1 m/s or less. Measurements and CFD analysis yielded quite similar characteristics of flow rate variation over time as well. Concentration decay as a result of the number of times the door was opened or closed per hour was measured, as shown in Fig. 4(a). The absolute rate of inter-zonal air exchange minus the effects of the rate of natural air exchange was compared to the results of CFD analysis, as shown in Fig. 4(b). The number

of times the door was opened or closed per hour was proportional to the absolute rate of inter-zonal air exchange. The measured absolute rate of inter-zonal air exchange (0.49 m^3) as a result of opening and closing the door once differed little (0.06 m^3) from the result of CFD analysis (0.43 m^3), so the results roughly coincided.
Comparison of the 3 measurements mentioned and CFD analysis verified that measurements and results of CFD analysis differed little. Results of CFD analysis were highly consistent with actual measurements.

3.2 Evaluation of ways to decrease leakage of pollutants from a smoking room

(1) Evaluation results on air curtain effect

The results of CFD analysis with corridor temperature in air curtain non-operation are shown in Fig. 6, and the results of CFD analysis in air curtain operation are shown in Fig. 7. The results in air curtain non-operation show that the leakage rates of pollutant from smoking room was increased with increasing the difference of air temperature between smoking room and corridor because the drive force was increased by difference of air density. By contrast, the results in air curtain operation show that the leakage rates of pollutant was high than air curtain non-operation cases irrespective of the difference of air temperature because the turbulence diffusivity was increased due to the air flow from an air curtain.
The results of the cumulative inter-zonal air exchange rate in all calculation cases are shown in Fig. 8. Although using sliding door limited the inter-zonal air exchange rate more than using of swinging door under isothermal condition, it was totally ineffective under thermal condition. Moreover, although the air curtain operation was ineffective when the temperature differential between smoking room and corridor is little, the air curtain operation was predictably effective when the temperature differential is large. That is more effective under cooling mode than heating mode of smoking room.

(2) Evaluation results on exhaust fan effect

The results of CFD analysis with a sliding door and a swinging door, an air exchange rate of 5 times/h by exhaust fan, and no human movement are shown in Fig. 9, and the results of CFD analysis with an air exchange rate of 5 times/h and 77 times/h, and human movement are shown in Fig. 10. The sliding door resulted in less pollutant concentration leaking into the corridor than did the swinging door, but a large concentration of pollutants leaked when a person left the smoking room and entered the corridor. The cumulative effective air exchange in 6 sec with air change rates is shown in Fig. 11. Based on the calculated results, a sliding door resulted in a lower inter-zonal air exchange rate, and the small amount of pollutant leakage into the corridor was verified quantitatively. In addition, the effective air exchange rate for the smoking room approximated an exponential function with $Q_{eff}=0.493e^{-0.029n}$ when the room had a swinging door and $Q_{eff}=0.063e^{-0.023n}$ when it had a sliding door. Moreover, the inter-zonal air exchange rate increased when human movement was included, and the sliding door resulted in a greater rate of increase in the inter-zonal air exchange rate.

4 CONCLUSIONS

This study predicted the inter-zonal air exchange rate commensurate with door opening and closing and human movement, and this study examined ventilation to effectively prevent pollutant leakage from a smoking room. Based on the results, this study yielded the following conclusions:

(1) There was little difference between actual measurements and results of CFD analysis, so CFD analysis was verified to be consistent with actual findings.
(2) The air curtain operation was ineffective to decrease the inter-zonal air exchange rate when the temperature differential between two adjacent spaces is little, the air curtain operation was predictably effective when the temperature differential between two adjacent spaces is large. Moreover, it is more effective under cooling mode than heating mode.
(3) Use of a sliding door in a room with an exhaust ventilation system limited the inter-zonal air exchange rate more than use of a swinging door. However, the inter-zonal air exchange rate increased when human movement was combined with use of a sliding door.

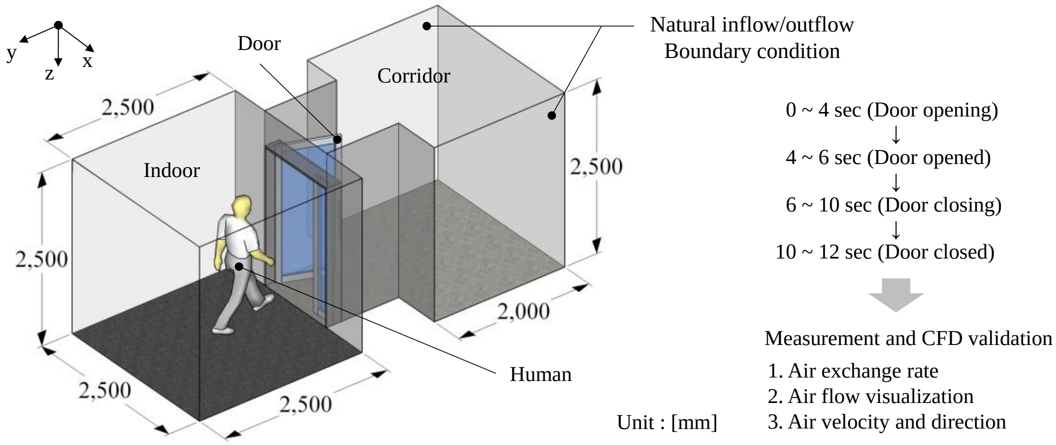


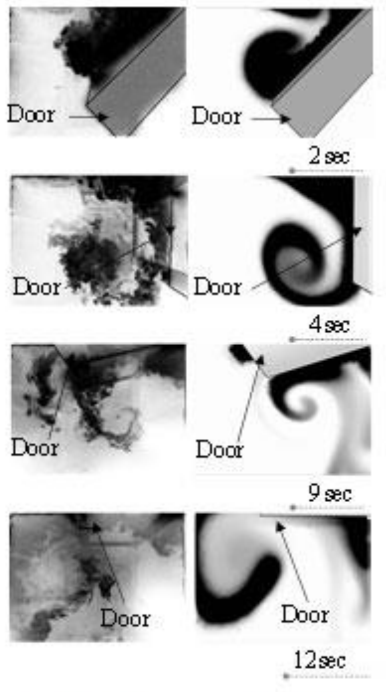
Figure 1: Schematic of calculation model

Table 1: Measurement units

Item	Contents
Air exchange rates	Multi-gas monitor (INNOVA 1312), Multipoint sampler (INNOVA 1303), SF_6
Air flow visualization	Water and glycol-based smoke generator, Nd:YV04 laser at 532 nm
Air velocity and direction	3D ultrasonic anemometer

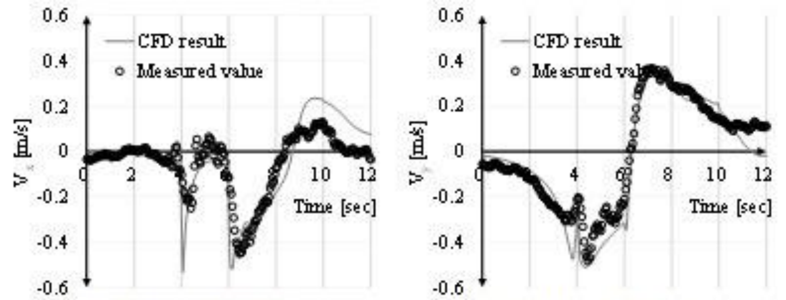
Table 2: Boundary condition for CFD analysis

Item	Contents
Indoor	2500(x) × 2500(y) × 2500(z) mm
Swing door	800(x) × 50(y) × 2100(z) mm (0.084 m^3, adiabatic) π/8 rotational motion (0~4sec : door opening, 6~10sec : door closing)
Turbulence model	High-Reynolds number k-epsilon model
Meshes (Moving object)	About 3,000,000 (Overset mesh technique)
Time dependent	Transient calculation (Courant number < 1)
Scheme	MUSCL scheme
Coefficient of mass diffusivity (SF_6)	1.6×10^{-5} m^2/s
Initial condition of SF_6 gas	C_s (indoor) = 1.0, C_s (corridor) = 0.0
Thermal condition	Isothermal condition



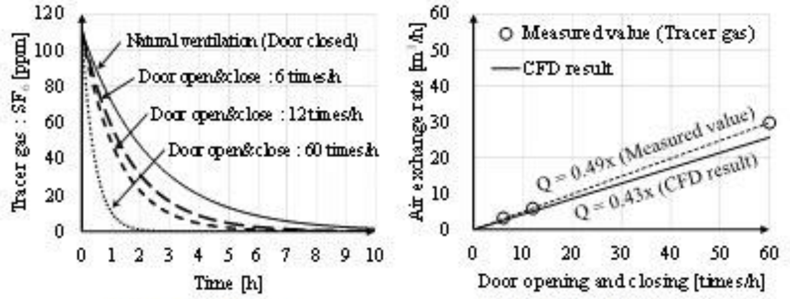
(a) measure (b) CFD

Figure 2: Airflow visualization



(a) air velocity in x-direction (b) air velocity in y-direction

Figure 3: Air velocity and air direction by 3D ultrasonic anemometer



(a) tracer gas decay (b) air exchange rate

Figure 4: Air exchange rate by tracer gas method

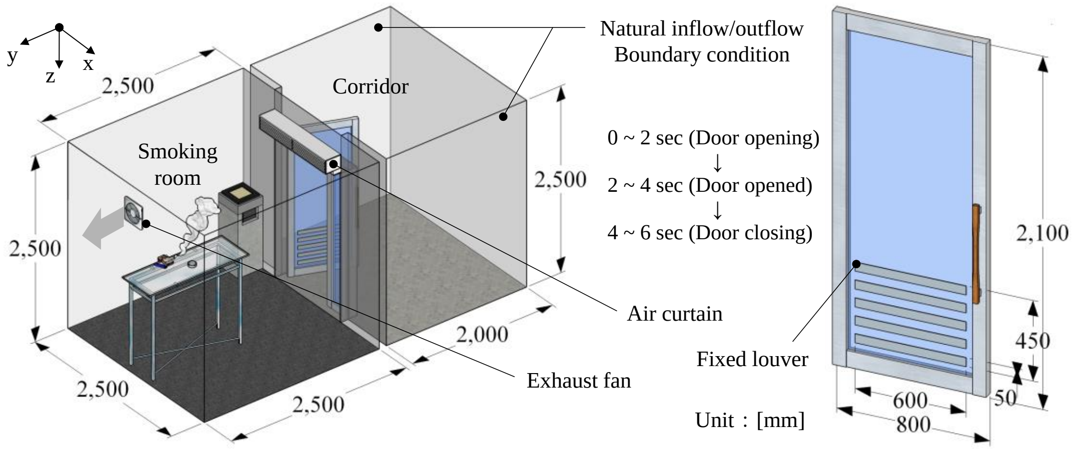


Figure 5: Schematic of smoking room model and detail of door

Table 3: Boundary condition

Item	Contents
Smoking room	2500(x) × 2500(y) × 2500(z) mm
Corridor	2500(x) × 2000(y) × 2500(z) mm
Human	690(x) × 27(y) × 1700(z) mm (0.053 m^3, adiabatic)
Door	800(x) × 50(y) × 2100(z) mm (0.084 m^3, adiabatic)
Fixed louver of door	600(x) × 50(z) mm (5 columns)
Exhaust Fan	300(y) × 300(z) mm
Air curtain (Inlet of air curtain)	800(x) × 200(y) × 200(z) mm (800(x) × 30(y) mm, v = 6.0 m/s)
Time dependent	Transient calculation after steady calculation

Table 4: Thermal calculation cases (36 cases)

Item	Contents
Smoking room temperature	20 ºC (fixed)
Corridor temperature	0, 5, 10, 15, 20, 25, 30, 35, 40 ºC (9 cases)
Door type	Swing door, Slide door (2 cases)
Air curtain operation	On / Off (2 cases)

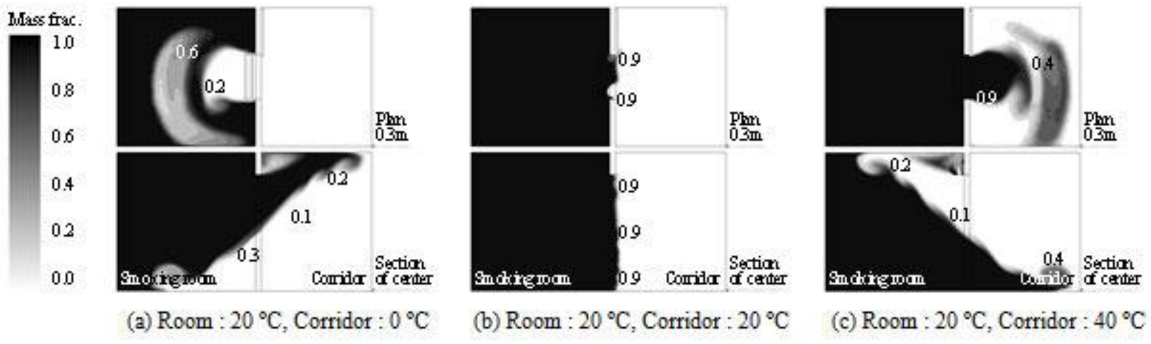


Figure 6. Calculated results with corridor temperature in air curtain non-operation (at 4 seconds)

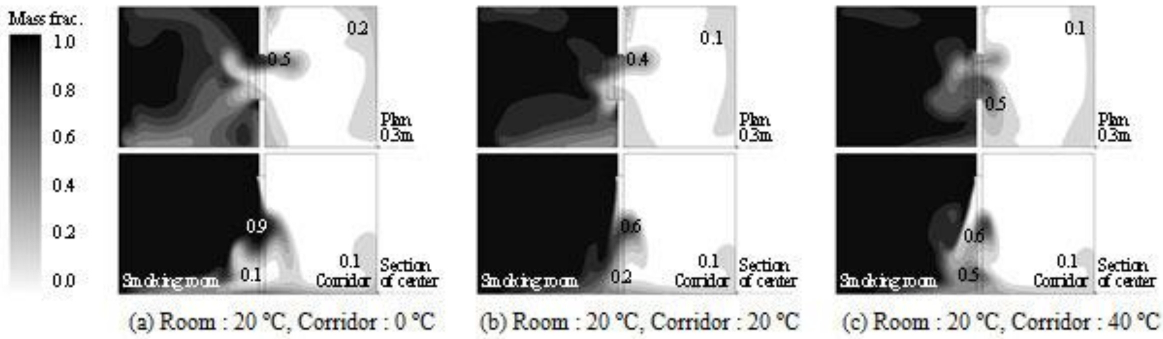


Figure 7. Calculated results with corridor temperature in air curtain operation (at 4 seconds)

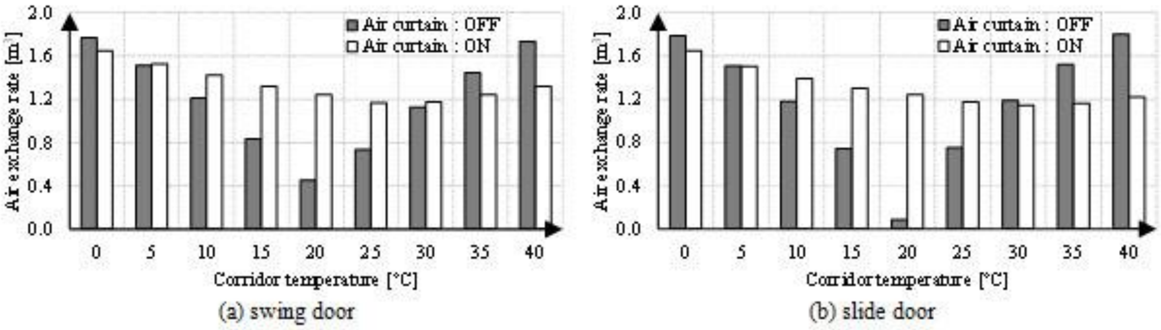


Figure 8. Calculated results of cumulative inter-zonal air exchange rate with corridor temperature

Table 5: Isothermal calculation cases (16 cases)

Item	Contents
Smoking room temperature	20 ºC (fixed)
Corridor temperature	20 ºC (fixed)
Air change rate (negative pressure)	5, 12, 30, 77 times/h (4 cases)
Door type	Swing door, Slide door (2 cases)
Human movement (1 m/s linear motion)	On / Off (2 cases)

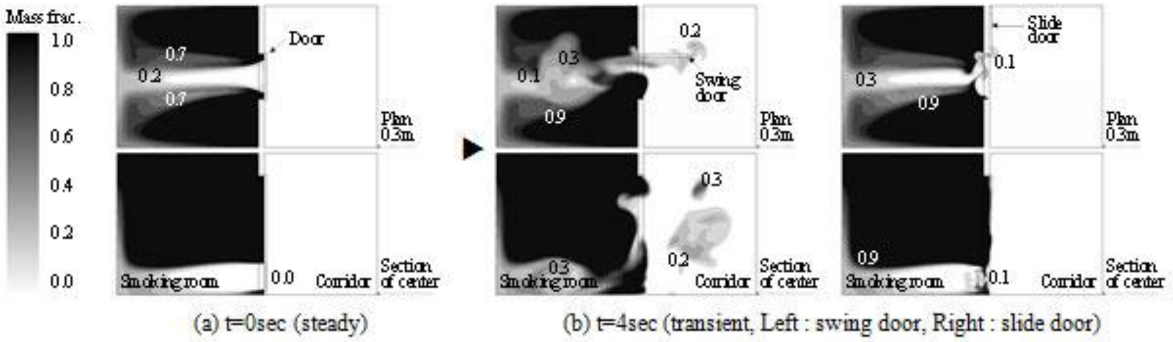


Figure 9. Calculated results of contaminant distribution with door type (5 times/h)

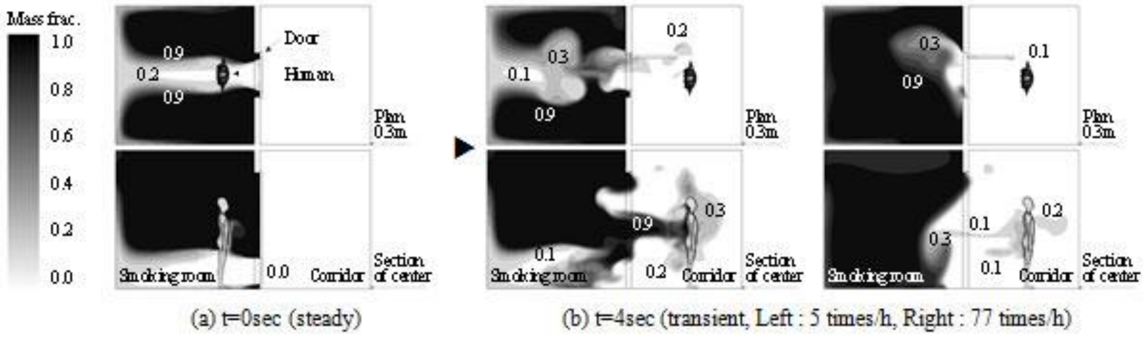


Figure 10. Calculated results of contaminant distribution with air change rate (swing door + human)

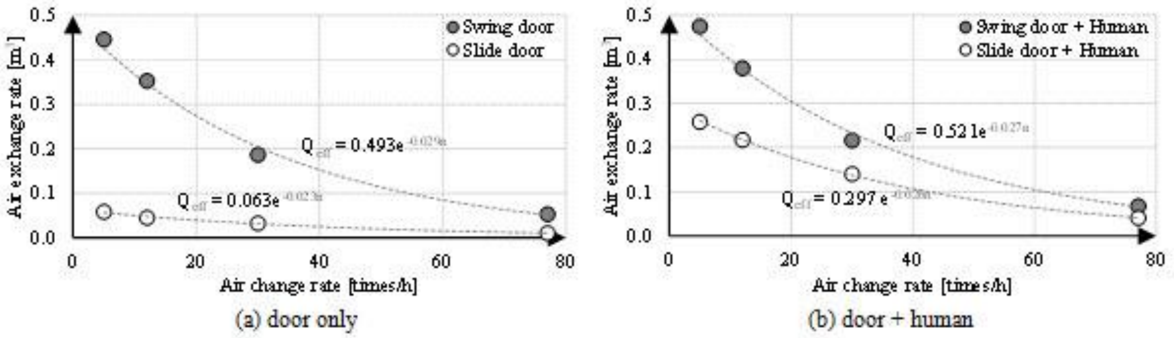


Figure 11. Calculated results of cumulative inter-zonal air exchange rate with air change rate

5 ACKNOWLEDGEMENTS

This study was made possible by the financial support received from Tokyo University of Science, Japan. We would like to thank students of Tokyo University of Science, for their generous cooperation.

6 REFERENCES

Nancy Rice, Andrew Streifel, and Donald Vesley (2001). *An evaluation of hospital special-ventilation room pressures*, Infection Control and Hospital Epidemiology Vol. 22, No. 01, pp.19-23.

Adams Noah J., David L. Johnson, and Robert A. Lynch (2011). *The effect of pressure differential and care provider movement on airborne infectious isolation room containment effectiveness,* American Journal of Infection Control 39(2), pp.91-97.

Petri Kalliomäki, et al., (2013). *Airflow patterns through a single hinged door in hospital isolation rooms*, CLIMA 2013.

Julian W. Tang, Andre Nicolle, Jovan Pantelic, Christian A. Klettner, Ruikun Su, Petri Kalliomaki, Pekka Saarinen, Hannu Koskela, Kari Reijula, Panu Mustakallio, David K. W. Cheong, Chandra Sekhar, and Kwok Wai Tham (2013). *Different types of door-opening motions as contributing factors to contaminant failures in hospital isolation rooms*, PLOS ONE, Volume 8, Issue 6.

J.C. Goncalves, J.J. Costa, A.R. Figueiredo, A.M.G. Lopes (2012). *CFD modelling of aerodynamic sealing by vertical and horizontal air curtains*, Energy and Buildings, Volume 52, September 2012, Pages 153-160.

D. Frank, P.F. Linden (2015). *The effects of an opposing buoyancy force on the performance of an air curtain in the doorway of a building*, Energy and Buildings, Volume 96, 1 June 2015, Pages 20-29.

MODEL HOME 2020 – FULL-YEAR MEASUREMENTS OF DAYLIGHT, ENERGY AND INDOOR CLIMATE IN FIVE SINGLE-FAMILY HOUSES OCCUPIED BY TYPICAL FAMILIES: WHAT HAS BEEN LEARNED

Peter Foldbjerg[*1], Thorbjørn Færing Asmussen[1], Christoffer Plesner[1], Jens Christoffersen[1]

1 VELUX A/S
Daylight Energy and Indoor Climate
Ådalsvej 99
2970 Hørsholm, Denmark
**Corresponding author: peter.foldbjerg@velux.com*

Note: the contact addresses may be re-arranged

ABSTRACT

This paper describes Post Occupancy Evaluation survey and physical measurements of five families living for one year or longer in five houses located in Germany, Austria, France and UK, all part of the Model Home 2020 project. The houses are built according to Active House principles and focus on high performance on indoor environmental quality, energy performance and environmental impact. The survey is carried out seasonally during the test year when the family lives in the house to capture seasonal variations. Physical measurements were made in all main rooms of the houses. The houses have high daylight levels, which is highly appreciated by the families. High daylight levels increase the risk of overheating, but this has been avoided by design and control as the families do not report overheating, and as the houses in general achieve category 1 for the summer situation according to the Active House specification. The families also indicate high satisfaction with the general indoor environment and the indoor air quality, better health, fewer sick days and improved sleep quality that their expectations often are fulfilled, and that house automation is acceptable. The physical measurements support the importance of building automation in order to achieve good performance.

KEYWORDS

Daylight, Thermal Comfort, Indoor air Quality, Dwellings, Post Occupancy Evaluation

1 INTRODUCTION

During 2009-2011, a demonstration project programme of five model homes were built in Denmark (Home for Life, HFL, 2009), Austria (Sunlighthouse, SLH, 2010), Germany (LichtAktiv Haus, LAH, 2010), France (Maison Air et Lumière, MAL, 2011) and United Kingdom (CarbonLight Homes, CLH, 2011). All houses are designed following the Active

House principles (Active House, 2011) with the three main elements: Comfort, Energy and Environment. The houses have been occupied by test families in periods of one year or longer and have been tested and monitored in use, under post occupancy evaluation schemes by national research teams of engineers and/or scientists (Feifer et al., 2014).

The Active House principles mean that a balanced priority of energy use, indoor environment and connection to the external environment must be made. The design has particularly focused on high performance of the indoor environment as well as on a very low energy demand. There is a particular focus on good daylight conditions and fresh air from natural ventilation. The thermal environment in the houses has previously been reported (Foldbjerg et al., 2014).

Use of natural ventilation for summer comfort is based on ventilative cooling principles, referring to the use of natural or mechanical ventilation strategies to cool indoor spaces. This effective use of outside air reduces the energy consumption of cooling systems while maintaining good thermal comfort (venticool, 2015). To ensure fresh air supply, the houses use natural ventilation in the warm part of the year and uses mechanical ventilation with heat recovery during cold periods. The exception is LichtAktiv Haus, which is a renovation project, using natural ventilation all year. There is external automatic solar shading on windows towards south and in most cases also towards east and west. Overhangs are used where appropriate.

2 METHODOLOGIES

2.1 Physical Measurements

Measurements of Indoor Environmental Quality (IEQ) include light, thermal conditions, indoor air quality, occupant presence and all occupant interactions with the building installations, including all operations of windows and solar shading. Each room is an individual zone in the control system, and each room is controlled individually. There are sensors for humidity, temperature, CO_2, presence and lux in all main rooms, used for both control and data recording. The building occupants can override the automatic controls, including ventilation and solar shading at any time. The recorded temperature data is evaluated according to the Active House specification (Active House, 2011), which is based on the adaptive approach of EN 15251 (CEN, 2007).

2.2 Post Occupancy Evaluation Survey

As part of the evaluation, a Post Occupancy Evaluation (POE) survey is carried out seasonally during the test year allowing to capture and explore variation on a seasonal basis with approximately three months in-between. The intent with four replies per house is twofold. Firstly, this is to identify if the occupants experience their perception changes during the stay; for instance – is their perception of indoor environment, expression, comfort or automation changing through their stay. The second aspect to the seasonal distribution is to explore if seasonal changes in weather (e.g. outdoor temperatures, daylight) influence occupant experience.

The questionnaire is translated into native language. It is a set of questions relating statements about satisfaction/dissatisfaction with energy consumption and production, indoor climate and air quality, daylight and electric lighting, house automation, and sustainability. Also addressed is the frequency of occupant interaction with elements of the house, and if the house fulfil expectations of the occupants (Olesen, 2014). In this study, the advantage of using a questionnaire is that it is easier to distribute several times, but the disadvantage is the limited number of houses studied, and thereby statistical tools that can be used to draw significant conclusions from the survey. Each family in four of the houses (HFL was not included) responded to the questionnaire four times during a year (at 3-month intervals) with two additional responses from CLH. In total 18 responses were made.

The questions about satisfaction were made as sets of Likert-scales categorised as *very satisfied, satisfied, neither satisfied nor unsatisfied, unsatisfied,* and *very unsatisfied*. Questions about how comfortable the subjects are in their indoor environments are categorised on a five-point rating scale by: *very rarely, rarely, occasionally, frequently, and very frequently*. Finally, the questions about energy, environment and sustainability were made as sets of statements and categorised as a three-point scale *yes, very, yes to some extent, no normally not,* or as sets of five-point scales *strongly agree - strongly disagree*, and *very good – very bad*.

The overall purpose of the evaluations is to get indications on how successful the houses are, if there are challenges or problems, and what can be learned and improved.

3 RESULTS

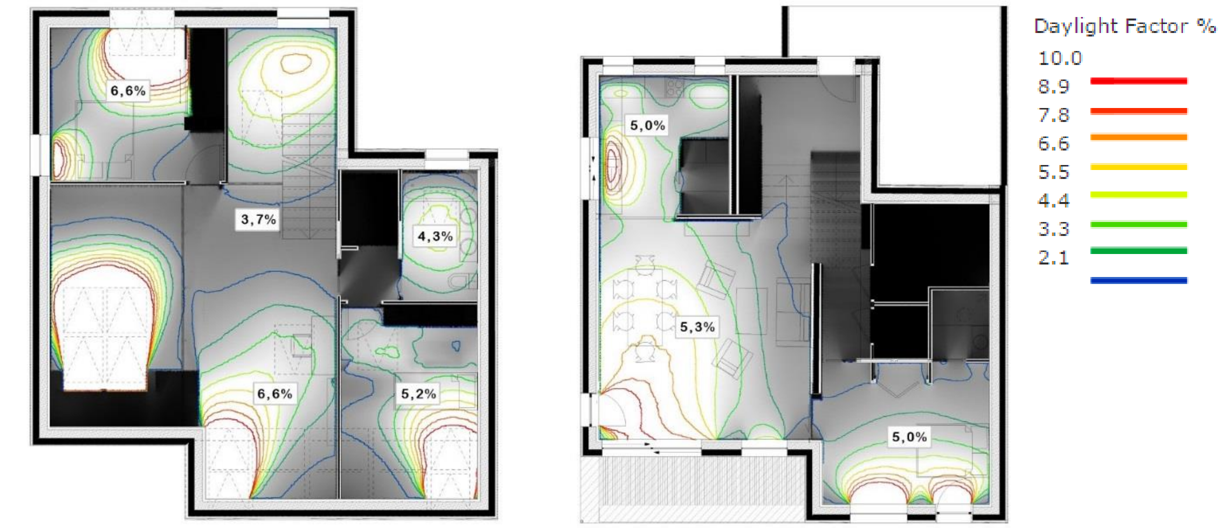
Generally, in the Post Occupancy Evaluation survey the indoor climate is rated as “very important” and the residents state most of the time that it is “good” or “very good” (>90% state “good” or “very good”).

3.1 Improved sleep, reduced number of sick days and emphasis on view out

The POE survey found that the families experience that their sleep quality compared to their former home is “better” (50%) or “almost the same” (39%), and when rating their children’s sleep quality, the tendency is a bit higher (“better” 56%; “almost the same” 44%). Furthermore, they have a significant experience that they have “less” sick days (83%) than in their former home, and they state their general health all in all as “good” or “very good”. View to the outside through the window is rated as “very important” (44%) or as “quite important” (50%). Between 72% and 83% of the residents reported that they were “satisfied” or “very satisfied” with the view in the house in general.

3.2 High daylight levels without overheating

All of the houses were designed for good daylight conditions, expressed by a target average daylight factor of 5% or higher in the main rooms. This was generally achieved, with only insignificant deviations. In the POE survey, the daylight levels in the houses is rated either as “much higher” (88%) or as “higher” (12%) than their former home. The families report that the daylight level is generally “appropriate” (>75%) in the kitchen, the living room, and the bedroom. Between 89% and 100% of the residents reported that they were “satisfied” or “very satisfied” with the daylight in the house in general. They also state the windows are “about right” for all the rooms (>89%).



1st floor Ground floor

Figure 3. Daylight calculation. The amount of daylight and the quality of its distribution in Maison Air et Lumière have been evaluated using VELUX Daylight Visualizer 2.

Good daylight conditions come with the potential risk of overheating, as plenty of sunlight also provides plenty of solar gains, which can lead to overheating in summer and intermediate seasons. The results from all houses show that overheating has been prevented. That is demonstrated by the fact that the buildings achieve category 1 according to the Active House specification for thermal comfort during summer (in less than 5% of the hours of the year the temperature is above category 1). See example of temperatures in Figure 1 from Sunlighthouse.

This is well in line with the POE survey, as the residents in all houses are either “very satisfied” or “satisfied” with the temperature conditions in general (90%). Most of the time, the temperature conditions is assessed as about right, but separated into the different seasons of the year, the winter and the spring/autumn is stated as time of the year when the temperature is sometimes evaluated as varying, while few state temperatures as too hot, even in the summer.

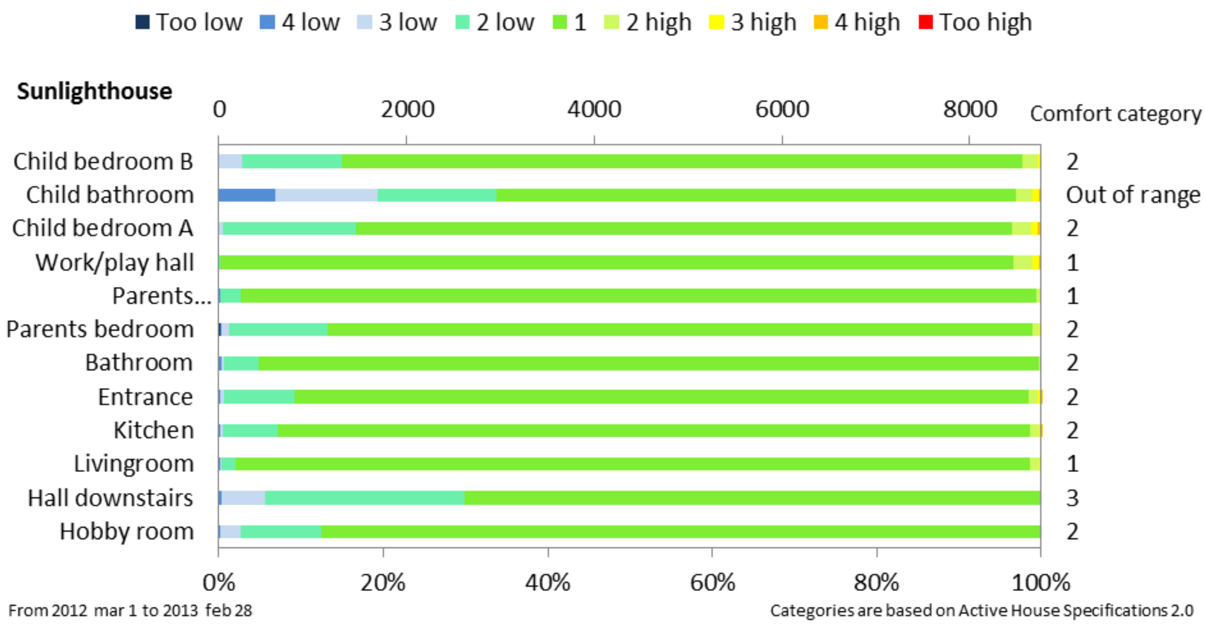


Figure 1. Thermal comfort of Sunlighthouse for each of the rooms evaluated according to Active House specification (based on adaptive method of EN 15251). Criteria are differentiated between high and low temperatures.

3.3 Electric light is not used between sunrise and sunset

The families state in the POE survey that they turn the electric light on “less often” (100%) than in their former home, and they evaluate the light levels as “appropriate” (>72%) in the focus rooms.

The measurements support this and show that electric light is generally not used between sunrise and sunset. This is the case not only during summer, but also during the darker winter months. This can be expressed by the term *daylight autonomy* (Reinhardt, 2001): Rooms can be expected to be daylight autonomous when the average daylight factor in a room is above 5%. The results support this.

3.4 No night-time overheating in bedrooms

Only limited research has been identified on the relation between temperature and sleep quality, but what is known is that the temperature in bedrooms during the night should not be *too high*, to prevent reduction of sleep quality (Laverge et al., 2011). In lack of a better threshold, category 1 is used as indicator of acceptable temperature for sleeping.

The time-of-day when the few annual hours with temperatures above Active House category 1 occur in the houses, have been identified. These hours occur in the afternoon. After sunset, the temperature drops to category 1 again with the exception of a few days per year in each house. This indicates that the houses have provided a thermal environment that did not pose a risk of reduced sleep quality. See Figure 2 for an example from the master bedroom in LichtAktiv Haus.

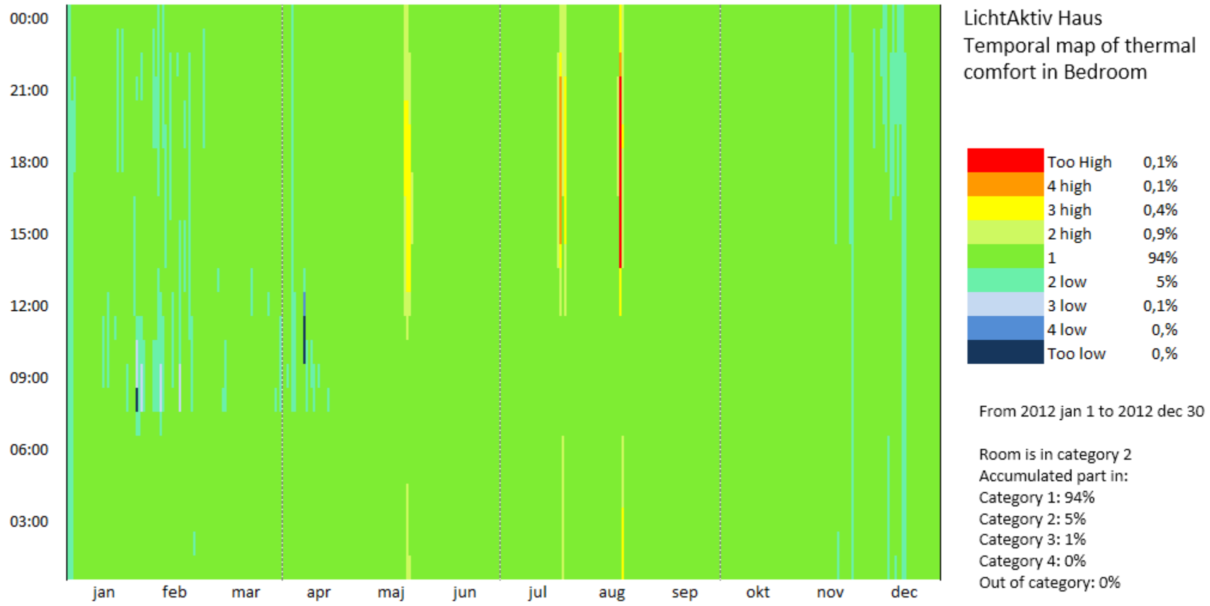


Figure 2. Thermal comfort of the master bedroom in LichtAktiv Haus. Each of the hours of the year are illustrated as a small square. The hours are presented so that hours of the same day are sorted from 00:00 to 24:00 (bottom to top), and from January to December (left to right). The few hours with temperatures above category 1 are seen as yellow, orange or red colour. These hours occur during three periods, each of 3-4 days. They begin around 13:00 and in most cases, the temperature has dropped to category 1 or 2 again before 22:00. During the night from 24:00 to 6:00 there is practically no hours with temperatures above category 2.

3.5 Ventilative cooling by natural ventilation prevents overheating. Night cooling is important.

A particular element of the present study is that the actual position of windows and solar shading has been included in the data recording, which provides detailed insights on the role of these components. The use of window openings follow the seasons; during spring and autumn windows are used on most days for approx. 50% of the time during daytime. During summer, windows are used more systematically during daytime hours, and also during the night. There is a correlation between the use of windows and hours without overheating. This is an indication that window openings have played an important role in maintaining good thermal comfort. See Figure 3.

Open windows during the night (night cooling) cools down the rooms from a temperature at the upper range of the comfort range to a temperature at the lower end of the comfort range, e.g. from 26°C in the evening to 20°C in the morning. The temperature can then rise during the day, in many cases without becoming uncomfortably hot at the end of the day. This underlines the importance of night cooling.

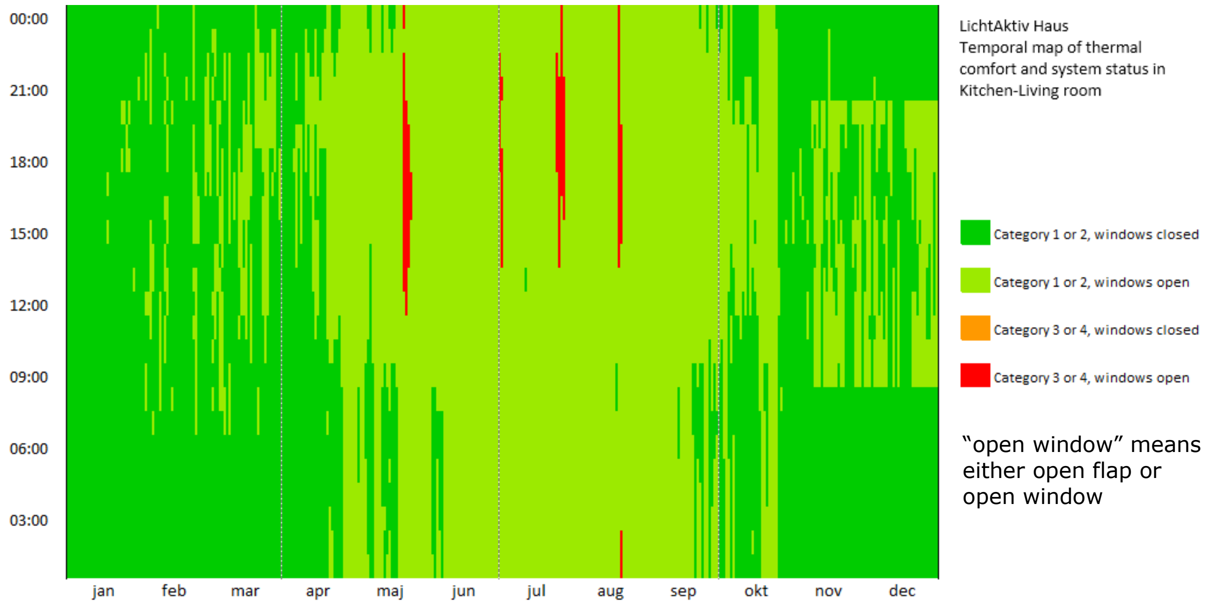


Figure 3. Thermal comfort of the kitchen/living room in LichtAktiv Haus, similarly as in Figure 2. On this figure, categories 1 and 2 are bundled, as well as categories 3 and 4. The position of windows is added (open or closed). The result is an illustration of when windows were open, and the relation to the thermal comfort at the same time. The light green squares represent hours when windows were open and good thermal comfort occurred; this happened during daytime in spring and autumn, and during the night in summer.

The results are supported by tracer gas measurements, which were used to investigate the airflow generated by ventilative cooling, and how large a temperature reduction ventilative cooling provided. The results showed that airflow rates of 10-20 air changes per hour could be achieved, and that the indoor temperature could be maintained 5°C lower than if ventilative cooling had not been applied (Favre et al., 2013).

3.6 Solar shading helps prevent overheating

The position of solar shading was recorded just as the position of windows. Awning blinds were the preferred type of external shading used on the houses, and the results show that the awning blinds had a role in providing good thermal comfort. The awning blinds were used the most during the summer, but also during spring and autumn. There is a correlation between use of awning blinds and hours without overheating.

3.7 Automation important

Automated control of window openings, solar shading and mechanical ventilation was used in all the investigated houses. The results show that solar shading and window openings are used frequently during work-hours on weekdays and during the night, e.g. at times when the families cannot be expected to be able to operate the products themselves. The same use of products could not have been achieved with only manual products.

The families respond in the POE survey that they are generally "very satisfied" or "satisfied" (>85%) with the way the house system operates the facade and roof windows, the indoor temperature, internal and external screen, and ventilation system (one house is solely naturally ventilated). They have a clear feeling that the way the control unit operates the house support their needs, and is either "easy" or "very easy" to use. It further shows that they "rarely" or "occasionally" use the control system to manually operate the facade and roof windows, internal temperature, but more frequently use the control system to operate the screening.

3.8 Satisfying CO_2-levels during summer

The CO_2 levels are low during the spring, summer and autumn seasons, typically below 900 ppm. Natural ventilation is used in this period as the only mean of ventilation, and the results clearly shows that with limited temperature difference to drive the stack effect, it is still possible to reach a reasonable level. During summer there is no electricity consumption for mechanical ventilation and no heat loss, so high ventilation rates and excellent indoor air quality can be achieved without any use of energy. It is also shown that openable windows were generally able to reduce or maintain low CO_2 levels in all the Model homes 2020.

The most challenging rooms are the bedrooms, as these are small rooms where approximately eight hours are spent each night, often two persons together in the same room. This is longer time than we spend in any other room in the home. Still, the CO_2-levels are maintained at a reasonable level in the bedrooms. Figure 4 is an example of the CO_2-level in a bedroom in Maison Air et Lumière.

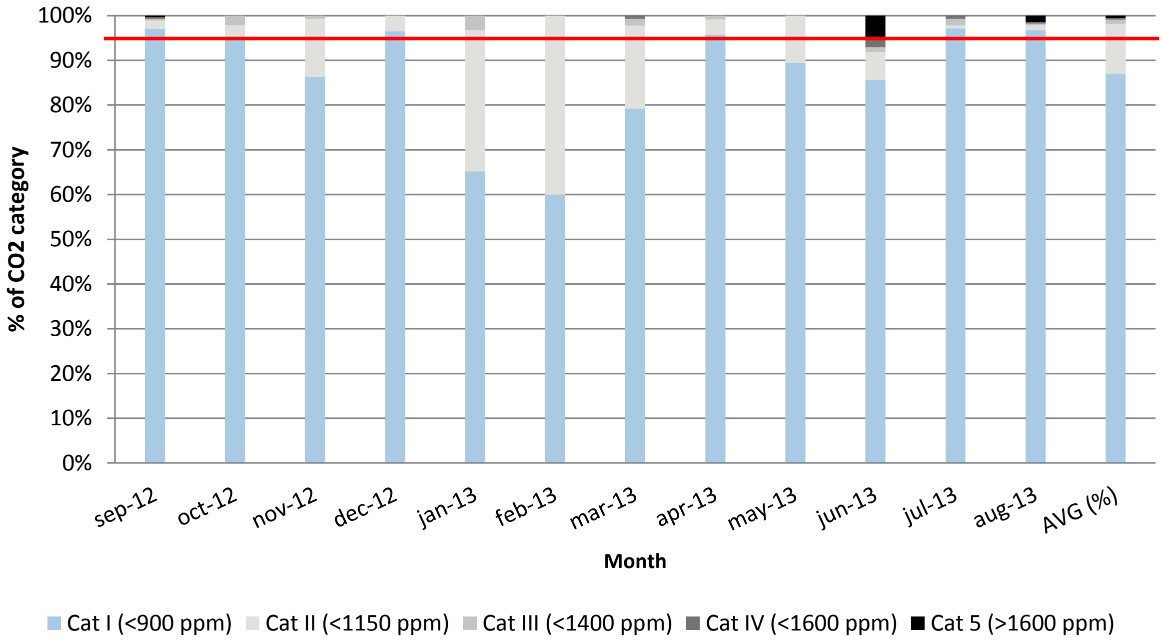


Figure 4. Bedroom in Maison Air et Lumière. Monthly distribution of night time hours in each of five categories for CO_2 level, based on Active House specification. The CO_2-level is lower during the summer than in winter.

The POE survey indicates that the perceived indoor air quality is good as it is rated as "very acceptable" (78%) or "acceptable" (22%), and the families state that they have not experienced any problems at all. If they want to improve the air quality, they open the facade and roof windows, and make airings. In most of the houses there is hybrid ventilation, so that mechanical ventilation with heat recovery is used during the winter to save energy. The mechanical ventilation systems are designed and commissioned to provide the ventilation rates required by the building codes, and they fulfil this requirement flawlessly. However, when the winter CO_2-levels are evaluated according to the Active House specification, particularly bedrooms only achieve a score of 2 or 3.

4 CONCLUSIONS

The five houses have good daylight conditions (DF > 5% in main rooms), and the results show that electric light under these conditions was not used between sunrise and sunset. The measurements show that good daylight conditions can be obtained without causing overheating, when solar shading and window openings are included in the building design and controlled automatically. Night cooling is a particular important aspect. It was found that high ventilation rates can be achieved also during summer with limited temperature difference available as driving force.

The use of ventilative cooling during summer also meant that the ventilation rates were high in this period, and as a consequence the measured CO_2-levels were low. The POE survey indicated that the families show high satisfaction with the indoor environment, that their expectations often are fulfilled, and that house automation is acceptable. Furthermore, combining excellent indoor environment with high quality homes, gives a clear indication that the residents experience better health and better sleep quality, as well as having less sick days than when living in their former homes.

5 REFERENCES

Active House (2011) *Active House Specification*, Active House Alliance, activehouse.info

CEN (2007). CEN Standard EN 15251:2007. *Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics*. European Committee for Standardisation.

Favre, B., Cohen, M., Vorger, E., Mejri, O., Peuportier, B. (2013) Evaluation of ventilative cooling in a single family house. Report. Ecoles de Armines.

Feifer, L, Foldbjerg, P, Asmussen, T F, Andersen, P (2014). Tomorrows buildings today – results, conclusions and learnings from a cross-european demonstration programme. *Proceedings of World Sustainable Buildings 2014*, Barcelona.

Foldbjerg, P, Asmussen, TF and Holzer, P, 2014. Ventilative Cooling of Residential Buildings - Strategies, Measurements results and lessons learned from three Active Houses in Austria, Germany and Denmark, *International Journal of Ventilation*, September 2014.

Hammershøj Olesen, GG (2014). *A Model for Enquiry of Sustainable Homes of Model Home 2020*. PhD Thesis, Aalborg University.

Laverge, J., Janssens, A. (2011) Physiological and Sensory Human Response to IEQ Indicators while Asleep. *Proceedings of Indoor Air 2011*, Austin, Texas

Reinhart, C., Walkenhorst, O. (2001) Dynamic RADIANCE-Based Daylight Simulations for a Full-Scale Test Office with Outer Venetian Blinds, *Energy and Buildings*, 33:7, pp. 683-697)

Venticool, the European platform for ventilative cooling. Published by Venticool. Last accessed January 2015. http://venticool.eu/faqs/.

OPTIMIZATION OF INDOOR AIR QUALITY THROUGH CONTROLLED CROSS VENTILATION IN THE RETROFITTING OF RESIDENTIAL BUILDINGS

Alberto Meiss*[1], Jesús Feijó-Muñoz[1] and Miguel Ángel Padilla-Marcos[1]

1 G.I.R. Arquitectura & Energía. Universidad de Valladolid
E.T.S. Arquitectura. Av. Salamanca, 18
47014 Valladolid, Spain
**Corresponding author: meiss@arq.uva.es*

ABSTRACT

As an alternative to adopting active architectural systems (mechanical systems) and taking advantage of the resources provided by nature, natural ventilation contributes interesting solutions to control the thermal balance and the air quality, and it is applicable in a variety of climate zones. Natural ventilation also solves some of the more common problems of mechanical systems, such as the noise factor and installation and maintenance costs.
However, it is not sensible to pretend that natural ventilation on its own can satisfy all of the air quality needs of a dwelling. Its greatest limitation is that its effects are random, variable, and difficult to control and regulate because the airflow varies as a function of the environmental conditions at any given time. The result is that the minimum required airflow rate for an acceptable level of air quality occasionally cannot be obtained.
To solve these lacks, it is possible to take advantage of the characteristics of hybrid ventilation (since these installations combine natural and mechanic ventilation systems) and demand-based regulation (in accordance with a specific concentration of contaminants or human presence).
The application of this design option to new construction is widely disseminated. However, its integration in retrofitting residential neighbourhoods, particularly those built in the second half of the twentieth century, is especially important and represents an added complication. This complication requires implementing a methodology to analyse its different components and design alternate construction solutions.
The methodology is then applied to the case study of CITyFiED European project (RepliCable and InnovaTive Future Efficient Districts and cities), in the case of Torrelago district, in the municipality of Laguna de Duero, surrounding the metropolitan area of Valladolid. The retrofitting actions involve 31 residential buildings in use nowadays and counting 1,488 dwellings.
Based on characterizing the temperature and wind factors that influence natural ventilation processes, the available flow rates for the case being studied are estimated. For this case, there are procedures based on complex numerical simulations using CFD software in combination with experimental tests.
Combined with experimental data (in situ measurements and pressurization tests to determine the air tightness of the enclosures), the values obtained in this way allow the quantification of the air renewal cycles in the different dwellings and the estimation of their ability to provide quality air to their users.
Those results must be subject to a critical review and to the development of new design alternatives, which may be incorporated in retrofitting works so as to guarantee at all times the indoor air quality by means of the most efficient option as regards energy.

KEYWORDS

Ventilation, building retrofitting, IAQ, ventilation efficiency, pressurization test

1 INTRODUCTION

Buildings are among the main consumers of energy in Europe, and it is commonly stated that, in 2013, the residential sector represented approximately 26.8% of all of the primary energy consumption in the EU (Eurostat, 2015). Therefore, to achieve the strategic 20-20-20 objectives, building retrofitting became a priority that has experienced a significant boom in recent years, with measures designed to improve the energy and comfort conditions in dwellings built under other, less-stringent guidelines.
The need to effectively ventilate affects both energy consumption and the indoor air quality. It seems reasonable that the design of ventilation should take advantage of the motion of natural air, acting in turn to address the larger problem of natural ventilation, which is its randomness resulting from its dependence on climate factors. It is possible to take advantage of these natural processes through support systems, something known as hybrid ventilation, and demand-based regulation, in accordance with a specific concentration of contaminants or human presence.
The application of this type of installation is widely accepted in the design of new buildings. However, when incorporating them into retrofitting occupied buildings, significant complications arise, including adequately quantifying the existing flow rate and prioritizing systems that minimize the disturbances resulting from their installation.
This study involves the analysis of and design proposal for a ventilation installation applied to a case study of the CITyFIED European project (RepliCable and InnovaTive Future Efficient Districts and cities (CITyFIED, 2015), which includes the comprehensive renovation of three large urban districts in Laguna de Duero-Valladolid (Spain), Soma (Turkey) and Lund (Sweden). The intervention in Laguna de Duero consists of retrofitting the Torrelago Urban Development (Figure 1), which is made up of 31 buildings with a total of 1,488 dwellings built between 1977 and 1981.

Figure 1: Aerial view of the Torrelago Development and the area being simulated

2 PRESSURE GRADIENT IN ENCLOSURES

The pressure gradient is the result of the combined effect of two simultaneous processes: natural convection (due to density differences in the air, a product of different temperatures on different sides of the enclosure) and forced convection (of natural origin resulting from the wind, or artificial origin, from mechanical ventilators).

In the Torrelago buildings, natural convection (or stack effect) becomes particularly important due to the height of the space where it is produced, that is, through the integrated vertical cores in the building (elevators, stairwells, shafts, etc.). The calculation of natural convection has been widely studied and compared (Etheridge, 1996; Meiss, 2015).
Wind action, however, is one of the most complex physical phenomena to simulate. The high number of variables that contribute to its effect makes it impossible to consider all of the parameters that affect the behaviour of wind in the urban environment. Of particular importance is the need to provide the best possible approximation of the fluid dynamic characteristics of the air and the geometric and roughness characteristics of the environment (Feijó, 2009) because the flow may enter the turbulent regime with the existence of obstacles or because of the air viscosity, fluid velocity and medium characteristics (Amorim, 2013; Blocken, 2007; Kim, 2004; Richards, 1993).
Computational Fluid Dynamics, or CFD, is a numerical-method based computational simulation that can be applied to fluid dynamics studies. This technique takes into account the physical processes that develop in a fluid domain bounded by different surfaces, and it can approximate the wind behaviour in predictive models that are simulated using Ansys Fluent R15.0© software. This study adopts the Reynolds-averaged Navier-Stokes model (RANS-Realizable-Enhanced Wall Treatment (EWT)) as the most appropriate model for simulating time-averaged static processes to obtain the fluid behaviour in urban environments under turbulent flow regimes (Hertwig, 2012; Moonen, 2011). The required validation was performed through stabilized solutions in an infinite time period with data obtained from a wind tunnel in static conditions (Leitl, 2000; CEDVAL, 2002), validating the velocity variables and turbulence profiles (k-e) and their behaviour in an isolated building and in the urban model (Padilla, 2015). The precision obtained in both models was considered acceptable for obtaining results applicable to the real case (Table 1).

Table 1: Precision of the validation model results

Variable	Isolated Building	Urban Model
Velocity	± 1.05%	± 3.68%
X Velocity	± 1.14%	± 2.01%
Y Velocity	± 12.30%	± 12.46%
Z Velocity	± 4.57%	± 4.23%

The study of the location was focused on the predominant wind velocities from the WSW and NE directions, during winter and summer and for the annual mean (IDEA, 2015), affecting both the isolated block (Figure 2) and the complete urban model (Figure 3) (Meiss, 2015). The CFD simulation provides a large quantity of data for the flow model: velocity distributions, static and dynamic pressures, particle trajectories, age of the air, and other parameters.

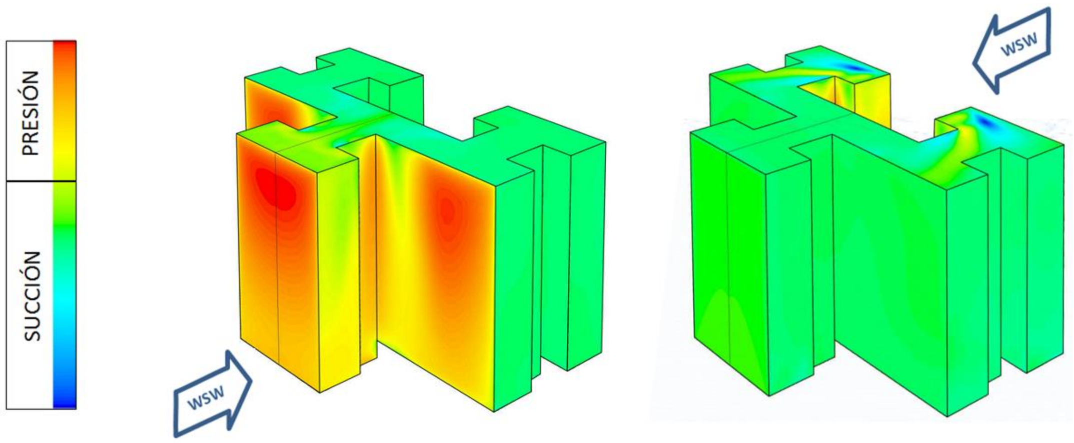


Figure 2: Wind effects on the building

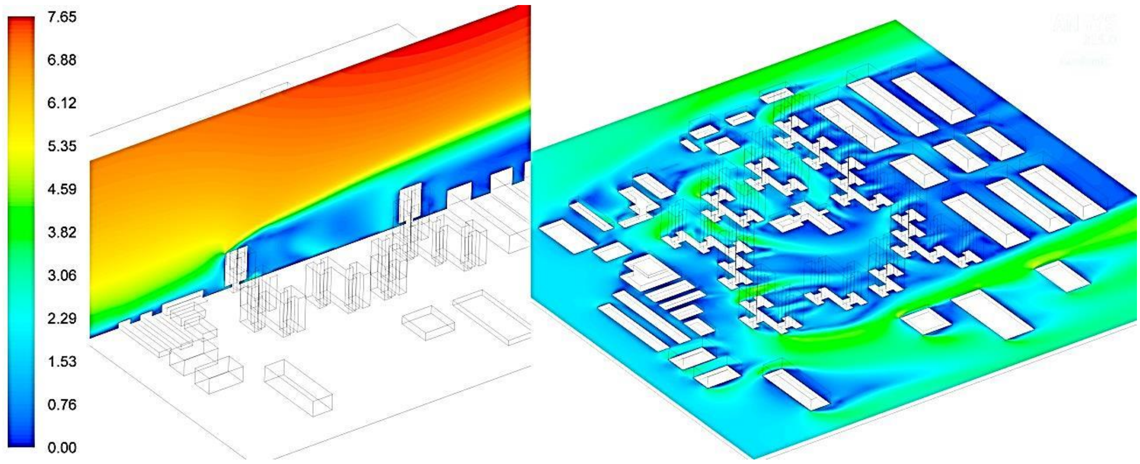


Figure 3: Wind velocity distribution from the WSW at district level (floor plan and cross-section)

The superposition of pressures, a product of thermal and wind action, allows calculating the effects of wind on the surface of each of the dwelling enclosures: the resultant pressures represent different cases throughout the year, with alternating seasonal pressures and suctions on the high and low floors.

3 AVAILABLE FLOW RATE OF NATURAL VENTILATION

The owners of the different dwellings in the urban district altered the apartments original design in different ways: replacing the original framing, changing the shutters, renovating the bathrooms and kitchens, blocking shared vertical ventilation shafts, etc. Thus, this study has sought a dwelling (Block 25, Apartment 9C) that in principle would serve as a reference because it retains the construction characteristics of the original project. The process to be developed seeks to quantify its available natural ventilation flow rate.

Pressure tests in dwellings can determine the permeability of the enclosure, distinguishing the values for blind spots and holes based on the method in the European Norm EN13829. The characteristics of these dwellings lead us to assume that infiltrations calculated this way occur mostly through the outer envelope (Meiss, 2015).

Based on these data, it is possible to quantify the natural ventilation flow rate of the dwelling (Table 2) through a process that jointly analyses the different enclosures exposed to different pressure gradients. The procedure is based on the European Norm EN13465 calculation methods for obtaining airflow rates in dwellings, which follows classical theoretical guidelines for natural ventilation. The systems of equations thus formulated are solved iteratively using the Engineering Equation Solver (EES) calculation software, obtaining air flow rates by solving for the unknown pressure inside the dwelling (considered a single volume).

Table 2: Summary of natural ventilation flow rates

<table>
<tr><th>Dwelling</th><th colspan="7">according to EN 13829</th></tr>
<tr><td rowspan="7">Block 25 Apartment 9C</td><td>n_{50} rate</td><td>n exp.</td><td colspan="4">Permeabilidad al aire de las carpinterías a 100 Pa ($m^3/h \cdot m^2$)</td><td>Presence of natural intake in the shunt</td></tr>
<tr><td>6.2</td><td>0.64</td><td colspan="4">48.6</td><td>Yes</td></tr>
<tr><td colspan="6">according to EN 13465</td><td rowspan="3">Requirement in the DB-HS3 (1/h)</td></tr>
<tr><td colspan="3">Estimated Volumetric Flow (m^3/h)</td><td colspan="3">ACH Natural Ventilation (1/h)</td></tr>
<tr><td>Winter</td><td>Summer</td><td>Annual</td><td>Winter</td><td>Summer</td><td>Annual</td></tr>
<tr><td>164.20</td><td>97.18</td><td>140.85</td><td>0.75</td><td>0.44</td><td>0.65</td><td>≥ 0.74</td></tr>
</table>

The results indicate that the air renewal rate (ACH), a product of natural ventilation, is not sufficient for the annual mean or during the summer. During the winter, the rate reaches the minimal values; however, this result is due to the low quality of the original framed openings (i.e., an aluminium sliding window with single-pane glass), which implies that its replacement would result in lower ventilation flow rates.

4 REQUIRED VENTILATION FLOW RATE

The Spanish Technical Building Code (CTE) includes the requirement HS3 “Indoor Air Quality”, which was added after the buildings in this study were built. Such a requirement, being performance-based, expresses that “buildings shall have means to ventilate their indoor spaces adequately, eliminating contaminants that may be produced commonly during normal use of the buildings, so that there is a sufficient flow rate of exterior air that guarantees the extraction and expulsion of contaminated air.”
The Basic Document that addresses this provision (DB-HS3) quantifies that requirement using reference flow rates by activities based on the number of people or the room surface area for different cases.

4.1 Fundaments

Even if the global flow rates predicted for each dwelling are fulfilled, it is easy to see that there may be areas inside of a dwelling that exhibit an excess or deficit of ventilation given their specific spatial configurations.
It is important to understand that reference flow rates are identified using the pattern of perfectly mixed flow, that is, with an efficiency ε^{a}=50%. This fact, in combination with the performance-based nature of the CTE, adequately justifies modifying the flow rates as a function of the ventilation efficiency in each case.
The problem is illustrated in the simulation of an operating ventilation system (Figure 4): the air penetrates the gap beneath a door and leaves the space through a similar adjacent opening. It can be observed that large areas of stagnant, un-renovated air are created, indicating that only a small proportion of the provided air is dispersed throughout the space. This result qualitatively shows that the design of this system is not efficient when “providing air to those areas of the dwelling that require it,” even though it fulfils the minimum flow rate specified by the CTE.

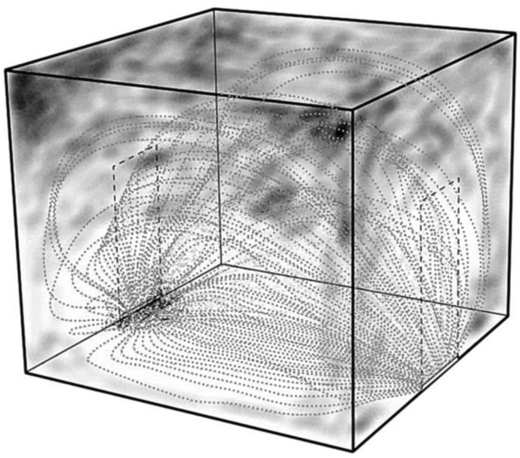


Figure 4: Stagnation of air with low efficiency ε^{a}

To establish the air flow pattern inside of the dwelling, the physical locations of the inlet, paso and exhaust openings have significant effects on the air change efficiency of a room, ε^a (Awbi, 2000; Gao, 2011). This concept is defined as the ratio between the minimum possible time it takes to replace the air, or turn-over time τ_t, and the corresponding mean time, or average time of exchange of air $\bar{\tau}_{exc}$, which is equal to twice the room average age-of-the-air (Sandberg, 1981):

$$\varepsilon^a = \frac{\tau_t}{\bar{\tau}_{exc}} \cdot 100 \ \ (\%) \tag{1}$$

In experimental tests, the efficiency ε^a is calculated by dividing the local mean age-of-the-air at the exhaust point τ_e by twice the room average age-of-the-air $\langle\bar{\tau}\rangle$ (because we use the optimum model, the so-called "piston flow," as a reference (Sandberg, 1983)):

$$\varepsilon^a = \frac{\tau_e}{2 \cdot \langle\bar{\tau}\rangle} \cdot 100 \tag{2}$$

When the situation arises in which the contaminant source is not located and one wishes to simultaneously ensure adequate air quality in the entire volume of the room, the efficiency ε^a enables a generic analysis of the rooms and their ventilation systems, so this coefficient is applied in this study (Mundt, 2004).

4.2 Basis for the experimental and numerical procedure

Having a test chamber in the Ventilation Laboratory of the Architecture School of Valladolid (Figure 5) has allowed us to carry out a comprehensive experimental study of air flow, at a 1:1 scale, of the different room configurations, which can be compared with those that make up the dwelling being studied.

Figure 5: Ventilation Laboratory of the Architecture E.T.S. of Valladolid

Boundary conditions were established for this phase of the study:

- Opened doors or windows is not considered, nor is any other action that produces a non-static regime (with the purpose of analysing the efficiency while fulfilling the minimum ventilation conditions);
- The regime is isothermal, with no differences in temperature between the inlet air and the air contained in the room (given that previous studies show that the global

efficiency result obtained is not altered (Meiss, 2013)). The remaining surfaces that make up the volume are considered smooth and isothermal walls.

In measurements carried out at control points arranged according to the European Norm EN13141, a tracer gas (SF_6) is used to characterize the flow of air from the time it enters until it leaves each room. The tests were conducted according to the method of falling concentration, obtaining results with mean deviations with respect to the experimental mean of ±2%. These values validate the numerical CFD model that reproduces the air flow in different rooms in the dwelling.

4.3 Application to the dwelling being studied (B25 – Apartment 9C)

The reference dwelling is made up of four bedrooms, a living room, a kitchen and two bathrooms, all of which have openings on their façades, except for one of the interior bathrooms (Figure 6). It is expected that exterior air flows through suction from the least contaminated rooms (the bedrooms and living room) to the more contaminated rooms (the kitchen and bathrooms).

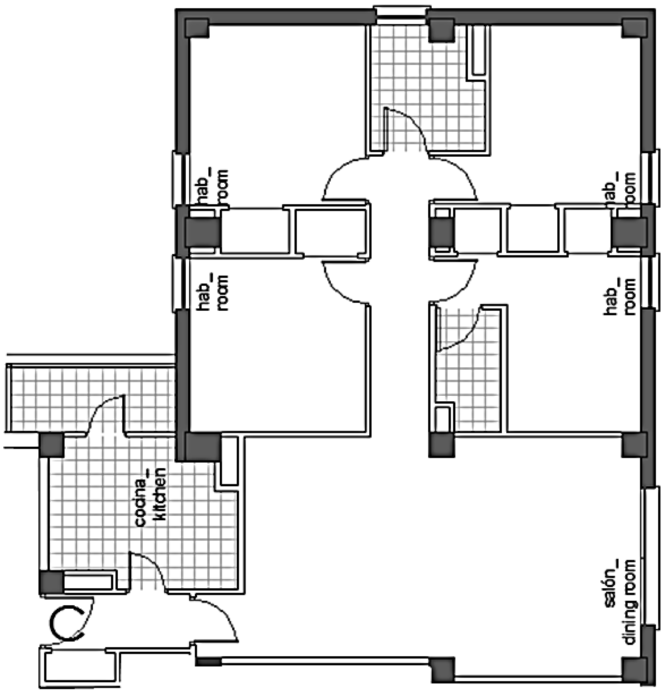


Figure 6: Reference dwelling

Calculating the efficiency allows exceeding the simple quantitative minimum value specified by DB-HS3 to qualitatively control the air change in the entire habitable space (with the possibility of being able to reduce or increase the flow rate recommended by the norm (Meiss, 2011)). If such an analysis is performed in the design phase of the building, it is possible to locate the openings to achieve the most efficient airflow. In the case of a retrofit, the CFD simulation is limited to calculating the efficiency according to the current locations of the openings.

Through the use of a collection of CFD simulations (1152 in total, encompassing practically all of the cases related to the rooms in dwellings) in our own laboratory with the support of the Spanish research project "Optimization of air inlet openings for the CTE-HS3" (Meiss, 2009), we were able to directly calculate the air change efficiency ε^a values for each habitable room (Figure 7). This calculation makes it possible to obtain the average age-of-the-air in the dwelling and compare it to the one deduced using HS3 (that is, an average age-of-the-air based on ventilation through a mixed air flow distribution pattern at a specific concentration of contaminants).

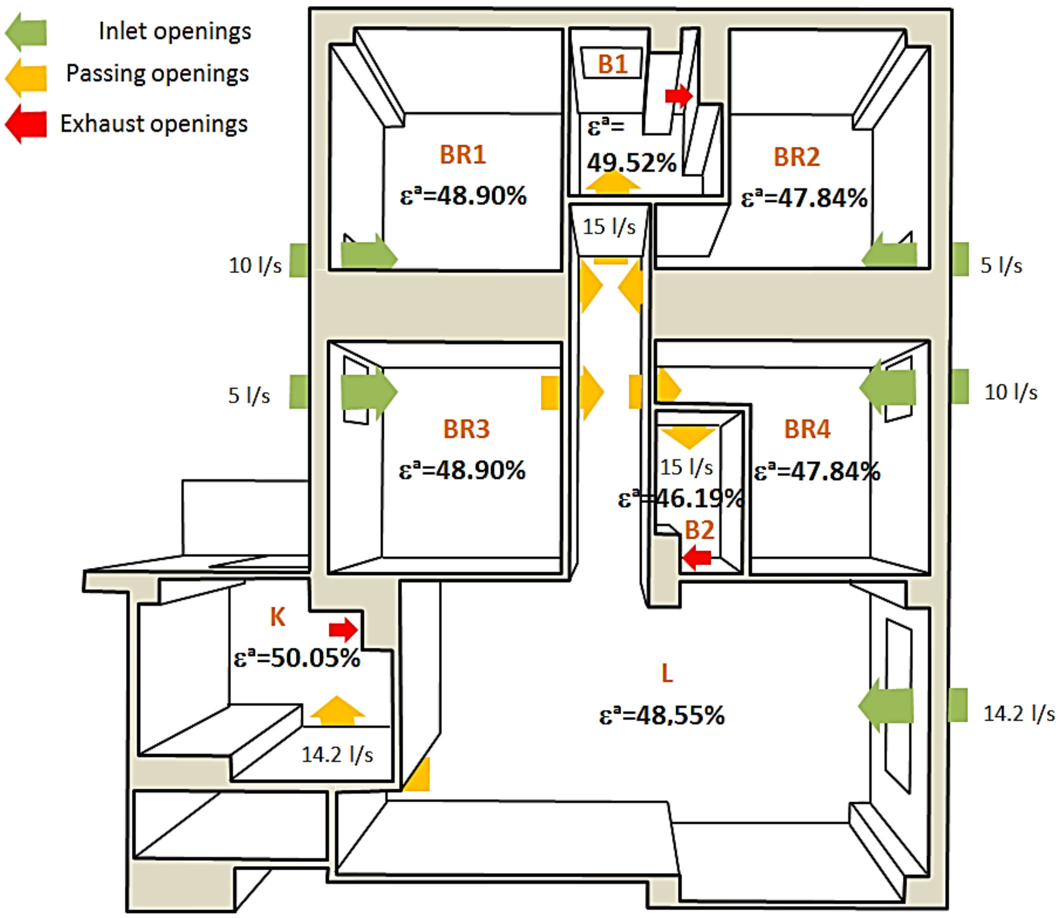


Figure 7: Study of the ventilation airflow

Table 3: Ventilation needs as a function of ventilation efficiency

Room	DB-HS3 Flow rate (l/s)	Vol (m^3)	τ_e room (s)	$\langle\bar{\tau}\rangle$ ε^a=50% (s)	ε^a=real (%)	$\langle\bar{\tau}\rangle$ ε^a=real (s)	Equivalent Flow rate (l/s)
Bedroom BR1	10	22.1	2210	2210	48.90	2260	**10.2**
Bedroom BR2	5	20.73	4145	4145	47.84	4332	**5.2**
Bath B1	15	9.20	615	615	49.52	621	**15.4**
Bedroom BR3	5	8.56	4280	4280	48.90	4376	**5.1**
Bedroom BR4	10	7.62	1905	1905	47.84	1991	**10.4**
Bath B2	15	2.13	355	355	46.19	384	**15.5**
Living room L	14.2	26.25	3697	3697	48.55	3807	**14.2**
Kitchen K	14.2	7.08	997	997	50.05	996	**14.2**

With efficiency ε^a values below to 50%, that in spite of complying with the norm-based ventilation flow rate, we are not able to adequately ventilate the space, and the resulting air quality is not good enough (that is, it does not fulfil the basic indoor air quality requirement). The only solution is to slightly increase the circulating ventilation flow rate up to a different equivalent flow rate because the arrangement of ventilation openings is not the most appropriate (Table 3).

5 CONCLUSIONS: CONTROLLED VENTILATION PROPOSAL

Applying the methodology allows correctly characterizing the availability of natural ventilation and the specific needs of the dwelling to be retrofitted. Results have shown that it is impossible to rely solely on natural contributions because the required continuity of air renovation cannot be guaranteed. The result is that controlled ventilation systems are required. DB-HS3 already predicted this situation in its so-called “hybrid ventilation,” but its novel implementation in 2006 was directed to a more generic dwelling, with a very rigid formulation in terms of the flow rates and material aspects of the installation. Clearly the case of retrofits, as would be relevant to our study, was not specifically considered.

As a conclusion to the experiences discussed in this paper, the objective of which was to solve the problems that have been detected when applying HS3, we propose an optimization of ventilation based on the following general principles:

- The air quality should be quantified and prescribed as a function of the use of the space.
- Natural ventilation should be prioritized over any mechanical procedure, which would be reserved for times with an absence of passive resources.
- The expected ventilation flow rates should be variable and should adjust to the needs of the activity that takes place at each moment.
- The system should control and ensure the minimum quality levels that were previously prescribed, taking into account the permeability of the dwelling.
- Damp rooms in the dwelling (kitchens and bathrooms) should be ventilated through suction with respect to the other rooms during the times in which they become contaminated.
- Fossil energy consumption should be minimized given the adverse impact of ventilation on comfort conditions.
- The ventilation system should allow the free contributions of heat and cold from the outdoor air to achieve comfortable conditions.
- To implement the system in retrofits, even in occupied buildings, its installation should be simple and independent from other dwellings.
- The system should be able to be managed individually for each dwelling, making the user aware of the advantages of installation through directly controlling its operation.

With these premises, a specific system we refer to as “controlled cross ventilation” (CCV) is proposed based on a hybrid behaviour resulting from its two operation regimes:

- A passive regime using crossed natural ventilation, with the aid of mixed inlet and exhaust openings; in other words, the openings can act as inlet or exhaust points depending on wind pressures. These openings could be in the form of aerators or use micro-ventilation procedures based on the exterior framing. The bathrooms and kitchens would also contribute to the movement of air through their extraction openings, which at those times would act as mixed depending on the specific pressure differential.
- An active regime using forced ventilation generated by mechanical extractions located in damp spaces. In this case, the mixed openings distributed throughout the remaining rooms, such as living and bedrooms, would act as entry openings.

The so-called active regime would be activated only when its contribution is necessary to ensure the ventilation quality. The quality would be controlled by detectors or sensors of various phenomena that indicate a lack of quality. Specifically in living rooms and bedrooms, a good air quality indicator would be the CO_2 concentration because in those locations, contamination comes from the metabolic activity of the occupants.

In kitchens, there would be a greater number of symptoms that could indicate the presence of contaminants. In addition to carbon dioxide, increased relative humidity or the presence of

combustible gases could also be indicators in a gas range. However, the ventilation system should also be activated by other signals that may be accompanied by contaminant emissions, such as turning on the electric range or oven, and even the detection of people, either directly or through turning on lights.
In bathrooms and laundry rooms, the most representative contaminant likely to be detected is relative humidity. Olfactory contamination would be detected indirectly by the presence of people who are directly detected, or as a result of turning on lights.
However, mechanical extraction could be accomplished through a single multi-zone aspirator for a series of damp rooms in the dwelling or through several individual aspirators in each bathroom or kitchen. Intermediate solutions could also be formulated, which would depend on the dwelling configuration and the limitations of the building.
The extracted air would leave directly towards the exterior of facades, except when the contaminated air is emitted into interior courtyards, in which case it would be necessary to install an extraction duct leading to the roof. If this duct were shared, there would have to be a gate in each mechanical aspirator with automatic opening during expulsion periods.
In addition, to recover the substantial energy lost to the ventilation flow rates, it would be beneficial to have a heat recovery system. This system could operate using a closed circuit with a fluid that facilitates an energy balance in the air that goes through the exterior openings of the dwelling.
Finally, in extreme cases of excessive exterior pollution, appropriate filters could be used in inlet or mixed-use openings. Their characteristics would depend on a map of the air quality in the city where the dwelling is located.

6 ACKNOWLEDGEMENTS

The research leading to these results has been carried out under the CITyFiED project, which has received funding from the European Union’s Seventh Framework Programme for research, technological development and demonstration activities under the Grant Agreement number 609129, and the OPABAD project, wich has received funding from the Ministerio de Educación y Ciencia of Spain under the Grant Agreement number ARQ2002-02177.

7 REFERENCES

Amorim, j.h.; rodrigues, v.; tavares, r.; valente, j.; borrego, c. (2013). Cfd modelling of the aerodynamic effect of trees on urban air pollution dispersion. Sci. Total environ. 641-642, 541-551.

Awbi, h. Air distribution in rooms. Ventilation for health and sustainable environment. Roomvent 2000. Amsterdam: elsevier.

Blocken, b.; carmeliet, j.; stathopoulos, t. (2007). Cfd evaluation of wind speed conditions in passages between parallel buildings: effect of wall-function roughness modifications for the atmospheric boundary layer flow. J. Wind eng. Ind. Aerodyn., 95, 941-962.

Cedval (2002). Compilation of experimental data for validation of microscale dispersion models. Meteorologisches institut, universität hamburg. Available at: http://www.mi.zmaw.de/index.php?id=433.

Cityfied (2015). Replicable and innovative future efficient districts and cities european project. Coordinator: cartif. Available at: http://es.cityfied.eu/

Etheridge, d.; sandberg, m. (1996) building ventilation. Theory and measurements. Baffins lane: john wiley & sons.

Eurostat (2015). Final energy consumption by sector. Available at: http://ec.europa.eu/eurostat/web/product s-datasets/-/tsdpc320

Feijó-muñoz, j.; meiss, a. (2009). El espacio exterior del cte-db-hs3: metodología para el estudio de soluciones alternativas. Ciudad y territorio: estudios territoriales, 167; 71-79.

Gao, c.f.; lee, w.l. (2011) evaluating the influence of openings configuration on natural ventilation performance of residential units in hong kong. Building and environment, 46, 961–969.

Hertwig, d.; efthimiou, g.c.; bartzis, j.g.; leitl, b. (2012). Cfd-rans model validation of turbulent flow in a semi-idealized urban canopy. J. Wind eng. Ind. Aerodyn., 111, 61-72.

Idae (2015). Atlas eólico de españa. Available at: http://atlaseolico.idae.es/meteosim/.

Kim, j.j.; baik, j.j. (2004). A numerical study of the effects of ambient wind direction on flow and dispersion in urban street canyons using the rng k–e turbulence model. Atmos. Environ., 38, 3039-3048.

Leitl b. (2000). Validation data for microscale dispersion modelling. Eurotrac newsl., 22, 28-32.

Meiss, a. (2009). Estudio de la eficiencia de la ventilación en viviendas a partir de parámetros de diseño arquitectónico. Universidad de valladolid. Doctoral thesis.

Meiss, a.; feijó-muñoz, j. (2011). Influencia de la ubicación de las aberturas en la eficiencia de la ventilación en viviendas. Informes de la construcción, 63, 53-60.

Meiss, a.; feijó-muñoz, j. (2013). Age-of-the-air in rooms according to the environmental condition of temperature: a case study. Energy and buildings, 67, 88-96.

Meiss, a.; feijó-muñoz, j.; padilla-marcos, m.a. (2015). Methodology applied to the evaluation and design of ventilation systems in residential building retrofits. In progress to publication.

Moonen, p.; dorer, v.; carmeliet, j. (2011). Evaluation of the ventilation potential of courtyards and urban street canyons using rans and les. J. Wind eng. Ind. Aerodyn., 99, 414-423.

Mundt, e.; mathisen, h-m.; nielsen, p.; moser, a. (2004). A ventilation effectiveness. Guidebook nº2. Helsinki: rehva – federation of european heating and air-conditioning associations.

Padilla-marcos, m.a.; feijó-muñoz, j.; meiss, a. (2015). Wind velocity effects on the quality and efficiency of ventilation in the modelling of outdoor spaces. Case studies. Building services engineering research and technology (accepted- published pending).

Richards, p.j.; hoxey, r.p. (1993). Appropriate boundary conditions for computational wind engineering models using the k-e turbulence model. J. Wind eng. Ind. Aerodyn., 46-47, 145-153.

Sandberg, m. (1981). What is ventilation efficiency?. Building and environment, 16, 123-135.
Sandberg, m.; sjöverg, m. (1983) the use of moments for assessing air quality in ventilated rooms. Building and environment, 18, 181-197.

WIND-INDUCED PRESSURE COEFFICIENTS ON BUILDINGS DEDICATED TO AIR CHANGE RATE ASSESSMENT WITH CFD TOOL IN COMPLEX URBAN AREAS

Stéphane SANQUER*, Guillaume CANIOT

METEODYN
14, Bd Winston Churchill
44100 Nantes - FRANCE
**Corresponding author: stephane.sanquer@meteodyn.com*
Presenting author: underline First & Last name

ABSTRACT

The paper presents a numerical methodology to assess the natural ventilation. UrbaWind is an automatic computational fluid dynamics code. It was developed to model the wind in urban environments. The turbulence modelling, namely the dependence of turbulence length on the distance from wall, and the model constants were calibrated in order to reproduce with good agreements flow separation around buildings walls and pressure coefficient field on façades. Numerical results match well with the experiments: separation patterns and pressure field on walls in dense urban areas. Examples are presented at the end of the paper in order to show the advantages of the methodology for urban designers as they need pressure coefficients to assess the air change rate of buildings.

KEYWORDS

Natural ventilation, Wind, Air change rate, Pressure coefficient, Numerical simulations

1 INTRODUCTION

From an urban designer point of view the knowledge of the urban climatology, especially the wind flow around buildings, is crucial in many applications such as air quality and thermal behaviour inside buildings since the air and heat exchanges depend on the wind pressure on façades). CFD tools dedicated to computing the wind flow inside a built environment give advantages to understand and interpret the wind in any urban area. Wind mappings become useful to urban designers to optimize the master plan: position and orientation of buildings to improve natural ventilation, small energy production and pedestrians wind comfort. In the initial sketch designers should give a technical answer to improve the master plan as quickly as possible. As simulations are carried out with a high level of accuracy, CFD may be time consuming for large and complex urban master plans. A short time response is expected

during the sketch process whereas computational time can be very long. In that context, methodology should be defined to increase the efficiency of the numerical approaches.
Three improvements axes were highlighted:

- Time of computation reduced by improving the mesher and the solver, using a turbulence model that speeds up the convergence
- For natural ventilation assessment, a macroscopic approach remains the best way to reduce the time to compute Air Change Rate (ACH). Compute the flow through windows and inside any building in an urban area is not common for consulting services
- Information delivered by the CFD tool is not always understandable by designers and may not be used without statistical treatment including the local climatology.

It is well known that the natural ventilation of a building is driven by the combined forces of wind and thermal buoyancy. The pressure field on the building's envelope generates flows through openings allowing the indoor environment refreshment. When opening areas are large enough, and except for high volumes, natural ventilation is mainly driven by wind forces.

Commonly cross-ventilation efficiency is assessed considering flow-rates through openings depending on the external wind pressure at openings. The cross-wind flow rate depends on the size of the openings A, on their aerodynamic efficiency, namely their aerodynamic discharge coefficients C, on the pressure coefficients on the building envelope Cp and on the reference wind velocity upstream the building U_{WIND}. For a simple volume with two openings, the cross wind flow rate was written by Aynsley et al. (1977):

$$Q = \sqrt{\frac{Cp_1 - Cp_2}{\frac{1}{A_1^2 C_1^2} + \frac{1}{A_2^2 C_2^2}}}\, U_{WIND} = A_a \sqrt{\Delta Cp}\, U_{WIND} \qquad (1)$$

Hence, mass flow rate depends on three variables: the wind velocity far upstream the buildings U_{WIND}, the aerodynamic area A_a and the pressure differential ΔCp. If the number of openings is greater or if the internal volume is divided into sub volumes, a resolution for non-linear system has to be used. These methods called macroscopic approaches give air change rate from the pressure field characteristics.

The pressure on the envelope of the buildings can be given with tables (among others Clarke et al. 1990, Liddament, 1986) or with parametric models based on wind tunnel data (among others Grosso 1992, Muehleisen and Patrizi 2013, Costola et al. 2009). Numbers of numerical thermal software use macroscopic ACH assessment such as among others Energy+ to design natural ventilated buildings. Pressure coefficient data are given by AIVC or ASHRAE databases. Unfortunately for designers, these tables or correlations are potentially inapplicable when buildings are irregular and different from the simple shape or when they are at the side of other buildings in a complex urban area. Costola et al. (2009) give analytical relationship for sheltered buildings for homogeneous urban master plan. Unfortunately, although useful, such analytical approach suffers from an obvious lack of universality.

In summary, the pressure coefficient on outside walls of buildings could not be known with this method in complex urban places for two reasons:

- They depend strongly on the buildings shape and the pressure field is very inhomogeneous on the building envelope
- They depend strongly on the influence of neighboring buildings

The pressure on the building walls depends on the flow patterns. CFD software used to compute wind in large urban area solve the Reynolds Averaged Navier- Stokes Equations where turbulent fluxes are known through the turbulence energy balance equation. Separation or reattachment of the flow on walls, length of a recirculation area in a building wake depend on the ability of the turbulence model to well consider the diffusion of momentum in the flow. The first purpose of this work is to present a CFD methodology with a one-equation turbulence model allowing to catch the mean flow separation and to extract pressure field on building walls. Validation will be shown with a detached cube (Costola et al. 2009) and a high rise building of the Architectural Institute of Japan (Tominaga et al., 2008).
The second aim of the paper is to present through examples the limitation of analytical approaches giving Cp values especially in complex urban areas. Order of magnitude of discrepancies will be given in order to suggest to any building designers to assess the pressure coefficient values to use complex method rather than analytical approach or databases.

2 NUMERICAL APPROACH: METHODOLOGY AND VALIDATION

2.1 Wind computation in any complex urban area

The CFD method of UrbaWind consists of solving the Reynolds-Averaged Navier-Stokes equations on an unstructured rectangular grid with automatic refinement of the mesh near obstacles. The CFD tool delivers the tri-dimensional mean velocity field, the turbulence energy field and the mean pressure for each point in the domain.

When the airflow is steady and the fluid incompressible, the mean equations for the mean velocity components $\overline{u_i}$ contains unknown quantities, the turbulent fluxes that can be solved by one-equation or two-equation models.

The equations resolution is based on a finite volume method with a rectangular multi-bloc refined mesh. A very efficient coupled multi-grid solver is used for a fast convergence for every kind of geometry. (Ferry, 2002). Boundary conditions are automatically generated. The mean velocity profile at the computation domain inlet is determined by the logarithmic law in the surface layer, and by the Ekman function (Garratt, 1992). A ‘Blasius’-type ground law is implemented to model frictions (velocity components and turbulent kinetic energy) at the surfaces (ground and buildings). The effect of porous obstacles is modelled by introducing a volume drag force in the cells lying inside the obstacle.

The transport equation for the turbulent kinetic energy k contains a dissipation term □ deduced from the mixing-length theory.

$$\frac{\partial}{\partial x_i}\left[\rho \overline{u}_i k - \left(\frac{\mu_T}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right] = P_k - \varepsilon = \mu_T\left(\frac{\partial \overline{u}_i}{\partial x_j} + \frac{\partial \overline{u}_j}{\partial x_i}\right)\frac{\partial \overline{u}_j}{\partial x_j} - \varepsilon \qquad (2)$$

The turbulence viscosity $□_t$ is considered as the product of a length scale with a speed scale, which are both characteristic scales of the turbulent fluctuations.

$$\mu_t = \rho k^{1/2} L_T \text{ and } \varepsilon = C_\mu \frac{k^{3/2}}{L_T}$$

(3)

The turbulent length scale L_T varies linearly with the distance at the nearest wall d. The dependence of L_T on the distance via a coefficient C_L was defined in order to well reproduce the flows separation around typical building façades and roofs. C_μ is usually fixed at 0.09 for grid turbulence (standard k-ε model). In a boundary layer, the C_μ constant depends on the stability. The model defining the turbulence length scale by Yamada and Arritt (Hurley, 1997) suggest a dependence of C_μ on the stability criteria in the range 0.01-0.09. Comparisons with experimental results were made in order to calibrate the C_L and C_μ. for such urban flows.

X_R is the length of the recirculation area in the wake of a tall building sized as 20x20x40 m (figure 1). Velocity magnitude on the wake axis was plot versus the distance from the backward wall (figure 2). Flow separation above the building roof is also highlighted (figure 2) and the flow separation length X_{R_TOP} extracted. All the results were summarized in the table 1. These results were compared with many numerical simulations and experiments of Meng and Hibi (1998) described by Tominaga *et al.*(2008).

The best choice to reproduce the flow separation both in the wake and the upper vortices gives a couple of reference values for C_L and C_μ.

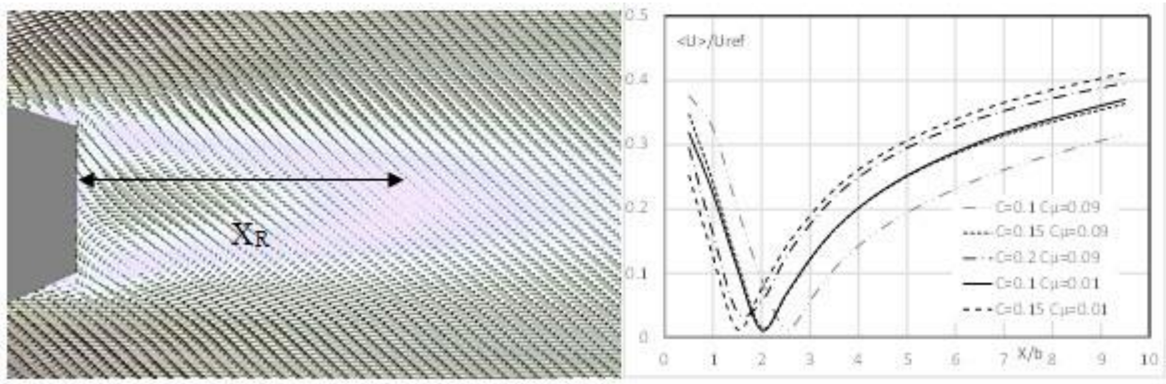


Figure 1 Recirculation in the wake of a square high-rise building

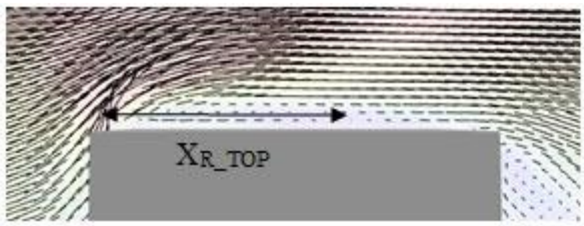


Figure 2 Flow separation above the roof of a square high-rise building

Table 1 : Comparison of Reattachment length of Numerical models and Experimental value (b is edge)

Turbulence model	**Reference**	**X_R/b**	**X_{R_TOP}/b**
k-□□□Standard□	**Tominaga el al.**	**2.7**	**No separation**
k-□□□Modified□	**Tominaga el al.**	**3 to 3.2**	**0.52 to 0.58**
Differential stress model	**Mochida et al.**	**4.2**	**>1**
LES	**Tominaga el al.**	**1 to 2.1**	**0.50 to 0.62**
k-l (C_L=0.2, C□=0.09)	**UrbaWind**	**1.7**	**0.25**
k-l (C_L=0.15, C□=0.09)	**UrbaWind**	**2.5**	**0.60**
k-l (C_L=0.1, C□=0.09)	**UrbaWind**	**2.5**	**0.70**
k-l (C_L=0.15, C□=0.01)	**UrbaWind**	**1.5**	**0.60**
k-l (C_L=0.1, C□=0.01)	**UrbaWind**	**2**	**0.70**
Experiment	**Meng and Hibi**	**1.42**	**0.52**

2.2 Wind-induced pressure coefficients on buildings

For a simple volume with two openings, the cross wind flow rate can be deduced from the analytical Aynsley formula (1). When more than two openings are connected with an indoor volume, the ACH computation needs to use the iterative process. The internal pressure is not known and the flow rate through openings are solved with a Newton-Raphson iterative method (Fhassis *et al.* 2010). In the case of a flat with several rooms, the free aerodynamic area of the opening *k* is replaced by an equivalent aerodynamic surface, taking into account the door aerodynamic surface A_{door} of the secondary room (Sanquer *et al.*, 2011).

$$A_k^{eq} = 1/\sqrt{\frac{1}{A_k^2} + \frac{1}{A_{door}^2}} \tag{4}$$

All the macrospcopic methods use as an input the external pressure fields on the walls where the openings are fitted. It becomes crucial to evaluate with precision the pressure field even in a complex urban area.

Before using a CFD tool to provide pressure value, comparisons were made with experimental data on Silsoe cube (Richards et al., 2007) resumed by Costola et al. (2009). This is valuable round robin test to check the ability of CFD to capture the flow separation around a cube. The pressure field obtained along the trajectory 0-1-2-3 with UrbaWind matches well the experimental results obtained for a detached low-rise building (figure 3). The wind speed reference level is the building height.

These value are in accordance to those given in the AIVC database for pressure coefficient on low-rise buildings up to 3 storeys (Liddament, 1986).

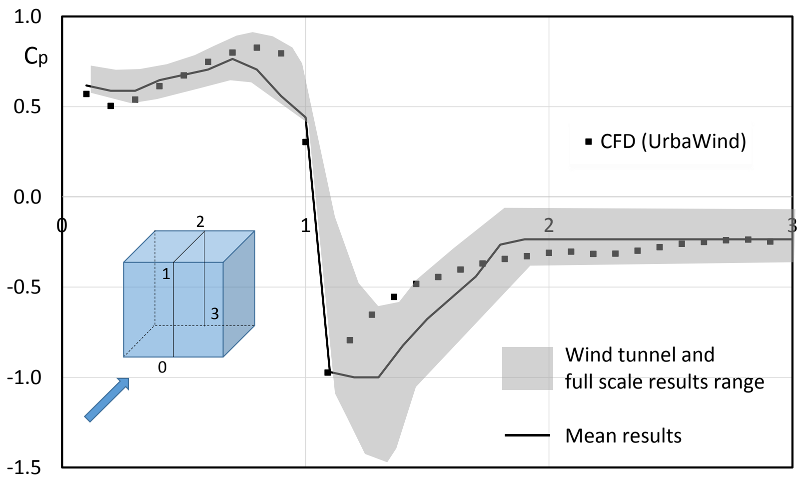


Figure 3 Pressure on the walls of a low-rise detached building: comparison of different wind tunnel experiments and numerical results (UrbaWind)

3 PRESSURE FIELD ON A BUILDINGS WALLS IN A COMPLEX URBAN AREA

Further computations were carried out for the same low-rise building located in a more complex urban environment (figure 4). Various wind incidences were computed.

The potential of natural ventilation defined as the maximum differential of pressure coefficient, is deduced from the curves on figure 4. For example □Cp decreases from unity (detached, 0°) to less than 0.1 (urban, 0°). For oblique wind incidence (45°), □Cp vanishes from 0.7 to less than 0.2. AIVC table gives □Cp=0.45 at 0° and 0.35 at 45° for urban configuration where surrounding buildings height is equal to the reference building height. By using (1), it may be easily shown that the gap on air change rate reach 100% because □Cp is overestimated by twice by using □Cp from tables rather than data from more complex assessment method. Influence of surrounded buildings should not be considered only as a modification of the roughness that changes the wind profile instead of considering the local influence of buildings on the wind.
Figure 4 shows also that the potential of natural ventilation depends on the position of the buildings in the urban layout. Buildings H, M and L have respectively high, medium and low level for Natural Ventilation. Although analytical formulae exist for this urban layout, it may be difficult to introduce the position of the buildings and the buildings's geometries as inputs of the formulae to reflect the sheltering effects.

This analysis shows the difficulties to extract Cp from numerical tables or analytic relations in order to describe the influence of urban environment since the signature is the wind aerodynamic interactions with all neighbour buildings.

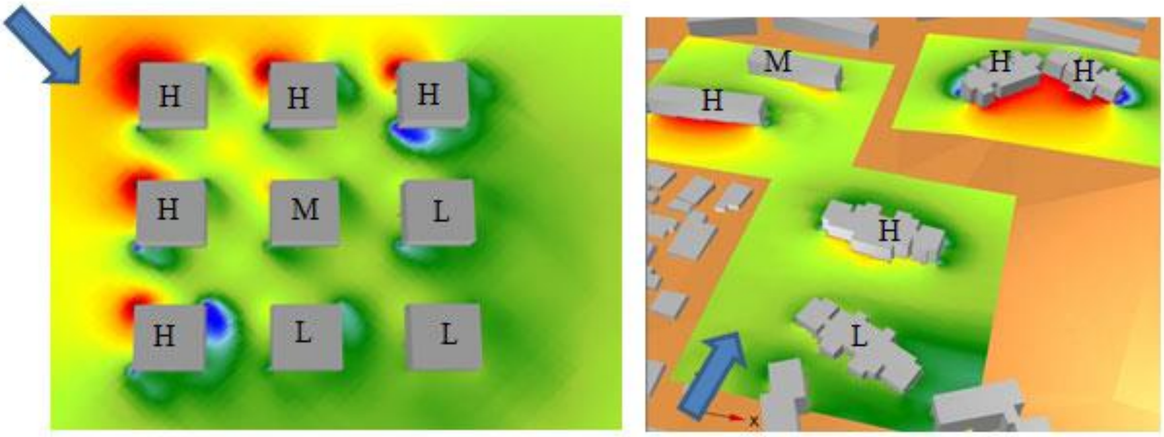


Figure 4 Example of CFD computation in complex urban area: mappings of pressure field

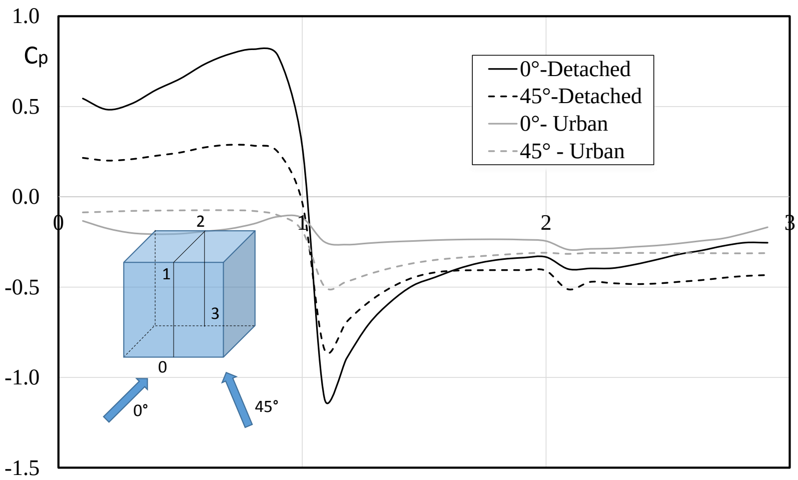


Figure 5 Pressure on the walls of a low-rise building Influence of the urban environment

On figure 5, the pressure field depends clearly on the shape of the buildings. Strong inhomogeneity of pressure field along the side wall should be considered in order to assess the scenario of the indoor ventilation. Figure 6 represents 4 screenshots for 2 similar shapes of a squared low rise building. We can observe that a slight change of building shape close to the corner can lead to a pressure field change because of a modification of the position of the flow separation. For instance for a wind incidence in front of the façade, the pressure coefficient close to the corner (in the red circle) becomes close to zero by changing the corner shape compared to a value close to +0.5. For a wind incidence of 45°, the influence of the shape is not too strong. Pressure field does not depend on the corner shape in this wind direction.

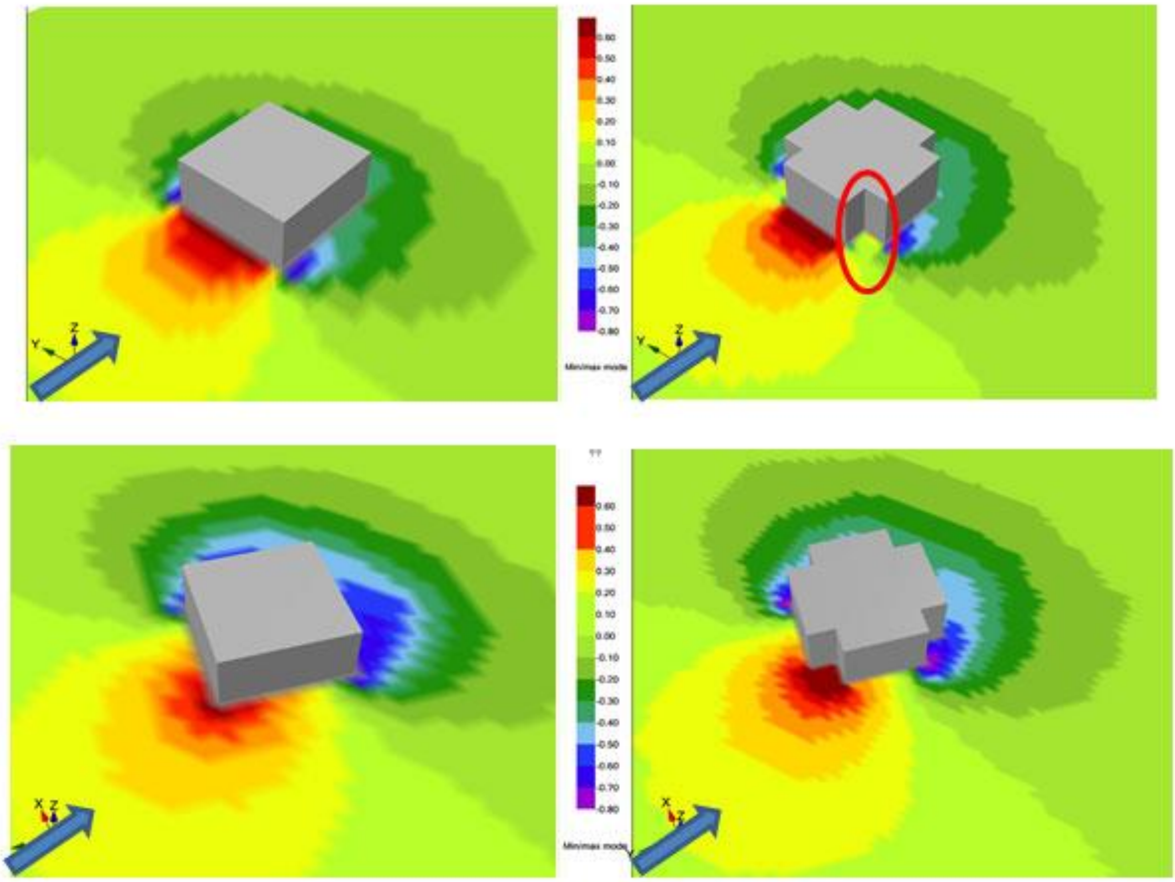


Figure 6 mappings of pressure coefficients at half height above the ground

Left : Silsoe Cube; Right : Cube with modified corners

Top : wind direction 0°; Bottom : wind direction 45°

This analysis shows the difficulties to extract Cp from numerical tables or analytic relations in order to describe the influence of every kind of buildings shape for various wind directions. Pressure field is strongly dependent on the flow separation patterns that cannot be summarized with precision with simple analytical formula.

4 CONCLUSIONS

The paper presents a numerical methodology to compute the wind-induced pressure coefficients on buildings in order to assess the natural ventilation in any urban area. UrbaWind, an automatic computational fluid dynamics code, was developed to model the wind in urban environments by optimizing the meshing and solving processes. Accurate results computed as quickly as possible will be useful for urban designers.
The turbulence modelling, namely the dependance of turbulence length on the distance from wall and the C□ value of the k-l model were calibrated in order to well reproduce the flow separation around builings walls. Numerical results were compared with well documented

experiments: Siloe cube as a low-rise building and a 1:1:2 tower as high-rise buildings. Comparisons were made on separation patterns and on pressure field on walls.
Some real examples are presented to show the advantages of the numerical methodology versus the analytical approach in order to extract pressure coefficients in buildings from numerical simulation CFD as inputs for thermal approach.

5 REFERENCES

Aynsley R.M, Melbourne. W., Vickery B.J. (1977), *Architectural aerodynamics*, applied science Publishers London

Clarke J, Hand J, Strachan P (1990) *A building and plant energy simulation system. Energy Simulatio*n Resarch Unit, Department of Mechanical Engineering, University of Strathclyde.

Liddament, M. W. (1986), *Air Infiltration Calculation Techniques*, An Applications Guide, AIVC.

Grosso M (1992) *Wind pressure distribution around buildings: A Parametrical Model,* Energy and Building, 18: pp 101-131.

Muehleisen R.T., Patrizi S. (2013) *A new parametric equation for the wind pressure coefficient for low-rise buildings*, Energy and Building, 57: pp 245-247.

Costola D., Blocken B., Hensen J.L.M (2009), *Overview of pressure coefficient data in building energy simulation and airflow network programs*, Building and Environment, 44, pp 2027-2036

Tominaga, Y., Mochida, A., Murakami S., Sawaki S. (2008) *Comparison of various revised k-ε models and LES applied to flow around a high-rise buildings model with 1:1:2 shape placed within the surface boundary layer*. Journal of Wind Engineering and Industrial Aerodynamics, 96, pp 389-411

Ferry M. (2002) *New features of the MIGAL solver*, in: Proceedings. Phoenics Users International Conference, Mos-cow, Sept 2002

Garratt J.R. (1992) *The atmospheric boundary layer*, Cambridge Atmospheric and space sciences series.

Hurley P.J. (1997) *An evaluation of several turbulence schemes for the prediction of mean and turbulent fields in complex terrain,* Boundary-Layer Meteorology 83(1)

Meng T., Hibi K. (1998). *Turbulent measurements of the flow field around a high-rise buildings*. Journal of Wind Engineering, Jpn, 76, pp 55-64

Fahssis K., Dupont G., Sanquer S, Leyronnas P. (2010). *Integration of the natural cross ventilation in the CFD software UrbaWind*, International Conference of Applied Energy, Conference paper, April 2010, Singapore

Sanquer S., Caniot G., Li W., Delaunay D. (2011), *A combined CFD-Network method for the cross-ventilation assessment in buildings,* 13th International Conference on Wind Engineering, Amsterdam

Richards P.J., Hoxey R.P., Connel B.D., Lander D.P (2009), *Wind tunnel modelling of the Silsoe cube*, Journal of wind engineering & industrial aerodynamics, 95, pp 1384-1399

FIELD APPLICATION OF ENHANCED DISPLACEMENT VENTILATION SYSTEM IN AN OFFICE OF A ZERO ENERGY BUILDING IN THE TROPICS

David K.W. Cheong[1], and Samantha C.H. Khor[1]

1 Department of Building
National University of Singapore
4 Architecture Drive
Singapore (117566)
Singapore
**Corresponding author: bdgckw@nus.edu.sg*

ABSTRACT

Conventional Displacement Ventilation (DV) system has been installed in an office of a Zero Energy Building (ZEB). Enhanced DV (EDV) system, consisting of fans mounted to the chair, which has been demonstrated in laboratory and field environmental chamber studies earlier was implemented for the first time in a full-scale office environment to assess its effectiveness of improving the thermal sensation of the occupants. Objective measurements and subjective assessments were conducted in the office with 12 occupants over a period of 2 weeks. Results show that the EDV is capable of improving the thermal comfort and sensation, and the personalised control feature can lead to higher occupants' thermal satisfaction. In conclusion, EDV system has been successfully implemented in the field with no issue of draft as tropically acclimatized subjects prefers higher air movement.

KEYWORDS

Enhanced displacement ventilation, Tropics, Thermal sensation, Air movement, questionnaire

1 INTRODUCTION

Displacement ventilation cooling uses a low-velocity (0.15 to 0.2 m/s) stream of moderately cooled air introduced via diffusers located through a raised floor (Hamilton, 2004). The cool supply air slowly rises as it pick up heat from heat sources. These heat sources create upward convective flows in the form of thermal plumes which create better IAQ in the space (Yu et al., 2004). The contaminated warm air then rises towards the ceiling and exhausted from the space. As such, DV system helps to maintain a better indoor environment, which enhances comfort and contributing towards the overall sustainability of the building. In the last few years, its application has been extended to classrooms, offices and other commercial spaces (Nishioka et al., 2000). There have been extensive researches on the performance of the DV system in temperate countries but very few in the Tropics. As Singapore aims to have at least 80% of the buildings to attain the minimum Green mark standards by 2030 (Building and

Construction Authority, 2010), it is necessary to adopt a more energy efficient air-conditioning and mechanical ventilation system (ACMV). The application of DV system was seen extended to the office of zero energy building in Singapore.

However, DV system works best in a space with high ceiling to floor ratio, such as in industrial applications or public halls. There are some limitations when used in offices as temperature gradient will be generated in the space. Hence, novel enhanced displacement ventilation (EDV) system was first implemented by Sun, (2010), with the intention to enhance convective cooling around the occupants in normal office served by the DV system. Li (2010) improvised on the work of Sun (2010) and developed a 2-fan EDV system. This study aims to assess the performance of the EDV system in a full-scale study of a relatively lower floor-to-ceiling height office served by DV system with respect to thermal comfort and thermal sensation.

2 EXPERIMENTAL SET-UP

Figure 1 shows the office, 18(L) x 18(W) x 2.7m (H), in a ZEB served by conventional DV system having 12 occupants with feedback of thermal dissatisfaction based on a survey conducted previously. The conditioned air is supplied to the room via circular floor diffusers on the raised floor to maintain air temperature of between 23.5 and 24.5degC. The EDV chair, as shown in Figure 2, is to replace the existing chair at four locations where the level of thermal comfort was of higher level of dissatisfaction.

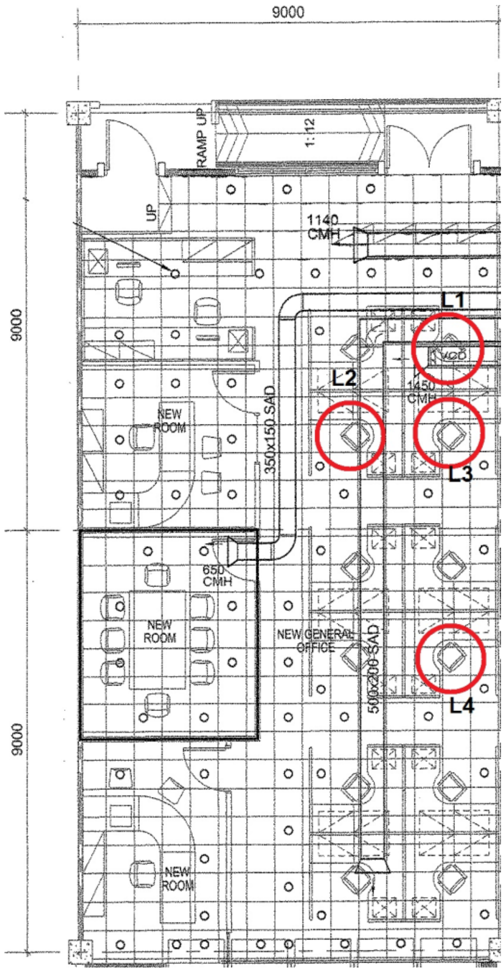


Figure 1: Layout of the office with the 4 EDV systems

Figure 2: The EDV chair

2.1 Objective Measurements

Objective measurements and subjective assessments were conducted between 9th and 20th December. Continuous measurements of air temperature and relative humidity using Hobo data loggers were taken between 9th and 15th Dec with DV system only and between16th and 20th December with the EDV system. In addition, spot measurements of air velocity and air temperature at different height using omni-directional anemometer system were taken on 13th and 17th in which both indoor and outdoor on these two days having similar condition. A detailed description of each measuring location is stated as follows and shown in Figure 3:

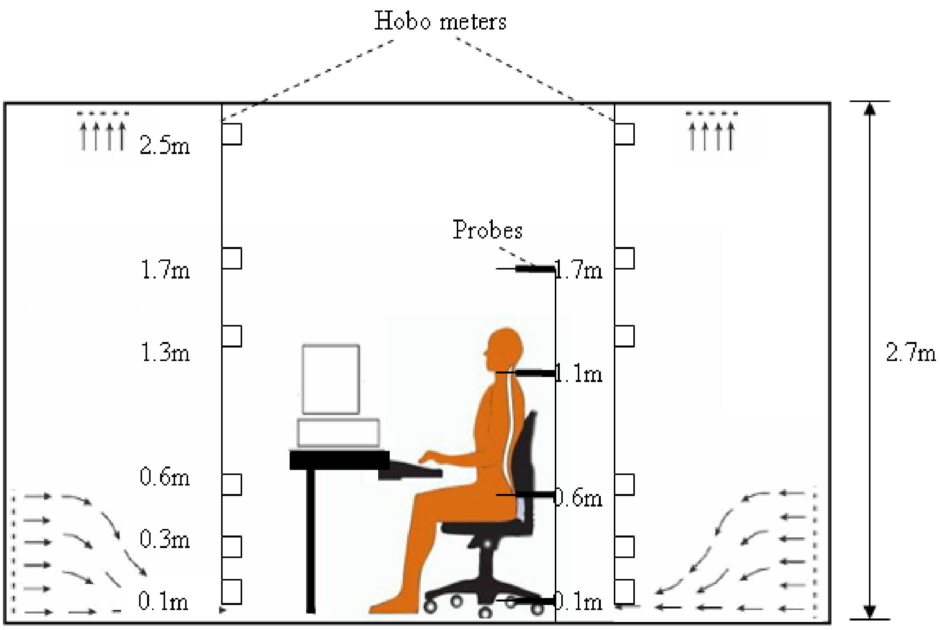


Figure 3: Locations of measurement on vertical plane

- Room air temperature (Tr) and RH measuring locations were measured at locations L1, L2, L3 and L4 with HOBO® meters. The HOBO® meters were placed at five heights: 0.1 m, 0.3 m, 0.6 m, 1.3 m and 1.7m.

- Air temperature (T), mean velocity (V), and turbulence intensity level (Tu) in the region around the occupants (locations L1, L2, L3 and L4) were measured with omni-directional anemometer. The probes were positioned at four heights, 0.1 m, 0.6 m, 1.1 m and 1.7 m respectively. The probe at 0.6 m height was close to the waist region of the subjects and the probe at 1.1 m height was right above the shoulder.

2.2 Subjective Assessment

The main objective of the questionnaire survey was to assess if the EDV system improves the thermal comfort of occupants. The ASHRAE scale, (−3) cold, (−2) cool, (−1) slightly cool, (0) neutral, (+1) slightly warm, (+2) warm and (+3) hot, was used for the assessment of subjects' Overall thermal sensation (OTS) and local thermal sensation (LTS) of 14 specific body parts (head, chest, back, stomach, right and left arms, hands, thighs, calves and feet). The overall thermal comfort (OTC) and local thermal comfort (LTC) of body segments was assessed using the Bedford's scale, (−3) much too cold, (−2) too cold, (−1) comfortable cool, (0) neither warm nor cool, (+1) slightly warm, (+2) too hot and (+3) much too hot.

3 RESULTS AND DISCUSSION

The average temperature measured in the office was between 24 and 25degC with relative humidity of about 60% over the period of measurement. Temperature profiles on 13^{th} and 17^{th} Dec where spot measurements were carried out for DV and EDV systems are similar. This provides a basis for fair comparison between them. The most preferred fan speed amongst the four EDV systems with fan discharge angle pre-adjusted at 30^{o} to the horizontal plane was "medium". Feedback from subjects shows that 'high' fan speed leads to draught sensation in areas like the waist, neck and facial region. Vertical temperature profiles at the four locations are similar for DV and EDV systems. Figure 4 showing a typical temperature profile at location L1. It shows that the temperature profile became steeper with the EDV system (with fan) between 0.6m and 1.7m as the fan is mounted at 0.6m. It demonstrated the effectiveness of the fan in bringing cool air from the DV diffuser up to subjects.

Figure 4: Temperature profile

Subjects were exposed to higher velocity at 0.6m as shown in Figure 5. The fans help to enhance the convective airflow up to a limited height given their capacity.

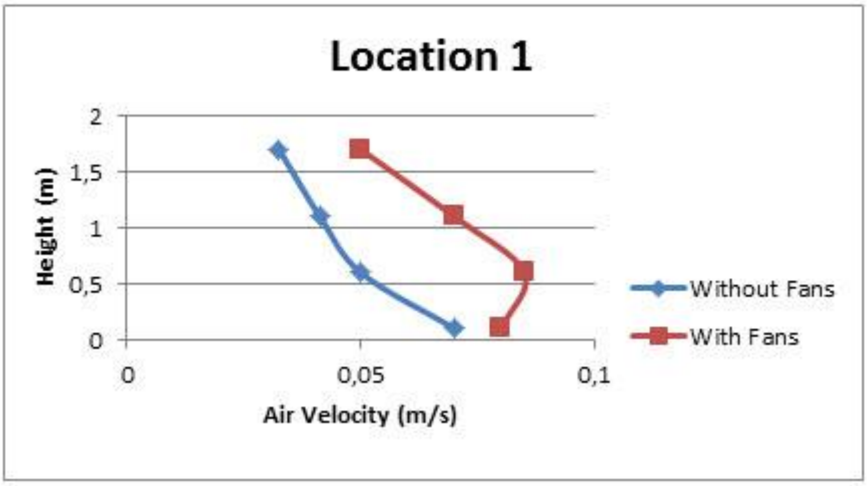


Figure 5: Air velocity profile

Figure 6 shows the average overall thermal sensations of the subjects. The average ambient temperature recorded in the office was about 24.5degC when the questionnaires were being completed. It is observed that the subjects felt slightly warm without the fans (0.75). However, subjects experienced close to comfortably cool (-0.25) and neutral condition when the EDV systems (Fans on) are used. Hence, it provides better thermal sensation to subjects.

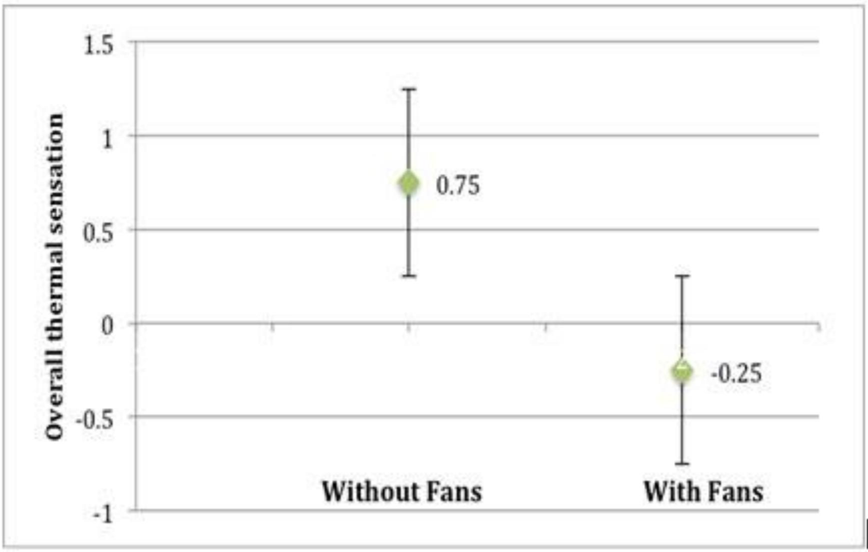


Figure 6: Overall thermal sensation

Figure 7 shows the local thermal sensation of different body segments at ambient air temperature of about 24.5degC before and after the introduction of EDV system. It demonstrates that the subjects would feel more comfortable with the EDV system at more body segments, i.e. closer to the average rating of 0. It was observed that the subjects did not experience a significant difference below the thighs when the fans were turned on. This is obvious given that the fans are mounted near the thigh region and therefore would not affect the thermal sensation of lower body segments such as the calves and feet. On the other hand, the right/left hand regions experienced the coolest thermal sensation (-0.75), followed by right arm/left arm (-0.5) after the introduction of the EDV system. It was also observed that without the fans, the upper body segments such as the torso, chest and shoulder have warmer thermal sensations as compared to with fans. The fans help to bring cool air upwards and thus provide convective cooling around the upper body segments.

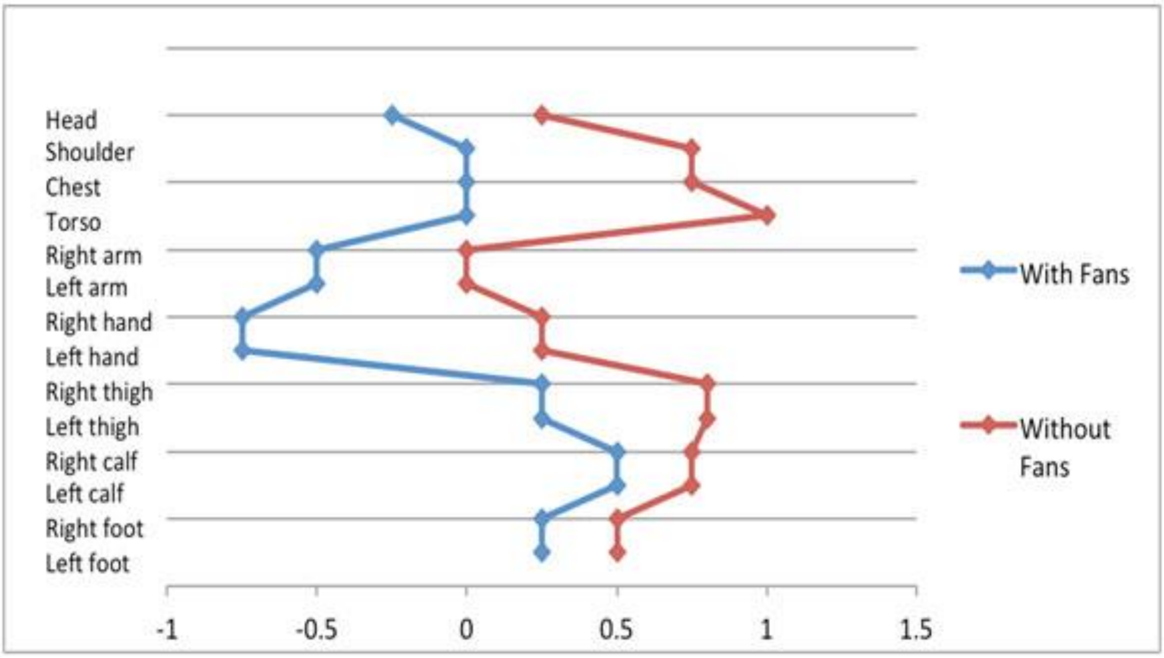


Figure 7: Local thermal sensation of body segments

4 CONCLUSIONS

75% of the subjects felt that the EDV chair is effective in providing better thermal comfort sensation especially when ambient temperature in the office was high. This concurs with the vertical temperature and air velocity profiles. Subjects experienced more air movement when the EDV system is in operation. Hence, the EDV system is successfully applied in the field.

5 REFERENCES

Building and Construction Authority (2010). *Existing building retrofit*. The Centre for Sustainable Buildings and Construction, Building and Construction Authority.

Hamilton, S. D. (2004). *Displacement ventilation*. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

Li, Q. (2010). *Assessment on the performance of the enhanced displacement ventilation system in the tropics.* (Unpublished master's thesis), National University of Singapore.

Nishioka, T., Ohtaka, K., Hashimoto, N., et al. (2000). Measurement and evaluation of the indoor thermal environment in a large domed stadium. *Energy and Buildings*, **32**, 217-223.

Sun, W. M. (2010). *Study of enhanced displacement ventilation in office environment in the tropics.* (Unpublished PhD thesis), National University of Singapore.

Yu, W. J., Cheong, K. W. D., Tham, K. W., et al. (2007). Local thermal sensation and comfort study in a field environment chamber served by displacement ventilation system in the tropics. *Building and Environment*, **42**(2), 525-533.

ECODESIGN DIRECTIVE FOR VENTILATION UNITS

Claus Händel

EVIA European Ventilation Industry Association
Avenue des Arts 46 Kunstlaan
1000 Brussels
Tel: +32 (0) 2 7327040

Claus Händel
Technical Secretary
Mobile:+49 177 2766379
E-Mail: claus.haendel@evia.eu

Note: the contact addresses may be re-arranged

1 INTRODUCTION

Ventilation technologies are the key aspects to reach the target of nearly zero energy buildings. From 1st January 2016 on ventilation units have to comply with minimal energy efficiency criteria according EU Regulation No 1253/2014 of 7 July 2014 implementing Directive 2009/125/EC of the European Parliament and of the Council. Furthermore, residential ventilation units will have an energy label according EU Regulation No 1254/2014 of 11 July.

This is a logical path for the EU-Commission because the expected energy savings potential of ventilation products is high. Expected energy savings of fans and ventilation units might reach 540 TWh primary energy and this is equivalent to 2-3% of EU 28 total energy consumption (Table 1).

Table 1: Expected energy savings with Ventilation Products (Taget 2020)

	End Energy	**Primary Energy**
Residential Heating Energy	222 TWh	244 TWh
Non Residential Heating Energy	150 TWh	165 TWh
Non Residential Cooling Energy	8 TWh	8 TWh
Non Residential Electricity	16 TWh	40 TWh
Total Fans covered by EU 327/2011	34 TWh	82 TWh
Total		539 TWh
		~ 2-3% ot Total EU28

Considering that the Ventilation market for Non-Residential buildings is quite developed, EVIA sees further potential in developing the market for mechanical ventilation systems in existing buildings. In EU28 only approx.. 16% of residential buildings have mechanical ventilation systems, less than 1% have heat recovery (Figure 1).

2 ENERGY LABELLING FOR RESIDENTIAL VENTILATION UNITS

From 1st January 2016 on all residential ventilation units will get a label with class A+ to G. It will be a common label for all types (bidirectional, unidirectional, heat recovery and demand controlled). The so called Seasonal Energy Consumption (SEC) is a value considering the primary heating energy savings minus the primary electrical energy demand. The calculation depends on the following parameters:

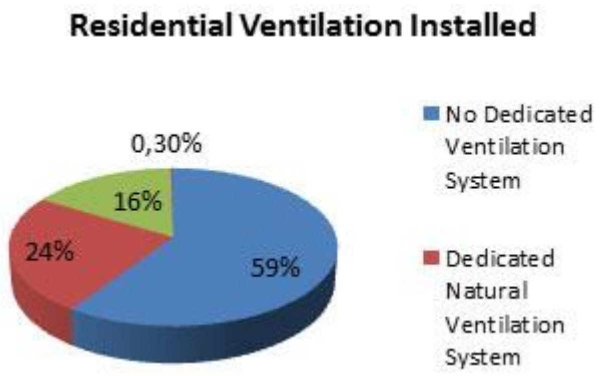
- Specific Power Input (SPI) depending on fan and controls electrical demand
- Heat recovery efficiency (if equipped with)
- Controls options (CTRL)

Consumer organisations are in favour of a common label, but there are also some critical aspects on this, because the scaling (A+ to G) does not consider aspects like:

- Tightness of building envelop and the resulting infiltration
- The impact of climate zone to the benefit of heat recovery units (cold climates higher, warm climates less)

So especially in refurbishments and in warm climates, a suitable ventilation system has to be selected based on full energy calculations for the entire building including the technical building systems with EPBD calculation methods.

Figure 1: Residential Ventilation Systems in building stock



3 CONCLUSIONS

Labelling might have an impact on consumers decisions investing in ventilation technology, because most of the other Techncal Building Systems have a label (heating, cooling, lighting, etc.) and ventilation is playing in the same market.
Ecodesign will help to develop a more transparent market within Europe because the main performance data have to be delivered transparent in a common format based on common test method. Additional national certifications and restrictions will loose the impact and this will help to develop innovative and energy efficient products for whole Europe.

4 REFERENCES

COMMISSION REGULATION (EU) No 1253/2014 of 7 July 2014 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for ventilation units EU 1254/2014

COMMISSION DELEGATED REGULATION (EU) No 1254/2014 of 11 July 2014 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to energy labelling of residential ventilation units

Ecodesign Lot 10 Study and Supplementary Study, FGK, 2010

PRIMARY ENERGY USED IN CENTRALIZED AND DECENTRALIZED VENTILATION SYSTEMS MEASURED IN FIELD TESTS IN RESIDENTIAL BUILDINGS

Alexander Merzkirch[1*], Stefan Maas, Frank Scholzen, and Daniele Waldmann

[1] *University of Luxembourg*
Faculté des Sciences, de la Technologie et de la Communication
6, Rue Coudenhove Kalergi
L – 1359 Luxembourg
Presenting author: Prof. Dr.-Ing. Frank Scholzen
**Corresponding author: alexander.merzkirch@uni.lu*

ABSTRACT

Ventilation systems can save heat energy by using heat recovery, but consume electrical energy to power the fans. In practice, the energy efficiency of those systems can be lower than expected, when compared to the nominal values provided by the manufacturer. In this paper, results of a comprehensive field tests with 20 centralized and 60 decentralized ventilation systems for residential buildings and the calculation of the primary energy savings of those devices are presented. Factors like volume flow unbalances, shortcuts, temperature change rates and specific fan power have been addressed by tracer gas technology and other means and been used as input factors to calculate the primary energy balance of those devices. Every system showed positive primary energy savings. The mean value for centralized systems was 2.92 Wh/m^3 with a high standard deviation of 2.23 Wh/m^3, while the decentralized systems showed higher savings of around 4.75 Wh/m^3 with a standard deviation of 0.01 to 0.15 Wh/m^3. In general, the calculated savings in field tests were significantly lower compared to the case of using nominal values as input parameters.

KEYWORDS

Primary Energy, Residential Ventilation, Decentralized, Centralized, Field Test

1 INTRODUCTION

Mechanical ventilation in a residential building serves two main purposes. The first being a good air quality at a given ventilation rate, avoiding over-ventilation and the second being the reduction of the primary energy used to heat the building by making use of heat recovery devices. This paper focuses on the primary energy balance of a ventilation systems and the different ways to calculate it. Savings can only be achieved, if the input of primary energy to run the ventilation system is smaller than the savings, in other words, if the net-energy-balance is positive (Roulet, 2001; Heidt, 1998). Only a complete assessment of all relevant parameters like heat recovery rate, shortcuts, specific fan power, etc. can lead to realistic calculation of the primary energy savings (Merzkirch, 2015). This calculation is often done with nominal values, derived under laboratory conditions and according to data sheets by the manufacturer. However, these nominal values can often not be reached on-site (Manz, 2000; Boerstra, 2012). An overestimation of the savings and an underestimation of the electrical energy consumed, can lead to energy demand calculations which often vary clearly from the actual energy demand of buildings (Merzkirch, 2014).

2 OBJECTS & METHODS

2.1 Types of mechanical ventilation systems

In field tests, key parameters of 20 centralized systems, installed in single-family homes and 60 decentralized devices, installed in single- and multi-family homes were assessed. The decentralized systems, installed directly in the façade of the building, were of two different kinds. The first using a regenerative heat exchanger can only be installed in pairs. Each of them uses only one fan to deliver air into the volume. While device one is transporting fresh air from outside to inside, device two is extracting air from the inside which heats up the heat storage made out of aluminium or ceramic. Every 60 seconds (the cycle time depending on the device and manufacturer) the fans switch their direction and the heat stored from the outgoing air can heat up the incoming air. The second decentralized principle is often called "single room ventilation unit". Each unit can be seen as a small centralized system since it provides supply air and extract air using a recuperative cross counterflow heat exchanger to transfer heat. The centralized systems are equipped as usual with two fans, a counterflow or cross counterflow heat exchanger and a ductwork to deliver the air at its destination. All devices come with filters for extract and outside air.

Following parameters were measured in field: Supply and extract air flow and their unbalances, internal and external shortcuts, sensitivity to pressure (decentralized units), power consumption, temperature change rate. The measurement of the flow parameters and shortcuts was done by the use of tracer gas technology, which is a commonly used method to access airflows in ventilation units and buildings (Roulet, 2008; Manz, 2001; Sandberg, 1989). The mentioned parameters above were then used to calculate the primary energy savings, expressed in Wh/m^3. This value gives us the saved primary energy per transported m^3 air in comparison to the case of natural ventilation at the same air flow. The calculations were first done with nominal values by manufacturers and then with values derived in the field tests, making a comparison possible.

2.2 Primary Energy Savings

On the negative side of the primary energy balance we find the electrical energy consumed by the ventilation device to power the fans and the electronic controls. Knowing the electrical power (P) and the delivered air flow ($\dot{V}$), one can calculate the specific fan power (SFP), which tells us, how much energy is needed to transport one m^3 of air.

$$SFP = P / \dot{V} \quad [Wh/m^3] \qquad (1)$$

Modern ventilation system show nominal SFP values between 0.15 [Wh/m^3] and 0.4 [Wh/m^3] (Heil, 2011, Roulet 2008). In practice, values can be higher due to high pressure losses or malfunctions (Merzkirch, 2015a; Roulet 2001).

For the measurement of the airflow $\dot{V}$, we have to consider, that possibly a part of the extract and/or exhaust air is recirculated into the supply air. The amount of fresh air is then reduced. By tracer gas measurements, this recirculation (R) was measured. For example, if 10 percent of the outgoing air was recirculated to the supply air, then value for $\dot{V}$ was also reduced by 10 percent and as a result, the SFP increases. According to manufacturer data, the internal recirculation inside the ventilation device is usually below 1 %. In field tests 17 of 20 centralized systems showed internal circulation below 1 %. The total circulation however, which consists of internal and external recirculation, which can happen within the ductwork or outside the building between outlet and inlet, was higher with a mean value of 6,5 & and a standard deviation of 12,5 % (Merzkirch, 2015a). Values in this range were also measured in other tests (Roulet, 2001; Manz, 2001) .

On the positive side of the balance, we find the savings of ventilation losses Φ_V by making use of a heat exchanger.

$$\Phi_V = \dot{V} \cdot \rho \cdot c_p \cdot \Delta T \cdot \eta_{HR} \qquad [\text{Wh/m}^3] \tag{2}$$

With:

ρ densitiy of air [kg/m^3]
c_p heat capacity of air [kJ/kgK]
ΔT temperature difference between inside and outside [K]
η_{HR} efficiency of heat recovery

The density of air ρ is assumed with 1.204 kg/m^3, the specific heat capacity c_p with 1.005 kJ/(kgK). For ΔT, a mean value of 15 K was assumed. This means, that the average temperature difference in the heating period is 15 K. Therefore, the calculation of the primary energy savings counts for the energy savings on an average winter day in comparison to a natural ventilation without heat recovery.

The efficiency of the heat exchanger η_{HR} can be described by the relation of the temperature differences of exhaust/extract and exhaust/outside air. The so called temperature change rate, here called η_{real}, can be calculated as follows:

$$\eta_{real} = \frac{T_{extract} - T_{exhaust}}{T_{extract} - T_{outside}} \tag{3}$$

Of course, this relationship is a simplification, since it neglects possible enthalpy change and condensation effects. Furthermore, unbalances between supply and extract air flows can lead to over- or underestimation of the efficiency. For a more detailed discussion, see (Merzkirch, 2015a). Nominal values for η_{HR} are always based on measurements under laboratory conditions with balanced air flows. Here, values between 0.8 and 0.95 are achieved by many devices (Heil, 2011, Roulet 2008). In practice however, the measured temperature change rate is often lower with mean values around 0.58 to 0.7 (Merzkirch, 2015a; Roulet 2001).

In order to gain the primary energy instead of the end energy, where production and transportation of the energy is not included, primary energy factors are used. For electricity, we assume a factor of 2.7 ($f_{p,el}$)and for heat a factor of 1.1 ($f_{p,h}$). The production and distribution of heat on-site usually encounters losses, which we assume with 1,25 (f_h). Furthermore, a part of the heat generated by the fans is recovered in the heat exchanger. For this calculations we assume a value of 0,25 (f_{fan}).

The primary energy savings can then be calculated:

$$PES = \rho \cdot c_p \cdot \Delta T \cdot \eta_{HR} \cdot f_h \cdot f_{p,h} - SFP \cdot f_{p,el} + \phi_V \cdot f_{fan} \cdot f_w \cdot f_{p,w} \qquad [\mathrm{Wh/m^3}] \qquad (4)$$

3 RESULTS

In field tests of 60 decentralized and 20 centralized system, following mean values and standard deviations of the relevant parameter were obtained:

Table 1: Results of field tests – parameters relevant for energy efficiency of mechanical ventilation system.

	centralized	decentralized (regenerative)	decentralized (recuperative)	nominal
Specific Fan Power (SFP) [Wh/m³]	0.475 ± 0.37	0.23 ± 0.02	0.22 ± 0.01	0.15 to 0.4
Total Shortcut [%]	6.5 ± 12.5	-	13 ± 6.2	< 1
Temperature Change Rate [%]	59 ± 25	76 ± 5	80 ± 4	80 to 95

As can be seen in Table 1, the measured mean values are well below the nominal values, provided by the manufacturers. Thus, an assumption of those values, when calculating the energy demand of a building and its HVAC technology, can lead to deviations between calculation and consumption. These values are the basis for the calculation of the primary energy savings:

Table 2: Primary energy savings of 20 centralized and 60 decentralized devices, based on measured parameters according to Table 1.

	centralized	decentralized (regenerative)	decentralized (recuperative)
PES_{real} [Wh]	2.92 ± 2.23	4.67 ± 0.01	4.8 ± 0.15
$PES_{nominal}$ [Wh/m³]	5.18 ± 0.98	5.6 ± 0.01	5.4 ± 0.01

The centralized devices did not reach the nominal PES value of 5.2 Wh/m^3, but showed significantly lower values around 3 Wh/m^3. This is due to higher specific fan power values, shortcuts and lower heat exchange efficiencies, than the nominal values would let expect. But, according to the high standard deviation, there are systems, which show very high PES values above 5 Wh/m^3 and very bad systems below 1 Wh/m^3. The range is quite big and is a sign for the extraordinary importance of a thorough installation, commissioning and maintenance of such a centralized system. The users, living in a building where the systems showed low values, did not recognize the low performance of the mechanical ventilation system at all, despite of also lower indoor air quality with CO_2 values above 2500 ppm due to high shortcuts or malfunctioning fans, resulting in lower fresh air flows.

The decentralized system showed values around 4.7 to 4.8 Wh/m^3, which is closer to the nominal values of around 5.5 Wh/m^3. The simple reason for that is, that during the installation, commissioning and maintenance phase, there is less to go possibly wrong. There is no complex ductwork, no valves and the devices are easy to install and operate. However, one has to consider the higher sensitivity to pressure differences of decentralized devices. Even small differential pressure on the façade around 2.5 to 10 Pascal, which can easily be induced by wind or stack effect, can lead to serious deviations between supply and exhaust of 25 to 100 % (Merzkirch, 2015a; Manz, 2000). Such unbalances would lead to a significant decrease of heat recovery efficiency and primary energy savings. Long term measurements would be necessary to further analyse this effect.

4 CONCLUSION

In the previous chapters, the results of a comprehensive field test of 20 centralized and 60 decentralized mechanical ventilation systems in residential buildings in Luxembourg were presented. Concerning the energy efficiency of the devices, the specific fan power, shortcuts and temperature change rates were presented, leading to the calculation of the primary energy savings of the devices compared to the case of natural ventilation without heat recovery.

Concerning the specific fan power, the decentralized systems showed half the specific power around of the centralized ones. Many devices showed high volume unbalances which lead to low temperature change rates, especially for the centralized systems. But also the decentralized systems showed high sensitivity to wind and as a result, lower temperature change rates than expected. But, ventilation systems can reach high heat recovery values of 90 % if they are carefully planned, installed and operated. In almost all systems, shortcuts inside and outside the ventilation device were measured in tracer gas measurements. Shortcuts lead to a decrease of the fresh air flow and to an increase of the specific fan power.

The energy balance of those devices consists of the heat energy savings by using heat recovery on the one hand and the use of electrical energy to power the fans on the other hand. The differential of those two sides, finally gives us the primary energy savings in Wh by transported m^3 of air, on an average day during the heat period, where the temperature difference between inside and outside is 15 Kelvin. This of course, is only in comparison to the case of natural ventilation without any heat recovery.
Calculations show, that every system shows a positive primary energy balance. For centralized systems, the primary energy savings were 2.92 Wh/m^3 with a high standard deviation of 2.23. This makes clear, how important a careful planning, installation and operation is.
Decentralized systems showed higher savings of around 4.7 to 4.8 Wh/m^3 with lower standard deviations of 0.01 to 0.15. The in theory lower fan efficiency and lower heat recovery efficiency of decentralized devices do not lead to lower overall energy efficiency compared to centralized systems. Furthermore, a low specific fan power due to the lack of ductwork, easy installation and operation lead to higher mean energy efficiency than centralized systems. However, decentralized systems often show higher noise levels which can lead to unsatisfied users. They also are limited in maximum volume flow at bearable noise levels which can easily lead to an under supply of fresh air in critical situations like two persons in a bedroom of high air tightness (Merzkirch, 2015a; Manz, 2000).

However, during summertime or in times of low temperature differences between inside and outside of less than 4 Kelvin, the operation of the systems costs more energy than it is able to save (Merzkirch, 2015a). In those times, a natural ventilation is preferable. If the user is able to ventilate manually, the mechanical ventilation system could be switched off. A demand driven system would be able to automatically react to window opening and stop operating if the indoor air quality is sufficient due to low concentration of carbon dioxide (CO_2) or volatile organic compounds (VOC) (Fisk, De Almeida, 1998; Fan et al, 2014; Merzkirch, 2015b). The user can act as he pleases and the ventilation system adapts to his behaviour.

For the future, a more profound planning and installation of mechanical ventilation systems is critical if we want to lower the primary energy consumption of those systems. This especially holds true for centralized systems, were many mistakes are made during installation and hydraulic balancing. Decentralized units can play an important role in cases, where their air flow and noise comfort is sufficient. The lack of any ductwork can lead to very low specific fan power. In times, when low temperature differences between inside and outside lead to low or even negative primary energy savings, a natural ventilation, should be preferred over a mechanical one if possible and reasonable.

5 ACKNOWLEDGEMENTS

The authors acknowledge the high value contribution of the Ministry of Economy, Luxembourg and the Société Nationale des Habitations à Bon Marché (S.N.H.B.M.) which assisted with personnel, material, installations and gave valuable technical knowledge and assessment. Supported by the Fonds National de la Recherche, Luxembourg (PhD-FSTC-2015-05).

6 REFERENCES

Boerstra, A., Balvers, J., Bogers, R., Jongeneel, R., van Dijken, F. (2012): Residential ventilation system performance: Outcomes of a field study in the Netherlands. *Proceedings of 33rd AIVC conference*, Copenhagen, Denmark.

Cheong, K. (1998): Tracer gas technology for airow measurements in hvac systems. *International Journal of Energy Research,* Volume 19, Pages 1081-1093.

Fisk, W.J.; De Almeida, A.T. (1998): Sensor-based demand controlled ventilation: a review. *Energy and Buildings,* Volume 29, Pages 35-45.

Fan, Y.; Kameishi, K.; Onishi, S.; Ito K. (2014): Field-based study on the energy-savings of CO_2 demand controlled ventilation in an office with application of energy recovery ventilators. *Energy and Buildings,* Volume 68, Pages 412-422.

Heidt, F.D.; Fischer, T.; Thiemann, T. (1998): Energetische Beurteilung dezentraler Raumlüftungsgeräte mit Wärmerückgewinnung — Methoden und Beispiele. *Heizung-Lüftung/Klima-Haustechnik*, Volume 49, Issue 11, Pages 52–62.

Heil, E. (2011): Wohnungslueftung – frei und ventilatorgestuetzt. Beuth Verlag, 2nd edition, Berlin.

Manz, H.; Huber, H.; Schlin, H.; Weber, A.; Ferrazzini, M.; Studer, M. (2000): Performance of single room ventilation units with recuperative or regenerative heat recovery. *Energy and Buildings*, Volume 31 Issue 1, Pages 37-47.

Manz, H., Huber, H., Helfenfinger, D. (2001): Impact of air leakages and short circuits in ventilation units with heat recovery on ventilation efficiency and energy requirements for heating, *Energy and Buildings*, Volume 33, Pages 133-139.

Merzkirch, A. (2015a): Energieeffizienz, Nutzerkomfort und Kostenanalyse von Lüftungsanlagen in Wohngebäuden: Feldtests von neuen Anlagen und Vorstellung bedarfsgeführter Prototypen. *Dissertationsschrift*. Shaker Verlag, Berlin.

Merzkirch, A. (2015b): A semi-centralized, valveless and demand controlled ventilation system in comparison to other concepts in field tests. *Building and Environment*. In Press, Accepted Manuscript, available online 18 June 2015.

Merzkirch, A.; Hoos, T.; Maas, S.; Scholzen, F.; Waldmann, D. (2014): Wie genau sind unsere energiepaesse? vergleich zwischen berechneter und gemessener endenergie in 230 wohngebaeuden in luxemburg. *Bauphysik*, Volume 36, Pages 40-43.

Roulet, C. A.; Heidt, F. D.; Foradini, F.; Pibiri, M.-C. (2001): Real heat recovery with air handling units. *Energy & Buildings*, Volume 33, Issue 5, Pages 495–502.

Roulet, C. A (2008): Ventilation and airflow in buildings, Earth Scan, London, UK.

Sandberg, M.; Stymne, H. (1989): The constant tracer flow technique. *International Journal of Energy Research*, Volume 24, Pages 209-219.

SPREAD IN ENERGY USE IN BUILDINGS DEPENDENT ON CHOICE OF HEATING AND VENTILATION SYSTEM

Pär Johansson*[1] and Paula Wahlgren[1]

1 Chalmers University of Technology
Sven Hultins gata 8
412 96 Gothenburg, Sweden
**Corresponding author: par.johansson@chalmers.se*

ABSTRACT

The energy use in buildings is dependent on the choices made during the design, construction and renovation. The causes for these differences are, among others, caused by the behavior of the occupant of the building and the choice of heating and ventilation system. The European scheme of Energy Performance Certificates (EPCs) aims at reducing the energy use in the built environment. It is most common to calculate (i.e. not measure) the energy use for the buildings which are affected by the scheme. In Sweden, on the other hand, the EPCs are normally based on measurements of the actual energy use in the building. This makes comparisons between calculated energy use for building permit and measured energy for EPCs possible. In this paper a study of the differences between the calculated and measured energy use and its correlation with the choice of heating and ventilation system is presented. This is done by detailed investigations of the calculated and measured energy use in 44 buildings. For further analysis, a database of 1 753 buildings with measured energy use (EPCs) is used to study the dispersion in energy use for buildings with different heating and ventilation systems. Analysis using numerical simulations tools on human behavior has also been performed. The results of the investigation can be used to further improve the measured EPCs.

KEYWORDS

Energy use in buildings, Energy performance certificate, Occupancy behavior

1 INTRODUCTION

The energy use in buildings should be decreased within the European Union to reach the goal of a 20% overall reduction in greenhouse gases until 2020 (European Parliament, 2010). The required energy performance of buildings is specified in the national building codes. In Sweden, the first code was implemented in 1946 (IEA, 2013) and the first energy use requirements were introduced in 1975 after the oil crisis in 1973-1974. The requirements were specified with maximum U-values and demands on the airtightness for different building parts. The codes were developed during the following years, tightening the demands on the energy use. The codes have the same requirements for new developments and retrofitted buildings, and since 2006 the code is based on performance criteria (Boverket, 2015).

The European scheme of Energy Performance Certificates (EPCs) aims at reducing the energy use in the built environment. It is most common to calculate (i.e. not measure) the energy use for the buildings which are affected by the scheme. In Sweden, on the other hand, the EPCs are normally based on measurements of the actual energy use in the building. The measured energy use is corrected for the climate variability by using a reference year. The energy use should also be corrected for 'normal' usage. This makes comparisons between calculated energy use for building permit and measured energy for EPCs possible. This paper presents an investigation on the difference between the calculated and measured energy and its correlation with the choice of heating and ventilation system. The aim of the study is to investigate parameters that cause deviations between the calculated and measured energy use. This is done by investigations of the calculated and measured energy use in 44 buildings in Lerum municipality (in south west Sweden). For further analysis, a larger database of EPCs for single and multi-family houses has been analyzed. This database represents Västra Götaland (south west region) and includes 1 753 EPCs. Analysis using numerical simulations tools on human behavior has also been performed, using one of the buildings in Lerum municipality.

2 ENERGY USE IN BUILDINGS IN SWEDEN

The energy use in buildings varies depending on, for example, the choices made during the design, construction and renovation of the building. The final energy use for space heating and domestic hot water is on average 171 kWh/m^2/year in Swedish multi-family buildings from before 1980 heated with district heating, which should be compared to 144 kWh/m^2/year for buildings built after 1980 (Statistics Sweden, 2012). For single family houses the average energy use for space heating and domestic hot water has reduced from 165 kWh/m^2/year in 1995 to 126 kWh/m^2/year in 2011. This is a 24% energy use reduction. However, at the same time the domestic electricity use has increased by 6%, from 38 kWh/m^2/year to 41 kWh/m^2/year. A part of this increase can be explained by more use of electricity to power circulations pumps, ventilation and floor heating (Swedish Energy Agency, 2014). Other causes for the differences are the behavior of the occupants in the building and the choice of heating and ventilation system. In the following section statistics of the heating and ventilation systems of the Swedish single family houses are presented.

2.1 Heating system

The technologies available, cost and legal demands changes the settings when a building is designed. Figure 1 presents how the percentage of different heating system in single family houses in Sweden has changed between 1990, 1999 and 2010.

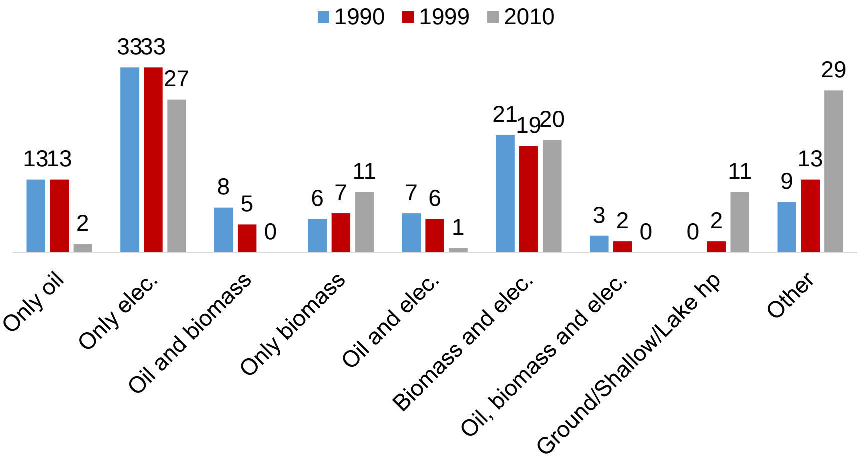


Figure 1: Percentage of different heating systems in Swedish single family houses in 1990, 1999 and 2010. The category 'Only electricity' also contains the houses with an air to air and air to water heat pump. The category 'Other'means combinations of heating systems that are not listed (Statistics Sweden, 2012).

It is clear that the number of houses with heating systems using only oil and electricity is declining and that it is more common in 2010 than in 1990 to use a variety of energy sources. In single family houses especially heat pumps have become very popular. In 2013 half of all single family houses had a heat pump. Air heat pumps are the most common heating system which accounts for 50% of the heat pumps. Ground, shallow and lake heat pumps were the second most common heat pump with 40% of the heat pump market (Swedish Energy Agency, 2014).

The space heating to the room can be supplied either by direct electricity, heating by air or hydronic heating. Hydronic heating is by far the most common supply system used in 70% of the single family houses. Direct electricity is used in around 25% of the houses and the remaining 5% are heated by air. The average indoor temperature is 21°C in the single family houses. Older houses have a 1°C lower average indoor temperature than newer houses (Boverket, 2010).

2.2 Ventilation system

Similarly to the heating systems, the ventilation systems in single family houses have changed during the years. In houses from before 1960, natural ventilation is used in 97% of the houses compared to 11% in the houses built in 1996-2005. Figure 2 shows the percentage of all single family houses that have natural ventilation, exhaust air, supply and exhaust air, supply and exhaust air with heat recovery and exhaust air with heat pump compared to the houses built in 1996-2005.

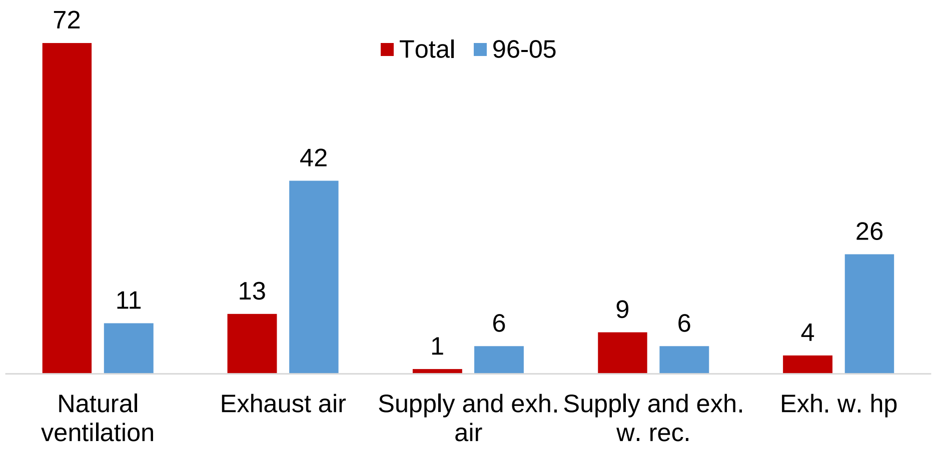


Figure 2: Percentage of different ventilation systems in all Swedish single family houses compared to the houses built 1996-2005 (Boverket, 2010).

The newer buildings has a larger percentage of exhaust air and exhaust air with heat pump compared to the total housing stock. This explains partly the increased electricity use in the buildings.

3 CALCULATED VS. MEASURED ENERGY USE IN LERUM MUNICIPALITY

Lerum municipality is located in south of Sweden close to Gothenburg on the west coast. The vision of Lerum municipality is to become Sweden's best performing municipality concerning the environment in 2025. One of the main parts of this vision is to stimulate buildings with less energy use. The energy advisors give advices to house owners on energy efficiency measures which leads to less energy use in existing buildings. To encourage construction of new buildings with low energy use, the politicians in 2010 decided to introduce a reduction in the fees for urban planning and building permits for low energy buildings in 2011. The municipality also has a lower maximum bought energy demand than what is defined in the national building code (BBR). The difference between the calculated and measured bought energy use for 44 single family houses in Lerum municipality is presented in Figure 3.

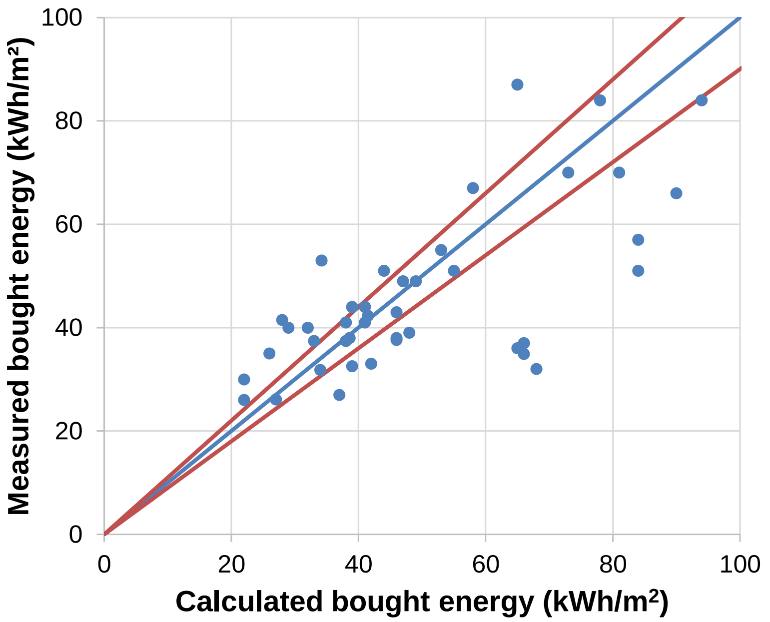


Figure 3: Calculated and measured bought energy use for 44 single family houses in Lerum municipality. The blue line indicates a perfect match between the calculated and measured energy use. The red lines indicate a deviation of 10% from the perfect match.

There are 29 houses of the 44 which have a larger difference between the calculation and measurement than 10%. Smaller than 10% is considered acceptable according to Sveby (2012). The average difference is 25% while the house with the largest difference has a 113% larger measured energy use than calculated. The sample is too small to make conclusions on the difference based on the choice of heating and ventilations system. Therefore a larger database of EPCs is used to study these influences.

4 VARIATIONS IN MEASURED ENERGY USE

The energy use is further studied with the EPC database containing 1 028 multi-family buildings and 725 single family houses from 2006 and onward. The reported categories with respect to ventilation systems are the same as shown in Figure 2. However, the types of ventilation systems that represented less than 5% of the database of buildings have been omitted from the analysis. This leaves exhaust air, supply and exhaust with heat recovery and exhaust with heat pump. As for heating systems, the categories are electricity (direct, hydronic and air borne), heat pump (exhaust air, air/air and air/water), ground source heat pump and district heating. Out of these, the categories with sufficient number of buildings are ground source heat pump, district heating and exhaust air heat pump. The hypothesis is that the type of ventilation system and type of heating system is affecting the accuracy of the measurement. This is studied by investigating the dispersion in the energy use of the different types of heating and ventilation systems for each building. The dispersion for each ventilation system is quite large. In all cases, the spread is at least ± 10% for half of the buildings (in one case corresponding spread exceeds ±25%). For the heating systems the dispersion is smaller. Maximum spread is ±17% for single family houses (ground source heat pump) and ±14% for multi-family buildings, see Figure 4. This can be interpreted as the energy use being more influenced by the heating system than by the ventilation system.

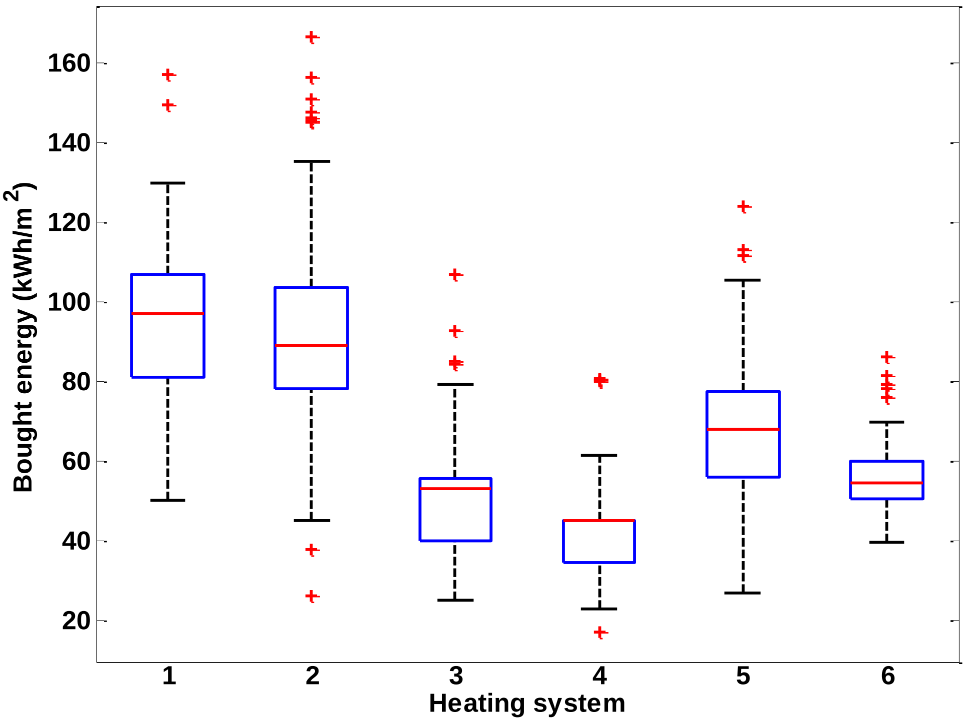


Figure 4: Measured bought energy use with respect to heating system in single family and multi-family buildings (from 2006 until now). Odd numbers are single family houses and even numbers are multi-family buildings. System 1-2 is district heating, 3-4 is ground heat pump and 5-6 is exhaust air heat pump.

Another important aspect that influence the energy use is human behavior. The occupants' behavior give rise to large variations in energy use. Fremling (2013) describes that 50 % of the energy use can be attributed to the occupant behavior. Hot water consumption, airing and indoor air temperature are three aspects that have major impact on the energy use. In single family houses, the measured EPCs is very much affected by the behavior of the group/family living in the house. In multi-family buildings, the dispersion will be smaller since some apartments have lower energy use (e.g. use less hot water) and some have higher. This will decrease the variation. As seen in Figure 4, the dispersion in energy use is slightly smaller in multi-family buildings (bar 4 and 6) and larger in single family houses (bar 3 and 5). An exception is district heating (bar 1 and 2). The reason is probably that there is an error in the measured energy use in multi-family buildings due to culvert heat losses, which is explained in section 6. This larger dispersion in multi-family buildings evens up the single family house dispersion due to behavior.

5 SIMULATIONS OF ENERGY USE AND OCCUPANT BEHAVIOR

Occupants affect the energy use by varying behavior and preferences when it comes to indoor temperature, hot water consumption, electricity use, airing habits, etc. The first three factors have been investigated using the numerical program IDA Indoor Climate and Energy 4.5.1 (IDA ICE) to simulate a single family house in Lerum. In total 54 cases have been simulated. The average indoor temperature is 21°C in single family houses and 22°C in multi-family buildings while 20°C is the temperature used when calculating energy use in passive houses (Sveby, 2012). Therefore these three levels (20°C, 21°C and 22°C) of indoor air temperature were used in the simulations. The number of people in the dwelling varies depending on the size on the house. The most common number of people in single family houses is 2 adults, followed by 1 adult, and 2 adults with 2 children (Statistics Sweden, 2010). The household electricity in dwellings has increased during the last years by the use of more household appliances which has not been counteracted by the more energy efficient equipment. During 2005-2008 measurements in 200 single family houses showed that the household electricity was 5 100 kWh per year on average. The recommended value for household electricity in single family houses is 2 500 kWh per household and year plus 800 kWh per person and year (Sveby, 2012). The hot water consumption varies depending on if the dwelling is located in a single family house or in a multi-family building. Measurements have shown that the hot water consumption is generally lower in single family houses than in multi-family buildings. In 1994 an area with mostly single family houses had a hot water consumption of 53 m^3 per person. In a study from 2007, the hot water consumption was found to be 12 m^3 per person in single family houses. Sveby (2012) recommends that a hot water consumption of 14 m^3 per person is used in energy use calculations for single family houses. Table 1 presents the input data used in the simulation study.

Table 1: Input data on behavior analysis in IDA ICE. The values are based on the recommendations given by Sveby (2012) and variations found in the literature (Sveby 2012).

Number of persons	Household electricity	Hot water consumption	
1	3.3W/m^2 = 5 100 kWh/year	145 l/pers/d = 53 m^3/year/pers	High (blue/left)
2	1.6 W/m^2 + 0.52 W/m^2/pers = 2 500 kWh/year + 800 kWh/year/pers	38.4 l/pers/d = 14 m^3/year/pers	Medium (red/middle)
4		32.9 l/pers/d = 12 m^3/year/pers	Low (green/right)

When all 54 combinations of these cases are simulated, on the same house, there are large differences in the results, and the energy use vary from 73 kWh/m^2 to 39 kWh/m^2 due to behavior. This corresponds to the results in Fremling (2012). The variation in hot water causes the largest difference, which is shown in Figure 5.

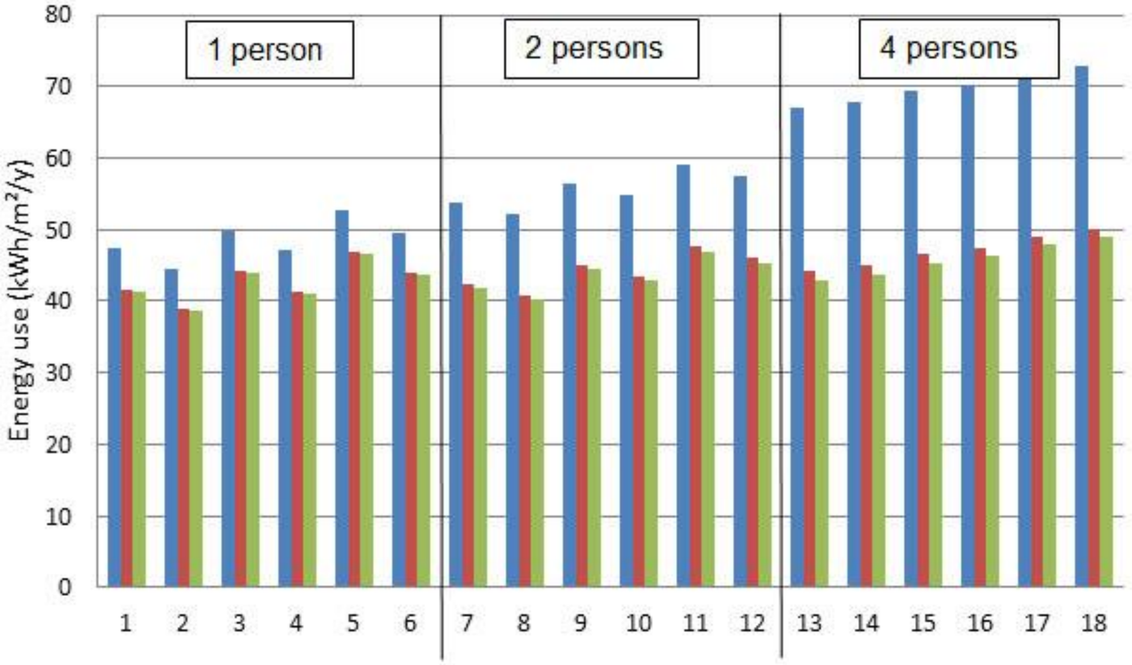


Figure 5: Energy use for space heating and hot water in a simulated single family house with 1, 2 and 4 people in the building. The different cases have different indoor air temperature (20°C, 21° and 22°C) and different household equipment use, as presented in Table 1. (Blue/left bar- high usage, red/middle bar-medium usage, green/right bar- low usage.)

The highest consumption compared to the lowest consumption give differences as large as 36% when it comes to hot water (four persons in the house). Corresponding number for indoor air temperature is 11.5% (20°C compared to 22°C) and for household equipment it is 6.5% (four persons compared to 1 person in the house). In the Swedish EPC, the measured energy use should be corrected to represent a normal year in terms of climate. The measurements should also be corrected with respect to ‘normal’ usage. However, this is very rarely done. A ‘normal’ usage is not defined in the EPC and the behavior in the house (for example indoor air temperature) is not followed up.

6 MEASURED ENERGY USE IN DISTRICT HEATING SYSTEMS

In addition to occupant behavior, the measured EPC is also affected by the location of the energy meter. Losses in the distribution of heat in a property with several buildings can be substantial if the culvert is placed outside of the building envelope, after the energy meter. In energy use calculations this heat loss is often omitted which could explain parts of the difference between measured and calculated energy use for buildings heated with district heating. Figure 6 presents a building with 117 rental apartments divided on 10 staircases.

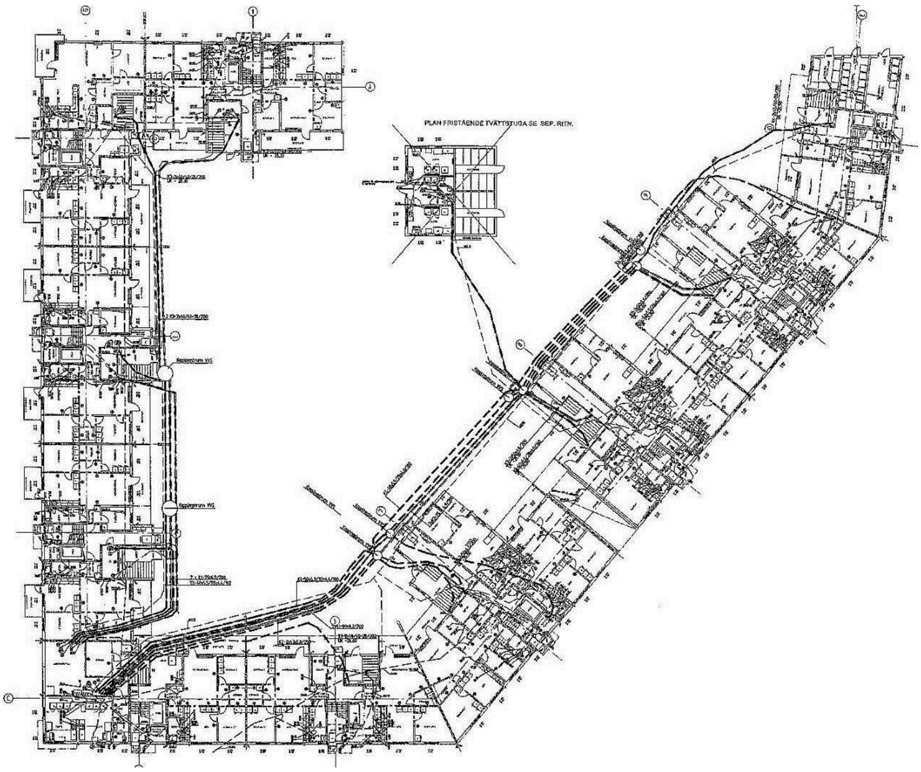


Figure 6: Buildings with an 800 meter long heating culvert placed outside of the building envelope (Bergqvist, 2011).

The building was constructed in 2008 and is located in Huddinge, south of Stockholm. It is heated by district heating and a supply and exhaust air handling unit with heat recovery is installed. In the EPC the reported energy use is 118 kWh/m^2. A detailed investigation (Bergqvist, 2011) of the building showed that this energy use is higher than expected by normal use. Three reasons were identified. A lower efficiency (50%) of the heat exchanger in the air handling unit and 50% higher hot water consumption than expected. Furthermore, there were also large heat losses from a heating culvert outside of the building envelope with a total length of approximately 800 meter. The measured energy use of the building is 82 kWh/m^2 when these abnormalities were accounted for. Of the additional heat losses the culvert accounted for 10.3 kWh/m^2, which increases the energy use with 12.6% if it is not corrected (Bergqvist, 2011).

7 DISCUSSION AND CONCLUSIONS

This study investigated the measured energy use of buildings reported in the EPCs in Sweden. The correlation between choice of ventilation system and heating system with the energy use was studied by using a database with 1 028 multi-family buildings and 725 single family houses constructed after 2006 in Västra Götaland (south west region). The dispersion in energy use is more influenced by the choice of heating system than by the choice of ventilation system. The occupant's behavior has large influence on the energy use which could partly explain the higher dispersion for single family houses than for multi-family buildings. It is difficult to evaluate the energy performance of a building with measurement since human behavior is included. There is supposed to be an adjustment in the EPC for usage that is not normal. However, the term normal is not defined and it is also difficult to follow up. The simulation study revealed that the energy use in a single family house can vary with more than 30%. The most important parameter is the variation in hot water use. In the database of EPCs only 1.5% of the single family houses and 5.4% of the multi-family buildings had this information. For further evaluation more information is needed on the measured hot water use in each of the buildings to be able to rate the buildings' performance independent of the occupant's behavior.

8 ACKNOWLEDGEMENTS

The work in this paper was supported by the Intelligent Energy Europe project 'Towards improved compliance and quality of the works for better performing buildings (QUALICHECK)'. The contribution to this work from Paul-Clément Rivière, Magnus Österbring and Christer Rosfjord at Boverket for the time spent assembling the Gothenburg EPC data set is also greatly acknowledged.

9 REFERENCES

Bergqvist, B. (2011). *KV KANSLIET, HUDDINGE - Orsaker till hög energianvändning (Causes for high energy use). [In Swedish]*. Energianalys, Lidingö, Sweden.

Boverket. (2010). *Energi i bebyggelsen - tekniska egenskaper och beräkningar - resultat från projektet BETSI (Energy in the built environment - technical properties and calculations - results from the BETSI study). [In Swedish]*. Boverket, Karlskrona, Sweden.

Boverket. (2015). *Regelsamling för byggande, BFS 2015:3 BBR 22 (Regulations for construction). [In Swedish]*. Boverket, Karlskrona, Sweden.

European Parliament. (2010). *Directive 2002/91/EC of the European Parliament and of the Council of 16 December 2002 on the energy performance of buildings* Retrieved from http://eur-lex.europa.eu/legal-content/EN/TXT/?uri=CELEX:32002L0091.

Fremling, A. (2013). *Users impact on energy consumption in a low energy building*. Departement of Civil and Environemental Engineering, Chalmers University of Technology, Göteborg, Sweden.

IEA. (2013). *Policy Pathway: Modernising Building Energy Codes*. OECD/IEA, Paris, France and United Nations Development Programme (UNDP), New York, NY, USA.

Statistics Sweden. (2010). *Hushållens ekonomi (HEK) 2008 – HE0103 (Households' economy). [In Swedish]*. Statistics Sweden, Enheten för ekonomisk välfärdsstatistik, Örebro, Sweden.

Statistics Sweden. (2012). *Bostads- och byggnadsstatistisk årsbok 2012 (Yearbook of Housing and Building Statistics 2012). [In Swedish]*. Statistics Sweden, Construction, Rents and Real Estate Statistics Unit, Stockholm, Sweden.

Sveby. (2012). *Brukarindata bostäder - Version 1.0, 2012-10-10 (Input data for occupant's behavior in dwellings). [In Swedish]*. Sveby Standardisera och verifiera energiprestanda för byggnader, Stockholm, Sweden.

Swedish Energy Agency. (2014). *Energistatistik för småhus 2013 Energy statistics for one- and two-dwelling buildings in 2013. [In Swedish]*. Swedish Energy Agency, Eskilstuna, Sweden.

DEVELOPMENT OF A COMPACT SINGLE ROOM VENTILATION UNIT WITH HEAT RECOVERY DEDICATED TO TERTIARY BUILDING

Samuel Gendebien*[1], Jonathan Martens[2], Luc Prieels[2] and Vincent Lemort[1]

1 University of Liege
Thermodynamics Laboratory
Chemin des Chevreuils, 7
4000 Liège
Belgium
**Corresponding author: sgendebien@ulg.ac.be*

2 Greencom Development SCRL
Rue Gilles Magnée, 92/3
4430 Ans
Belgium

ABSTRACT

In the frame of the European project called Bricker, a new prototype of single room ventilation with heat recovery has been developed. This new unit is supposed to be installed in class rooms of an educational institution. This paper deals with the development of the first prototype of this unit. An empirical model of such device is also proposed in order to be coupled with a building model. This aims at determining the seasonal performance of the device and thus the potential energy saving (compared to other technologies) resulting from its use.

The first part of the paper presents the specifications and the final characteristics of the developed device. In this context, a by-pass for free cooling in summer conditions as well as the strategies under frosting conditions are described.

Secondly, the coefficient of performance (COP) of such device is recalled. In the early stage of the development process, the COP is determined based on the manufacturer data of the heat recovery exchanger and the fans. The coupling between fan curve and the predicted hydraulic performance of the unit allows for determining a first approximation of the fans electrical consumption for several delivered flow rates.

The third part of the paper focuses on the experimental investigations carried out in order to determine the flow rate really delivered by the unit. Electrical consumptions of several flow rates are also measured in order to characterize the COP of the unit in those conditions.

Finally, a comparison between the measured and the predicted performance based on manufacturer data has been realized in terms of COP. A performance map based on experimental results is proposed in order to be coupled with a building model.

KEYWORDS

Innovative ventilation, heat recovery, laboratory measurement, air-to-air heat exchanger, fan energy use

1 INTRODUCTION

The Bricker project (2013) aims at developing a retrofitting solution package for existing public-owned non-residential buildings in order to achieve a drastic reduction of the energy consumption (beyond 50%) and GHG emissions in this sector. The retrofitting package is based on envelope retrofitting solutions, zero emissions energy production technologies and the integration and operation strategies. The so-called solutions will be implemented in three real demonstration multi-buildings complexes, located in three different climate conditions and with different end-users. One of the investigated solutions focuses on the development of decentralized ventilation with heat recovery also called single room ventilation with heat recovery (SRVHR). This kind of unit has recently been investigated for residential application by Gendebien (2014) but in the frame of the Bricker project, the units are supposed to be installed in tertiary buildings and more particularly in classrooms of an educational institution, in Liège (Belgium). It involves a higher delivered flow rate by the unit compared to a decentralized system dedicated to a residential application.

The development of such units faces many challenges. The major ones are listed herebelow:

- The device has to be constructed in such a way that it can be installed in the windows frame, or in the false ceiling of a room. The device can also be wall mounted in a horizontal or a vertical position. For each case, the condensate evacuation has to be well designed;
- As for every heat recovery ventilation system, the developed device faces a trade-off between a high thermal effectiveness and a related rise of pressure drops inducing a degradation of the global performance of the unit due to a higher energy use for the fan. Greater attention has to be paid to hydraulic performance than in centralized systems since they are directly related to the noise generated by the fans;
- Strategies under frosting conditions have also to be carefully investigated;
- The device has to be equipped with an electronic control to manage the overheating, during summertime, by opening a by-pass valve. The electronic control has also to allow for an auto-regulation of both air flow rates by using a CO_2 sensor or a time clock;
- The device has to be properly designed in order to allow for a high easiness of placement and maintenance by also taking care of the compactness.

2 FEATURES OF THE DEVELOPED DEVICE

2.1 Main components description

The developed single room ventilation unit with heat recovery mostly consists of a parallelepiped box containing:
- two **AC fans** (one dedicated to the indoor air and one dedicated to the outdoor air);
- **filters** (for both indoor and outdoor air flow rates)
- a **by-pass valve** for ventilative cooling during summer time;
- a **mixing chamber** dedicated to frosting strategies;
- an **electronic fan control** for manual or automatic regulation;
- a **heat recovery exchanger**, often considered as the key component of the unit.

The overall dimensions of the developed box and its integration in the building façade are given in Figure 1:

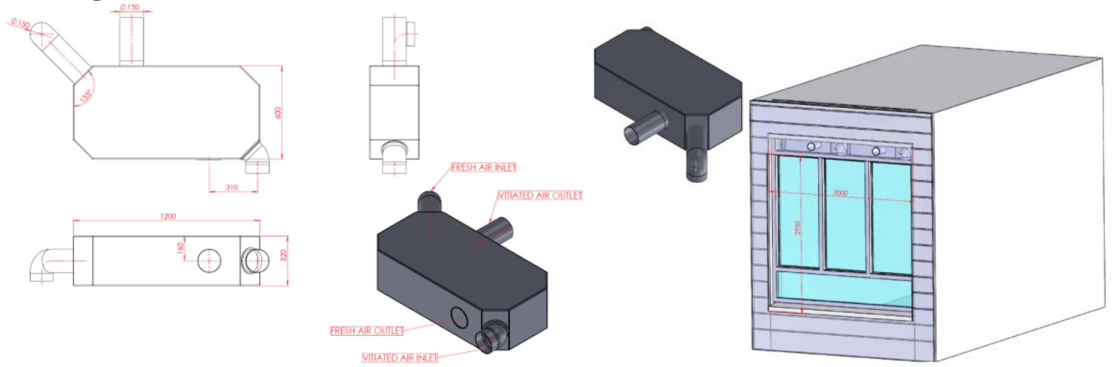


Figure 1: Overall dimensions of the investigated unit

2.2 By-pass

During summertime, when the indoor temperature is higher than a comfortable temperature (25°C for instance) and the outdoor temperature is lower than the indoor temperature, the passage of the outdoor air through the heat exchanger should be avoided in order to take benefit of this "free cooling".

The principle of the by-pass is explained in Figure 2. The fresh air is going through a secondary channel, in parallel of the heat exchanger, while the indoor air is still going through it. When the by-pass valve is open, no heat transfer occurs between the two flows while the room is still ventilated.

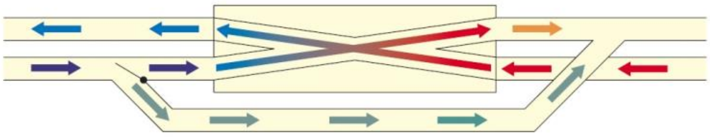


Figure 2: Principle of the by-pass

The by-pass mode is based on two temperature sensors, one measuring the outdoor air and one measuring the indoor air. 3 conditions must be fulfilled for the by-pass to be automatically activated:

- The indoor air is higher than 25°C,
- The outdoor air is lower than the indoor one (with a difference of minimum 1°C),
- The outdoor air is higher than 10°C.

When one of these conditions is not fulfilled anymore, the by-pass is automatically deactivated.

2.3 Strategies under frosting conditions

Every ventilation system equipped by a heat exchanger needs to adopt a defrost strategy. Indeed, when the outdoor temperature is lower than 0°C, a risk of frost appearance may occur in the heat exchanger on the indoor air side. The increasing frost layer leads to a diminution of the indoor airflow rate. This promotes the frost formation resulting in the complete freezing of the whole heat exchanger.
One method under frosting conditions consists in a preheating of the fresh air going to the heat exchanger by means of a mixing with indoor air. The principle is to derive a part of the indoor air flow rate to the fresh air side heat exchanger supply. It can be realized by adding a

duct/section/chamber in direct contact with outdoor air. When the threshold temperature is reached, a valve can be proportionally opened and the mixture can take place. In other operating conditions, the valve remains closed. The situation under frosting conditions is schematically represented in Figure 3:

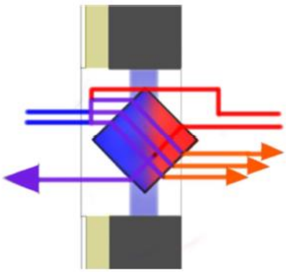


Figure 3: Schematic representation of the "preheating by mixing" strategy

The newly developed ventilation unit is equipped with a mixing chamber. A trap is closed when the outdoor air is above 0°C and can take three different positions when the outside temperature is negative. The three positions are respectively activated at 0°C, -5°C and -10°C. The openings of the trap are studied in order to ensure a positive temperature of the mixing air entering the heat exchanger. By this strategy, no frost can appears inside the heat exchanger but a drawback of this process is the partial recirculation of the different contaminants of the room.

3 OVERALL PERFORMANCE OF SINGLE ROOM VENTILATION WITH HEAT RECOVERY UNIT

Overall performance of centralized heat recovery ventilation is highly dependent on the hydraulic circuit (singularities, such as bending of the pulsing and extracting ducts) and therefore on the building and ducts configuration. In contrary, the overall performance of single room heat recovery ventilation is not influenced by the rest of the installation. As a result, performance of single room ventilation with heat recovery does not depend on the building characteristics but only on the characteristics of the device itself.

As proposed by Gendebien et al. (2013a), the overall performance of each unit can be defined as the ratio of the recovered heat transfer rate to the electrical power of the fans and is given by Equation 1:

$$COP_{SRVHR} = \frac{Recovered\ heat\ power}{Electrical\ supplied\ power} = \frac{\dot{Q}_{recovered}}{\dot{W}_{fans}} \tag{1}$$

By only taking into account the sensible part of the heat transfer rate (the total amount of latent heat rate compared to sensible recovered heat can be neglected in moderate climate as Belgium, according to Gendebien et al. (2013b)), the recovered heat transfer rate is given by Equation 2 and depends on the heat exchanger effectiveness (varying with the mass flow rate), the delivered mass flow rate and on the indoor/outdoor difference temperature:

$$\dot{Q}_{recovered} = \dot{M}_{fresh} \, . cp \, . \varepsilon \, . (T_{ind} - T_{out}) \tag{2}$$

With:

- $\dot{M}_{fresh}$ the fresh air mass flow rate in [kg/s];
- cp the air specific heat in [J/kg-K];
- ε the heat exchanger effectiveness [-];
- T_{ind} the indoor temperature;

- T_{out} the outdoor temperature.

As represented in Figure 1, the parameters influencing the COP of the unit are:

- **Fan** performance;
- **Hydraulic performance of the unit**. This can be divided in two parts: one related to the passage of the air flow in the **heat exchanger itself** and another one related to the flows through **the rest of the unit** (filter, supply and exhaust of the unit);
- **Effectiveness** of the heat exchanger;
- **Climate**: indoor/outdoor temperature difference. From a yearly performance point of view, the interest of use of heat recovery ventilation is highly dependent on the climate (recovered heat over one year vs electrical consumption due to fans).
-

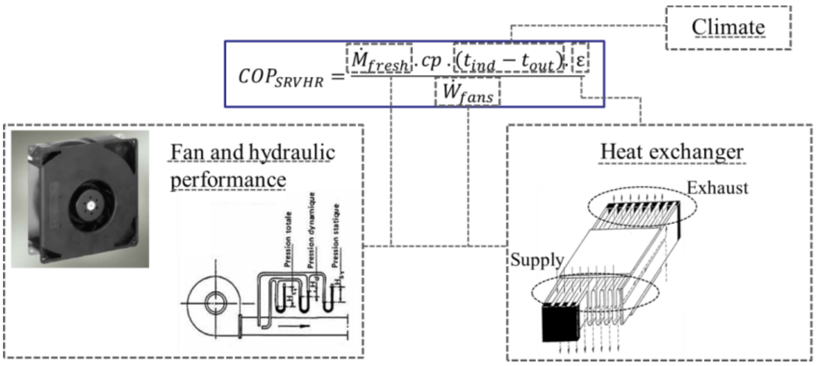


Figure 4: Parameters influencing the COP_{SRVHR}

4 DESIGN STEPS OF THE UNIT

In the frame of the Bricker project, it was decided to manufacture the overall box in order to integrate the several components in the most effective way. One important part of the design was to choose the best set of components (especially the heat exchanger and the fan). Of course, the control of the device was also a major part of the design process. The control involves a flow rate regulation based on the room occupancy, the opening of valves for free cooling during summer time and for freezing strategies during winter time.

Concerning the heat exchanger, a benchmarking between off-the-shelf products has been performed. For a same compactness and standard dimensions, the criteria of selection were the thermal performance, the hydraulic performance, the long term as well as short term mechanical robustness and of course the unitary price. The chosen heat exchanger presents a quasi counterflow configuration and overall dimensions of 360*300*360 mm.

Concerning the fans, the choice had to be made between AC and DC power supply. Both of them presented the same volute diameter. It was decided to choose the AC one for two main reasons. First, it avoids the use of a power consuming AC/DC current converter, and thus a higher price for the fan and its control. Secondly, it presents a better acoustic performance.

Once the appropriate options for fan and heat exchanger were selected, it was possible to compute a performance map based on performance presented in manufacturer catalogues. Those predicted overall performance will be compared with measured performance in Section 6.

Delivered flow rate for a specific rotational speed of fan can be deduced from the intersection of the fan performance curve with the hydraulic performance curve of the unit. Hydraulic curve of the unit is the sum of several contributions. The total pressure drop is due to the passage of air through filter, through the heat exchanger, and through the rest of the unit (the inlet and the exhaust of the unit). During this step, one assumed the use of G4 filters for both sides of the unit (indoor and fresh air side). It has been arbitrarily assumed that the pressure drop related to passage of air through the inlet and exhaust of the unit accounts for 40% of the pressure drop related to the air passage in the filter and the HX. The hydraulic reconciliation is given here below in Figure 4.

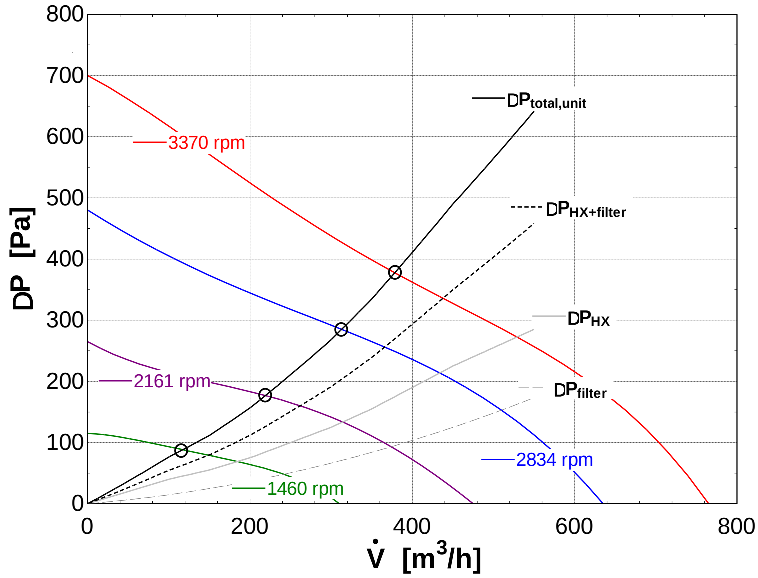


Figure 5: Intersection between the total pressure drop of the unit and fan curves

For determining a performance map, two more pieces of information are needed: the heat exchanger effectiveness and the fan electrical consumption depending on the rotational speed and on the flow rate. Those pieces of information, based on manufacturer catalogue, are given in Figure 5.

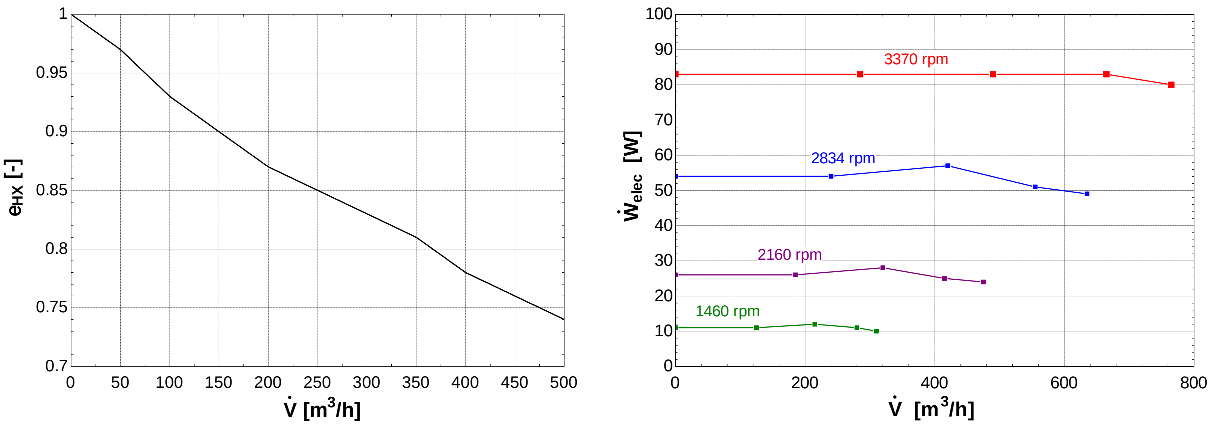


Figure 6: Heat exchanger effectiveness (left) and electrical consumption for fan depending on the volumetric flow rate and the rpm (right)

It is now possible to determine the evolution of the overall performance of the unit as a function of the volumetric flow rate for a specific indoor/outdoor temperature difference, on the basis of Equation 1 and 2. The expected evolution of the COP of the unit for an indoor/outdoor temperature difference of 11.5K is given in Figure 6. For the COP estimation, a difference temperature of 11.5K has been chosen. This corresponds to an indoor temperature of 20°C and an outdoor temperature of 8.5°C (mean indoor and outdoor Belgian temperature).

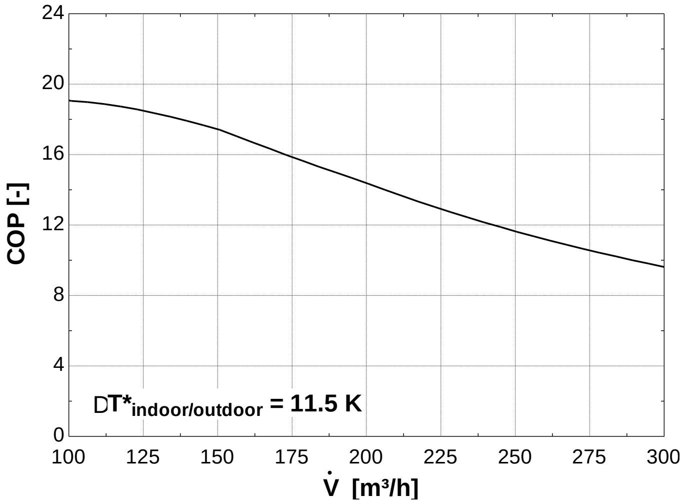


Figure 7: Expected coefficient of performance of the unit (design step)

5 EXPERIMENTAL INVESTIGATIONS

This section aims at presenting the experimental investigations carried out in order to determine the flow rates delivered by both fans, as well as their electrical consumption.

5.1 Delivered flow rate

A nozzle has been used in order to determine the flow rate delivered by both fans for various rotational speeds. The knowledge of the pressure drop between the inlet/outlet of the nozzle allows for determining the flow rate. The passage through the nozzle induces an additional pressure loss leading to a decrease of the delivered flow rate. In order to eliminate this decrease, a balancing (also called compensating) fan is used. A differential pressure drop sensor is placed between the atmosphere and the exhaust of the unit. The compensating fan rotational speed can be modified. Once the measured differential pressure between the atmosphere and the exhaust of the unit is equal to zero, the device is supposed to operate in nominal conditions and the volumetric flow rate measured by means of the nozzle is the one really delivered by the unit in normal conditions. Electrical consumption has been realized for various rotational speeds of both fans by using a calibrated AC current analyzer.

A practical achievement of the experimental set-up is given in Figure 4:

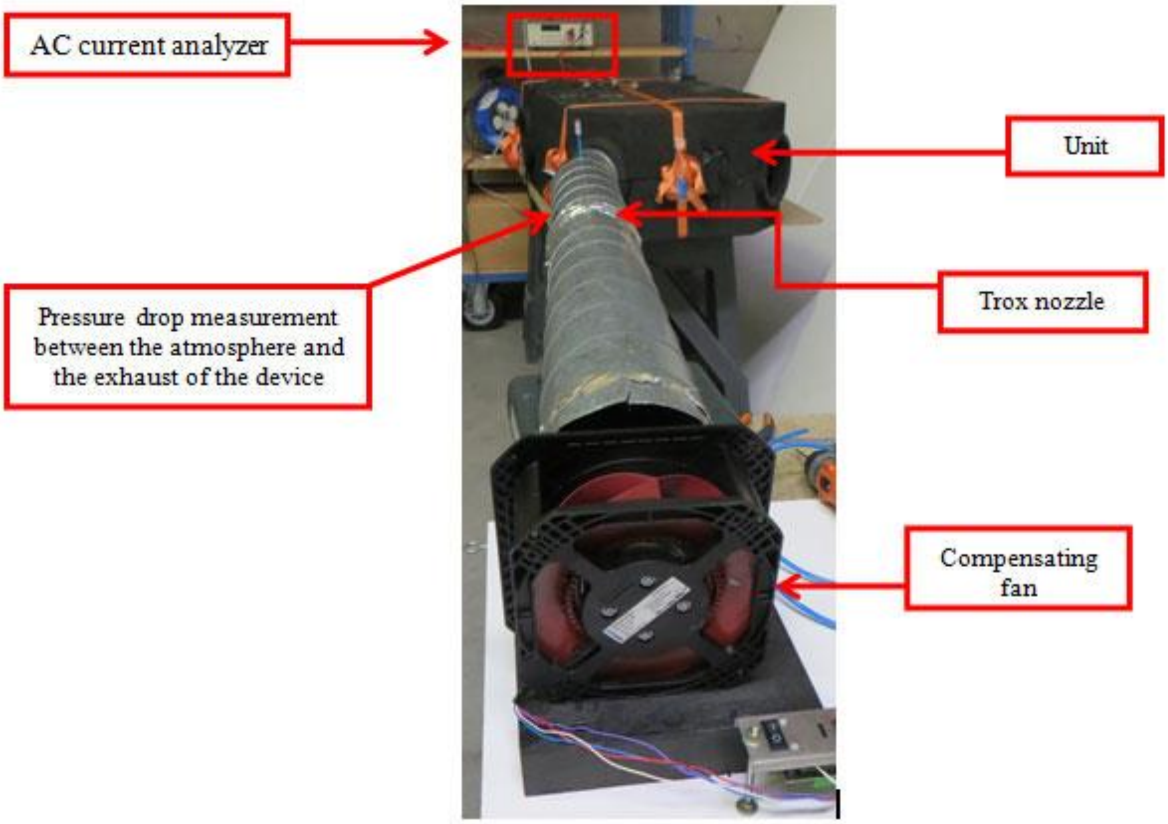


Figure 8: Practical achievement

5.2 Heat exchanger effectiveness

The heat exchanger effectiveness has been experimentally determined for three different balanced air flow rates. Thermocouples have been installed at the inlet and the exhaust of both sides of the unit. Tests have been carried out in the laboratory of the heat exchanger manufacturer on the finalized unit. Effectiveness of the recovery heat exchanger has been determined by means of stabilized tests of 10 min under dry conditions (without condensation appearance). The supply temperature of the indoor air was around 25°C and the fresh air temperature was around 5°C. Effectiveness has been determined for three different balanced mass flow rates (see Figure 8).

6 COMPARISON BETWEEN DESIGN AND EXPERIMENTAL RESULTS

The aim of this section is to compare the expected results obtained during the design steps with the measured ones obtained during the experimental campaign.

6.1 Hydraulic performance comparison

Table 1 gives a numerical comparison between the expected and the measured hydraulic performance of the unit (for both indoor and fresh air side). It can be observed that generally predictions are in good agreement with measured results. However, it can be noticed that for the lowest flow rate, measurements shows better performance compared to the predicted ones.

Table 1: Hydraulic performance comparison

Expected hydraulic performance			Measured hydraulic performance (fresh air side)			Measured hydraulic performance (indoor air side)		
$\dot{W}_{fan}$ [W]	$\dot{V}$ [m³/h]	SFP [W/m³-h]	$\dot{W}_{fan}$ [W]	$\dot{V}$ [m³/h]	SFP [W/m³-h]	$\dot{W}_{fan}$ [W]	$\dot{V}$ [m³/h]	SFP [W/m³-h]
76.29	366	0.208	86	366	0.235	84	366	0.229
55.18	315	0.175	55	315	0.175	55	314	0.175
30.5	232	0.131	27	232	0.115	27	226	0.119
15.3	153	0.1	10	153	0.065	10.5	142	0.074

6.2 Thermal performance comparison

Figure 8 shows a comparison between the expected and the measured heat exchanger effectiveness. It can be observed that the measured HX effectiveness is approximately 10 percentage points lower than the expected HX effectiveness. One reason that could explain this difference is a potential misdistribution of the air flow rate in the heat exchanger because of the proximity of the fan exhaust with the heat exchanger entrance.

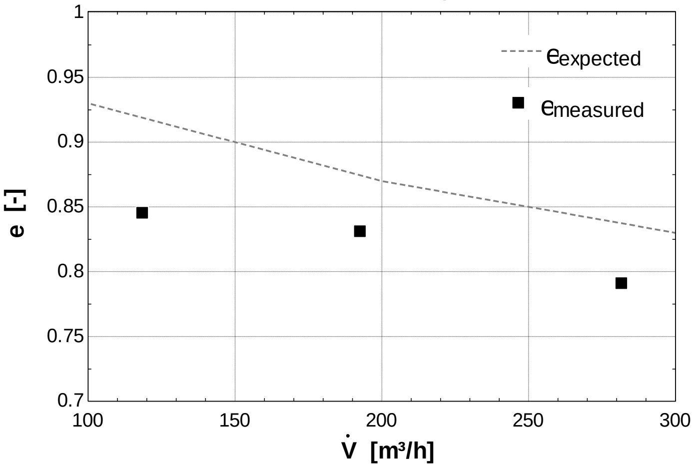


Figure 9: Comparison between expected and measured heat exchanger effectiveness

6.3 COP comparison

A comparison between the COP (as defined in section 2) obtained during the design step and the experimental procedure is given in Figure 9. Interpolation between experimental data has been realized for establishing the measured COP evolution. Once again, the indoor/outdoor temperature difference has been chosen equal to 11.5K and the flows are considered perfectly balanced.

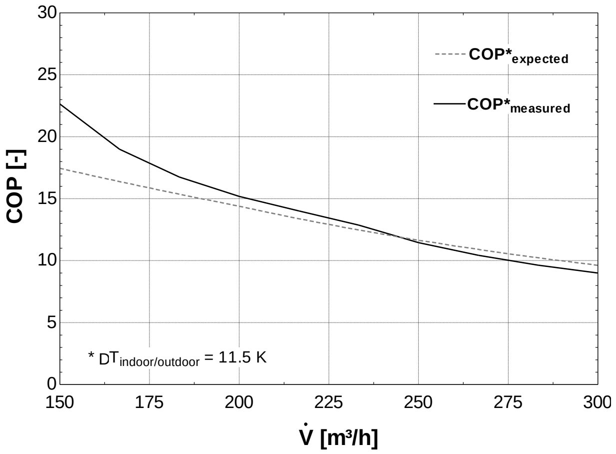


Figure 10: Comparison between expected and measured performance

As shown in Figure 8, the COP obtained during the design step shows good agreement with the measured COP, even if some differences between predicted and measured characteristics exist. The decrease of the actual heat exchanger effectiveness in comparison with expected results is counterbalanced by a better measured hydraulic/fan performance.

7 CONCLUSIONS AND FUTURE WORK

The present paper focuses on the development and the experimental characterization of a decentralized ventilation system with heat recovery to be installed in classrooms of an educational institution. The proposed system integrates solutions for avoiding freezing in the heat exchanger, for free cooling and to adapt air flow rates to occupancy. A design procedure has been performed in order to assess an expected overall performance of the unit based on HX, fan and filter manufacturer data. Then, an experimental procedure has been carried out in order to compare expected results with measured results. Expected results shows good agreement with the overall performance measured on the real unit. Future work will consist in installing the developed unit in classrooms of an educational institution in Liège (Belgium) and testing in situ the control of the unit.

ACKNOWLEDGEMENTS

This study has been carried out in the frame of the Bricker project. This project is performed with the support of the European Commission and is carried out by a large consortium of research centers and industrial partners, such as Greencom development s.c.r.l.

REFERENCES

Bricker Project. Launched in October 2013. Energy reduction in public building stock. http://www.bricker-project.com/

Gendebien, S., Georges, E., Prieels, L., Lemort, V., 2013 (a). Experimental performance characterization of a new single ventilation device with heat recovery. AIVC conference. Athens. Septembre 2013.

Gendebien, S., Bertagnolio, S., Lemort, V., 2013 (b). Investigation on a ventilation heat recovery exchanger: modeling and experimental validation in dry and partially wet conditions. Energy and Buildings, 62 (2013), pp. 176–189

Gendebien, S. 2014. Contributions to the development of a single room ventilation unit with heat recovery. PhD Thesis. University of Liège. 2014

IEA EBC ANNEX 68 – INDOOR AIR QUALITY DESIGN AND CONTROL IN LOW ENERGY RESIDENTIAL BUILDINGS

Carsten Rode[*1]

1 Department of Civil Engineering
Technical University of Denmark
Building 402, DTU
Kgs. Lyngby, Denmark
E-mail: car@byg.dtu.dk

1 INTRODUCTION TO IEA EBC ANNEX 68

The overall objective of the IEA EBC Annex 68 is to provide scientific basis usable for optimal and practically applicable design and control strategies for high Indoor Air Quality (IAQ) in residential buildings. Naturally, those strategies should ensure minimal possible energy use. The project aims to gather existing and provide new data on pollution sources in buildings, model the indoor hydrothermal conditions and air quality as well as thermal systems, and will look to ways to optimize the provision of ventilation and air-conditioning.

The work of the Annex is organized into five subtasks:

- Subtask 1 will set up the metrics for required performances which combine the aspiration for very high energy performance with good indoor environmental quality.
- Subtask 2 is to gather existing knowledge or provide new data about indoor air pollutants as well as combined heat, air and moisture transfer.
- Subtask 3 will identify and/or further develop modelling tools that can assist designers and managers of buildings.
- Subtask 4 will build up on the fundaments laid by previous subtasks and develop a guidebook on design and control strategies for energy efficient ventilation in residential buildings that will not compromise indoor air quality.
- Subtask 5 will identify and investigate relevant case studies and do field measurements where the above mentioned strategies can be examined and optimized.

The different subtasks are presented in more detail in extended abstracts for each of them.

2 KEY OBJECTIVES

The Annex has the following specific key objectives:

- To develop design and control strategies for energy efficient buildings that will not compromise the quality of the indoor environment. Operational parameters that will be dealt with will comprise, but not be limited to the means for ventilation and its control, thermal and moisture control and air purification strategies - and their optimal combination.
- To set up the metrics for required performances which combine the aspiration for very high energy performance with good indoor environmental quality.
- To identify or further develop the tools that will be needed to assist designers and managers of buildings in achieving the first key objective.
- To benefit from recent advances in sensor technology and controls, e.g. model based control principles, to identify methods to enhance indoor air quality while ensuring minimal energy consumption for operation.
- To gather existing or provide new data about indoor pollutants and properties pertaining to heat, air and moisture transfer that will be needed for the above analysis.
- To identify and investigate relevant case studies where the above mentioned performances can be examined and optimized.
- To disseminate about each of the above findings.

3 TARGET AUDIENCE

The project addresses the following primary stakeholders:
- Building designers (engineers and architects),
- Suppliers of HVAC and control systems
- Suppliers of materials used for building constructions and indoor furnishing,
- Providers of building management systems.

The project shall also address the interests of building owners, facility managers and users, as well as authorities that stipulate the building regulations and who administrate the rules. The perspective is that the project may indicate ways how future energy classes for buildings can be stipulated as being dependent on which pollution targets they can achieve.

4 ROLE OF VENTILATION IN IEA EBC ANNEX 68

The rationale of carrying out the proposed Annex is that buildings in the future will have to be optimized just to the limit in order to become as close as possible to being zero energy buildings. This means that the ventilation will also be reduced to just the absolutely necessary, while the quality of the indoor air must not be sacrificed. There is a need to adopt and demonstrate an integral view in the optimization that consider the sources, sink and transport of relevant pollutants that occur in buildings against the effect of ventilation.

The project is one of several past, recent and ongoing IEA EBBC Annex projects where ventilation plays a role. The AIVC (being the perpetual IEA EBC Annex 5) is one of them. Others are IEA EBC Annex 59, 60, 61, 62, 66, 67 and 69, and the EBC Executive Committee has facilitated a platform for coordination between them.

5 ANNEX DURATION AND PARTICIPATION

The project has commenced its preparation phase in 2015, which is planned to be followed by a three year working phase (2016-18). Commitments to participate shall be organized during the preparation year. While formal commitments are still in preparation for most participants, some 33 institutions from 16 countries have shown interest in the project (by June 2015).

The purpose of the Annex 68 Workshop at the 2015 AIVC Conference is to invite comments and possible project involvement or interaction with experts from the AIVC community and ventilation industry.

IEA EBC ANNEX 68 – SUBTASK 1: DEFINING THE METRICS

Marc Abadie*[1], Pawel Wargocki[2]

1 Laboratoire des Sciences de l'Ingénieur pour l'Environnement (LaSIE) - University of La Rochelle
Pôle Sciences et Technologie
Avenue Michel Crépeau
17042 La Rochelle Cedex 1, France
marc.abadie@univ-lr.fr

2 Technical University of Denmark (DTU)
Department of Civil Engineering
Nils Koppels Allé
2800 Kgs. Lyngby, Denmark

1 EXTENDED ABSTRACT

Subtask 1 of IEA EBC Annex 68 will aim at defining the metrics to enable a proper consideration of both energy and IAQ benefit in building design and operation.

A first step will consist in determining a list of target pollutants commonly found in residential buildings by identifying pollutants that are listed by cognizant authorities as harmful and verifying whether they are present in residential environments and at the concentrations, which can surpass the recommendations of the authorities. Since the 1980s, guideline values for pollutants typically found in buildings indoor air have been proposed. Those values are based on comprehensive review and evaluation of accumulated scientific evidence by multidisciplinary groups of experts studying the toxic properties and health effects of these pollutants. The world health organization (WHO, 2010), the European INDEX project (2005) and national health agencies such as ATSDR (USA), ANSES (France)… have proposed guidelines values for chemicals commonly present in indoor air. By considering in-situ measurements in specific indoor environments (dwellings, offices, schools…), target pollutant lists have been established by WHO, INDEX and French IAQ Observatory. Benzene, carbon monoxide, formaldehyde, naphthalene, nitrogen dioxide, PM and PAH are clearly identified as high-priority target pollutants. Biological pollutants such as molds have been considered separately by those studies as no dedicated guideline values are available and will have to be accounted for because of their potential effects on health.

The existing IAQ metrics will then be reviewed to propose the best scientifically-sounded index (or set of indices) for the evaluation of indoor air pollution. According to Sofuoglu and Moschandreas (2003), an index of IAQ must be able to communicate to a non-scientific audience on indoor air pollution levels, must be correlated to the symptoms experienced by the occupants and can be used as a management tool to effectively improve air quality. A literature review of existing indoor environmental quality indices has been recently carried

out by the French Observatory of Indoor Air Quality (Kirchner et al., 2006). On the whole, the IAQ indices considered different pollutants, exposure limits and aggregations. As an illustration of the construction and use of such IAQ index, the IAPI (Indoor Air Pollution Index) has been chosen here because it is the only multi-pollutant index of indoor air quality that has been validated to date against health effects. This index, developed by Sofuoglu and Moschandreas (2003), is estimated from the averaged concentration of 8 pollutants: VOCs (formaldehyde and TVOC), inorganic gases (CO and CO_2), particulate matters (PM2.5 and PM10) and biological particles (bacteria and fungi).

A last part of this subtask will be dedicated to the inclusion of energy in the proposed evaluation. In particular, the index will also have to include additional energy consumption needed to improve IAQ in comparison with standard practice such as increased fan consumption induced by higher air change rates or additional particle/gas filters, or use of portable air cleaners. As an example, Figure 1 presents a comparison of 5 solutions to improve IAQ (Tourreilles, 2015): 3 HVAC air-cleaners and 2 higher ventilation fresh air rates. The metric is defined as the ratio of an IAQ index (0 being bad, 1 being good) to an Energy one (0 being the solution inducing the lowest additional energy consumption). In this example, the electronic filter is clearly the best choice for PM, the "F7+Carbon Filter" and doubling the fresh air rates are equivalent to treat both formaldehyde and PM and the two other solutions induce too high energy consumption.

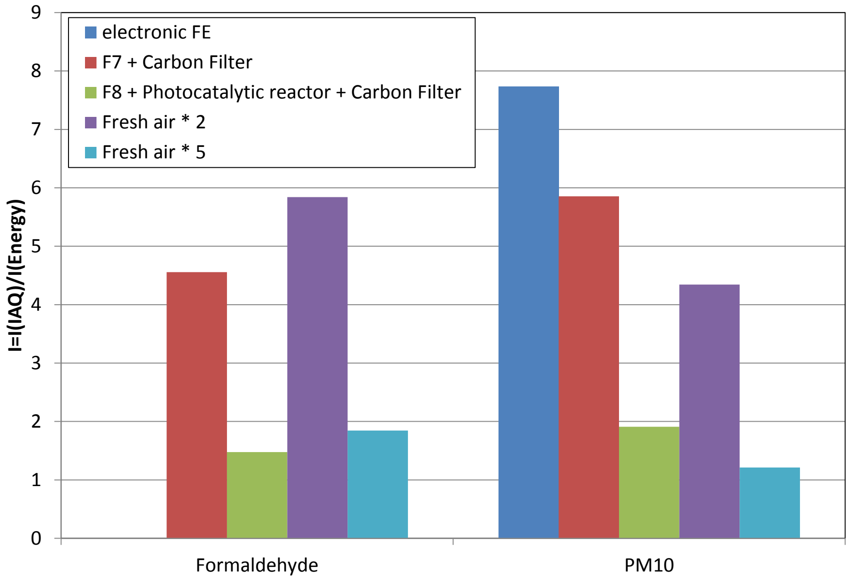


Figure 1: IAQ/Energy impact of 5 solutions proposed to improve IAQ in an office room

2 REFERENCES

INDEX. (2005). *The INDEX project: Critical Appraisal of the Setting and Implementation of Indoor exposure Limits in the EU*. European Commission, Joint Research Centre, Institute for Health and Consumer Protection, Physical and Chemical Exposure Unit, Ispra, Italy (JRC/IHCP/PCE), Report, 338 p.

Kirchner, S., Jédor, B., Mandin, C. (2006). *Elaboration d'indices de la qualité de l'air intérieur : phase 1 – Inventaire des indices disponibles*. Report DDD-SB/2006-065.

Sofuoglu, S.C., Moschandreas, D.J. (2003). The link between symptoms of office building occupants and in-office air pollution: the Indoor Air Pollution Index. *Indoor Air, 13*, 332-343.

Tourreilles, C. (2015). Qualification énergétique et sanitaire des systèmes d'épuration intégrés aux réseaux de ventilation. To be defended in September 2015.

WHO. (2010). *WHO guidelines for indoor air quality: selected pollutants*. World Health Organization, Regional office for Europe. http://www.euro.who.int/__data/assets/pdf_file/0009/128169/e94535.pdf

IEA EBC ANNEX 68 – SUBTASK 2, POLLUTANT LOADS IN BUILDINGS

Menghao Qin*[1], Jianshun Zhang[2], Xudong Yang[3], Carsten Rode[4]

[1] School of Architecture and Urban Planning
Nanjing University
Nanjing, China
Email: mqin@nju.edu.cn

[2] Building Energy & Environmental Systems Laboratory (BEESL)
Syracuse University
Syracuse, NY, USA

[3]Department of Building Sciences
School of Architecture
Tsinghua University
Beijing, China

[4]Indoor Environment and Building Physics Section
Department of Civil Engineering
Technical University of Denmark
Lyngby, Denmark

INTRODUCTION

One obstacle to integrating energy and IAQ strategies for buildings is the lack of reliable method and data for estimating pollutant loads in buildings in the way heating/cooling loads are routinely estimated. Subtask 2 of IEA EBC Annex 68 is to collect existing data and to a limited extent provide new data about properties for transport, retention and emission of chemical substances in new and recycled materials under the influence of indoor heat and moisture conditions. Formaldehyde, benzene and other harmful volatile organic compounds (VOCs) are of main concern. Collection of results from lab tests on material and room level will be part of this study. Specifically, results will be collected and analysed from tests of emission of harmful compounds under various temperature, humidity and air flow conditions, since such data under combined exposures generally do not exist today.

ACTIVITIES

First the Subtask will organize a literature survey and make researcher contacts to gather relevant data and existing knowledge on major pollutant sources and loads in buildings, including models.

Laboratory testing and model setup to provide examples of new types of data which shall be beneficial to improve knowledge on combined effects that must be taken into consideration in order to achieve new paradigms for energy optimal operation of buildings. It is anticipated that the Subtask will gather data about combined effects describing how temperature and moisture conditions influence the emission and sorption of various pollutants in materials.

STAKEHOLDERS INVOLVED

It is anticipated that manufacturers of building materials, furniture, and inventory products shall be involved regarding testing and possible co-development of products that have minimal emission of harmful substances or which may have function to absorb indoor pollutants. In addition, architects, HVAC engineers and developers will also be possibly involved in different stages of "high IAQ, low energy" building projects.

DELIVERABLES

The Subtask will end with some mechanistic emission source/sink models and IAQ simulation tools for estimating the net loads of pollutions over time under realistic environmental conditions. This will be published in scientific journal articles and in a project report.

Furthermore, the Subtask will produce a database of emission and transport properties of materials for use in the models that will be developed and used in the project's Subtask 3.

Finally, the Subtask will produce a database of common pollution loads in new and existing buildings.

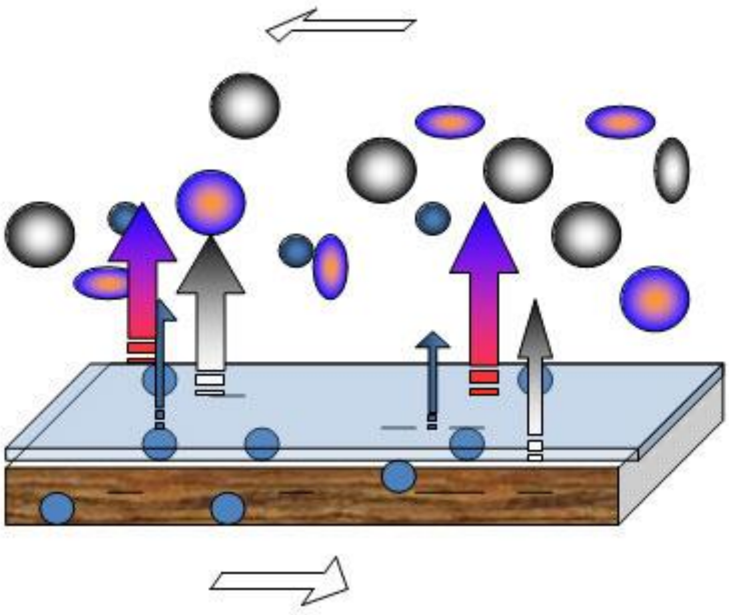


Figure 1 Pollutant emission from materials

REFERENCES

1. Zhang J. and Qin M., Combined heat, air moisture and pollutant simulations (CHAMPS) for buildings, *Building Simulation*, Vol.4-4, 2011, p.279-282.
2. Liang W, Wang C, Yang C, Yang X. Volatile organic compounds in different interior construction stages of an apartment. *Building and Environment*, 2014, 81: 380-387.
3. Liang W, Yang X. Indoor formaldehyde in real buildings: Emission source identification, overall emission rate estimation, concentration increase and decay patterns. *Building and*

Environment, 2013, 69: 114-120.

4. Zhang J. Shao C. and Sender D. et al. MEDB-IAQ: a material emission database and single-zone IAQ simulation program – a tool for building designers, engineers and managers, consortium for material emission and IAQ modelling. Final report 4.2 IRC/NRC, Canada, 1999.
5. Haghighat F. Huang H. Integrated IAQ model for prediction of VOC emissions from building materials. *Building and Environment*, 2003, 38(8): 1007-1017.

IEA EBC ANNEX 68 – SUBTASK 3, MODELLING

Jensen Zhang[1]*, John Grunewald[2]

1 Building Energy & Environmental Systems Laboratory (BEESL)
263 Link Hall, Syracuse University
Syracuse, NY, USA
E-mail: jszhang@syr.edu

2 Institute for Building Climatology
Faculty of Architecture
TU Dresden
Zelleschen Weg 17
1069 den, Germany

1 INTRODUCTION AND OBJECTIVES

The objectives of Subtask 3 Modeling are to improve the understanding and develop prediction models on the impacts of outdoor pollutants, thermal environment, building materials and envelope, and indoor furnishing and occupant activities on the indoor air quality, and the energy necessary to achieve the desired IAQ level in residential buildings, considering the IAQ metrics and pollution loads to be developed in Subtask 1 and 2, respectively.

2 SCOPE AND EXPECTED OUTCOMES

Existing knowledge is inadequate for predicting the combined effects of hygrothermal conditions and chemical reactions on the indoor pollution species and concentrations in light of most recent revelation of the importance of secondary emissions such as Ozone-initiated indoor air and surface chemistry in affecting the indoor air quality. The approach of modeling the effects of combined heat, air, moisture and pollutant (CHAMPS) transport and their impact on energy and IEQ is needed. This task is to collect and develop guidelines about use of contemporary whole building analysis tools and methods to predict the hygrothermal conditions, absorption and transport of humidity and chemical substances, and energy consumption within buildings. The whole building perspective is realized by integral consideration of indoor air and building envelope, building users and the building services systems. Focus, and what can be seen as a new development, will be on methods to predict the emission and absorption of chemical compounds from materials under realistic in-use conditions regarding the CHAMPS-exposure in buildings. Notwithstanding the perspectives in new paradigms for modeling the chemical and atmospheric conditions, it will be a top priority that the methods also facilitate prediction of the energy consumption associated with the operation of buildings, such that the tools can be used to optimize for the minimal energy consumption that satisfies the needs with respect to indoor environmental quality. Activities for the Subtask will include:

- Literature survey and provision of knowledge about contemporary modeling capabilities in thermal whole building energy and hygrothermal analysis in combination with air flow and emission models. Development of a paradigm for work with these models such that they can be used as optimization tool for good building energy performance under high IEQ conditions. Identification of gaps in current modeling capabilities.

- Development of new procedures, when and if necessary, that will be needed to model the interaction between energy efficient operation and high IEQ. Incorporation of the methods for analysis in modeling paradigms from other ongoing IEA activities, e.g. within IEA Annex 60.

Apart from partners from academia, it is anticipated that building design companies and companies involved in building design tool development may contribute to the subtask. It is anticipated that a modeling framework (such as CHAMPS) and design tool (such as a Virtual Design Studio) will be developed for evaluating the energy and IAQ performance of residential building design and operation strategies.

3 COLLABORATIVE WORK PLAN

Conceptually, the modelling framework will include a multizone network model (e.g., CHAMPS-MZ) that integrates an envelope model (e.g., CHAMPS-BES), HVAC model (e.g., E+) and space/room model (e.g., TUD's room model, Tokyo U's response factor model, U. of La Rochelle's Proper Orthogonal Decomposition-POD model, Tsinghua U's residential IAQ model) supported by shared databases of pollutant properties, sources and sink models, chemical reaction models, material properties and weather conditions (Figure 1). The CHAMPS modelling framework will be tailored it to residential applications and be improved through the following planned specific new model developments and implementations:

1. Develop and incorporate models to account for the impacts of indoor air chemistry and surface chemistry/physics on the concentrations of pollutants of interests, including O3 initiated gas phase and surface reactions, particle deposition and resuspension, VOCs and SVOCs adsorption and desorption. Tsinghua U (lead), SU, UT Austin, Health Canada, Missouri University of Science and Technology, and others.

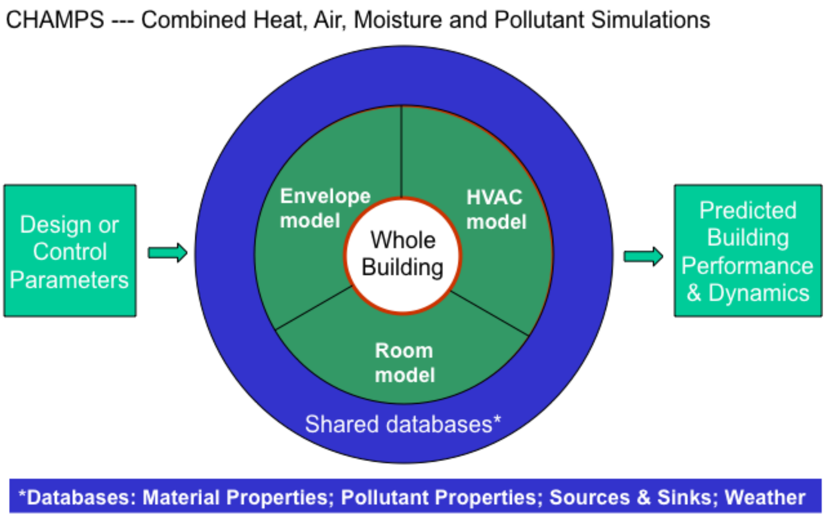


Figure 1 A conceptual CHAMPS model for building systems

2. Develop and incorporate better models for outdoor to indoor pollutant transport, including gas and particle penetration through building envelope, and through outdoor air supply systems. Nanjing University (lead), SU and others.
3. Develop a model for better describing the impacts of temperature and humidity on the transport of VOCs and SVOCs in building materials and envelope systems. Danish Technical University (lead), SU, Tsinghua U and TUe.
4. Develop a fast room simulation model for integration with the multizone network model. TU Dresden and TUe.

5. Develop a response factor-based modelling approach for fast simulation of air and pollutant distribution in rooms for use in the multizone network flow models. University of Tokyo and TUe.
6. Develop a CHAMPS modelling framework that incorporate the above models for residential building applications. SU, TU Dresden and TUe.

IEA EBC ANNEX 68 – SUBTASK 4, STRATEGIES FOR DESIGN AND CONTROL OF BUILDINGS

Jakub Kolarik[1*] and Gilles Rusaounen[2]

1 *Department of Civil Engineering*
Technical University of Denmark
2800 Kgs. Lyngby
Denmark
E-mail: jakol@byg.dtu.dk

2 CETHIL - INSA de LYON
Bat Sadi Carnot
9 rue de la Physique
69621 Villeurbanne Cedex
France

1 SUBTASK 4 - OBJECTIVES

The objectives of Subtask 4 are to develop design and control strategies for energy efficient ventilation in residential buildings which ensure high indoor air quality. The strategies must go beyond the current common practice and actively utilize recent research findings regarding indoor air pollutants and combined heat, air and moisture transfer as well as benefit from recent advances in sensor technology and controls.

2 WORK DESCRIPTION

Subtask 4 will utilize results of previous subtasks (metrics models and databases developed in the Subtasks 1, 2 and 3) together with existing knowledge to devise optimal and practically applicable design and control strategies (see Figure 1). The strategies will take into account requirements for IAQ based on current standards, particular building codes in different countries, as well as newly developed metrics based on health effects. Moreover, the type of ventilation systems (decentralized ventilation, active overflow systems etc.) and air supply mode (e.g. intermittent vs. continuous ventilation) will be considered with respect to different building types. Optimal strategy is understood as one that takes into account building energy performance, user comfort and health conditions.

Use of models and databases developed under the Annex will enable addressing new paradigms for multi-scale and local thermal and air quality management including demand controlled ventilation in residences that considers indoor/outdoor transport of pollutants. The subtask will study and make benefit from recent advances in sensor technology and model based control to identify ways to optimize the indoor air quality without penalizing on the energy efficiency. With this respect, the subtask should seek to establish correlation factors between IAQ and energy consumption. With a base in scientific methods employed in the project, the ambition is that such correlations can be used in future standards and by legislators when specifying regulations for IAQ requirements in highly energy efficient buildings.

Because of the complexity of the assignment described above, the Subtask 4 will invite different stakeholders to actively participate in the project, share their practical experiences and provide feedback to the subtask work. Target stakeholders are mainly building designers, companies that provide ventilation systems and controls as well as housing associations, producers of prefabricated houses and facility management companies.

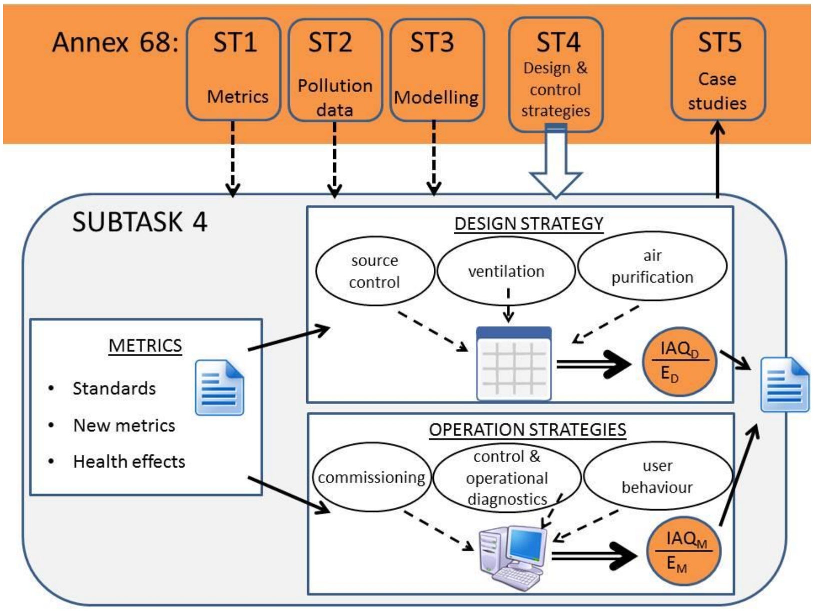


Figure 1 - Schematic overview over composition of Subtask 4; IAQ_D/E_D and IAQ_M/E_M refer to designed and measured correlation factors between IAQ and energy use respectively

3 SUMMARY OF PLANNED ACTIVITIES

4.1 STATE OF THE ART: Review of relevant international activities related to IAQ design and control in residences.

4.2 DESIGN STRATEGY: Determination of an innovative design strategy - using tools and methods from Subtasks 1, 2 and 3 to evaluate design alternatives for several building types. The design strategy should also account for different typologies of ventilation systems.

4.3 CONTROL/OPERATION: Investigation of possible operation strategies - systems designed by means of methodology developed in 4.2 must be operated in a reasonable way to ensure that the designed operational parameters are met. Outcome from Subtasks 1, 2 and 3 can be also applied here, but it needs to be supported by suitable systems for operational diagnostics (for example air quality sensors). This task includes also assessment of suitability of different control strategies and operation modes.

4.4 GUIDEBOOK: Preparation Annex 68 guidebook: This activity will comprise completion of a guidebook on design and operation of ventilation in residential buildings to reach impeccable indoor air quality occupancy with minimum possible energy consumption. The guidebook will summarize results of previous subtask activities. The guidebook will be divided into sections related to design, operation and communication to building managers and occupants. The guidebook will be main outcome of the Subtask 4.

IEA EBC ANNEX 68 – SUBTASK 5, FIELD MEASUREMENTS AND CASE STUDIES

Kristine Nore[1]*, Sarka Langer[2] and Wei Pan[3]

1 Norwegian Institute of Wood Technology
Oslo, Norway
E-mail: kristine.nore@treteknisk.no

***2** IVL Swedish Environmental Research Institute*
Göteborg, Sweden

***3** The University of Hong Kong*
Department of Civil Engineering
Pokfulam, Hong Kong

1 INTRODUCTION

With a tighter building envelope more minimal influences come into consideration. As how the thermal and hygrothermal properties of the materials may improve the ventilation or whether the dust distribution is altered with the new surface temperatures and flows and of course how all other new chemicals which we introduce to living environments affects the IAQ. Preferably the in situ measurement data from the filed campaigns and case studies will be defined as ST5 in Annex 68.

2 SUBTASK 5 – OBJECTIVES

- To carry out field tests and analysis of buildings for testing and verification of the results from the other subtasks.
- To investigate new ventilation patterns in highly energy efficient residential buildings based on improved airtightness, increase insulation, use of materials, and possibly also new residential behaviour.
  .

3 WORK DESCRIPTION

Subtask 5 will investigate and identify relevant case studies through a literature survey and run measurement campaigns to provide data for investigation and validation in ST 1-4. Several sites/climates will be proposed, and the field tests will include buildings declared as being energy efficient or recently refurbished to become so. The field tests will focus on testing and demonstrating in practice which low energy operational strategies can be used which will provide amenable indoor environments. Subtask 5 will as far as possible test buildings with the ventilation strategies, both current and novel, as identified in Subtask 4.

The test buildings will be inspected with respect to the building and interior materials, furnishing and occupants' activities. Special attention will be given to documenting the materials' emission status, i.e. checking for the use of low-emitting materials. Availability of information on ventilation system control and flows as well as the necessary energy consumption data will add to the assessment of IAQ in the buildings used in the case studies.

Building operations may be optimized with only small improvements to become more energy efficient but uphold the user comfort given as IAQ. New sensor technology and model based control will provide data for the proper ventilation for the future. By investigating possible adjustments of the building design, material use and ventilation system and control Annex 68 will deliver a feasibility study of the potential energy savings in the highly energy efficient residential buildings.

ST 5 will involve stakeholders. The field tests will be carried out in cooperation with industry partners from the previous subtasks and with building owners. The subtask will involve engineers and building owners/operators from the studied buildings

4 SUMMARY OF PLANNED ACTIVITIES

5.1 STATE OD THE ART AND MEASUREMENT STRATEGY: Summary of the literature review on the necessary parameters, eventually accompanied by guideline values, necessary to describe the IAQ in the tested buildings. Development of a methodology to obtain the relevant values needed to study, simulate and verify IAQ in highly energy efficient residential buildings.

5.2 CONTROLLED MEASUREMENTS: In labs and test houses available at the universities and institutes involved in Annex 68.

5.3 IN SITU MEASUREMENTS: Examples of residential buildings from different geographical regions, and which are either new buildings or existing (possibly retrofitted) buildings will be chosen for investigation. It shall be possible in the chosen buildings to interact with the relevant operational parameters, e.g. for ventilation control, and to monitor the relevant performance parameters for energy consumption and indoor environment.

5.4 ANALYSIS AND DISSEMINATION: The results of ST5 will demonstrate and analyse residential buildings which achieve optimal energy and good indoor environmental conditions under various climatic situations.

RCLOUD - CAPTURING THE MOMENT, A NEW ERA IN AUTOMATED TESTING

Colin Genge

Retrotec Energy Innovations Ltd
306-2695 Granville Street,
Vancouver, BC, V6H 3H4,
CANADA

1 ABSTRACT

Hand-written results are a thing of the past. See how your phone or tablet and common testing gear can perform an automated cloud based test with secure data storage. Retrotec's new rCloud app allows technicians use their iPhone or 4G tablet to perform a fully automated blower door or duct test.

The rCloud app is more than an automated testing, it provides an entire data set from exactly where, when and the conditions of the test. It "captures the moment"

The app uses GPS location services built-in to the phone/tablet to confirm the testing location and places location on a live map. It collects local weather and housing information from local resources. The rCloud connects directly to the DM32 smart gauge to perform any number of tests - then securely stores all this data and images on a remote server. Share the results immediately thru email or view the test online.

Additional concepts-

Individual accounts,

Multiple testing protocols,

Learning objectives & take a ways.

1. How the test data and conditions at the time of the test can be captured, stored and shared from the job site.
2. How captured data can be directly imported/exported into local or municipal data bases to determine compliance.

3. The required tools or common phone/tablets, intuitive software and equipment are affordable to perform automated testing and cloud storage of test data.

4. This type of testing is the future of testing and provide Quality Control in the industry and will soon be a requirement.

UNCERTAINTY IN AIRFLOW RATE ESTIMATION OF DAYTIME VENTILATION ASSOCIATED WITH ATMOSPHERIC STABILITY

Jongyeon Lim*, Ryozo Ooka, and Hideki Kikumoto

The University of Tokyo
4-6-1 Komaba, Meguro-Ku
Tokyo, Japan
**Corresponding author:jylim@iis.u-tokyo.ac.jp*

ABSTRACT

We conducted observations of wind velocity profiles above a high-density area in Tokyo, Japan, using a Doppler LIDAR system. Obtained data of the exponent index for the power law, which is commonly used to describe the wind velocity profile, displayed diurnal variation, decreasing in the daytime, which is expected in unstable atmospheric conditions. This paper provides information on the uncertainty in the calculated ventilation airflow rate due to the use of a constant value for the exponent index. The study was performed with data on wind pressure coefficients obtained from a numerical parametric study using computational fluid dynamics based on observation data. The uncertainty was assessed based on comparison of the ventilation airflow rate calculated using a constant value for the exponent index, and the ventilation airflow rate calculated considering a diurnal change in the exponent index. The results indicate that the ventilation airflow rate obtained from a constant value for the exponent index for an isolated building with two openings is underestimated by up to 8% in the daytime. A large relative uncertainty occurs at a lower height, i.e., a large relative error of the approaching wind velocity and a resulting error in the wind pressure coefficient.

KEYWORDS

Uncertainty analysis; ventilation performance; wind profile; power law; wind pressure coefficient.

1 INTRODUCTION

The local wind velocity affects the ventilation performance. Since the wind data in weather data files are usually measured at a meteorological station at a given height, the approaching wind velocity $U(z)$ for each height z of the building surface is modified from the measured meteorological wind velocity by Eq. (1), which is well known as the power law:

$$U(z) = U_n \left(\frac{z}{z_n} \right)^{\alpha} \tag{1}$$

where U_n [m/s] is the wind velocity at the height of the meteorological measurement z_n [m] and α [−] is the exponent index. It is common in engineering applications to describe the wind velocity profile using the power law because of its simplicity. The exponent index for the power law (α in Eq. (1)) is regarded to depend on ground roughness, and is taken to be constant. Several data sources have provided the relationship between α and terrain types, e.g., 0.22 for urban terrains, or 0.33 for towns and cities, as found in the ASHRAE Handbook (ASHRAE, 2001). However, these standard values are based on a predominant mechanical turbulence (very strong wind). Panofsky and Dutton (Panofsky and Dutton, 1984) noted that α should be modified further when the contribution of convective turbulence becomes significant. This implies that the use of a constant value, e.g., 0.22 or 0.33 for α, contains an approximation error in calculating the approaching wind velocity when the effect of stratification becomes strong because of unstable atmospheric conditions. This error will contribute to the uncertainty in calculating the ventilation airflow rate Q [m^3/s], which is a function of wind velocity, as shown in Eq. (2),

$$Q = U_{ref} C_v A \sqrt{\Delta C_p} \tag{2}$$

where U_{ref} [m/s] is the reference wind velocity, which is often taken at the height of the rooftop in the free stream region, C_v [-] is the discharge coefficient of openings, A [m^2]is the area of the openings, and C_p [-] is the wind pressure coefficient.
The ventilation airflow rate is also a function of the wind pressure coefficient, which is defined as Eq. (3):

$$C_p = (P - P_0)/(\rho U_{ref}{}^2 / 2) \tag{3}$$

where P [Pa] is the static pressure at a given point on the building surface, P_0 [Pa] is the static reference pressure of the free stream, and ρ [kg/m^3] is the air density.
It is difficult to perform an accurate evaluation of C_p (Hensen, 1991) because of the various influencing parameters, including building configuration, details of the building surface, surrounding elements, and the characteristics of the approaching wind. Recently, with increasing application of computational fluid dynamics (CFD) to study the flow field around buildings, evidence where CFD has been used as a source of custom C_p data for building energy simulation has surfaced (Choi et al., 2012). However, when the inlet flow for a wind tunnel experiment or CFD is defined by the power law, the approximation error mentioned above, which results from using a constant α value, will also contribute to uncertainty in the C_p value. From Eq. (3) it is clear that C_p does not depend on the reference wind speed (U_{ref}), but the difference in α can cause changes in the approaching wind velocity profile, and subsequent changes in the wind pressure profile on the building surface. However, several studies have actually defined the inlet flow by the power law (Choi et al., 2012; Huang et al., 2007).
It can be concluded that the uncertainty of α effects the estimation of both U_{ref} and C_p values in Eq. (2), and consequently leads to uncertainty in calculation of the ventilation airflow rate Q in Eq. (2). This paper quantifies the uncertainty in the calculated wind-driven ventilation airflow rate due to diurnal changes in the approaching wind velocity profile. This diurnal variation is described by the fluctuation of the exponent index α for the power law, which is obtained from observations of the wind velocity profile using a Doppler LIDAR system (DLS).

2 OBSERVATION OF WIND VELOCITY PROFILE USING DOPPLER LIDAR SYSTEM

The wind velocity profile data used here were collected from a DLS (WindCube8, manufactured by LEOSPHERE) that was setup on the rooftop of the Institute of Industrial Science of the University of Tokyo, Japan (35°39'46"N, 139°40'41"E, 27.5 m altitude). The field of about 1-km radius surrounding the DLS is comparatively flat, and is mainly occupied by residential housing with varying heights of 3–9 m (73.8%), and a few buildings with heights over 30 m (0.5%). The mean height of the roughness elements is about 7 m, and the standard deviation of the heights of the roughness elements is about 4 m.

The observations were conducted from September 1, 2013 to June 30, 2014. The DLS used in this observation transmits a pulsed laser with a wavelength of 1.54 μm, receives the light backscattered by aerosols such as dust and other particles in the air, and measures the line-of-sight component of wind velocity using the Doppler frequency shift of the backscattered light. The orientation of transmission changes in the four cardinal directions, so that three components of wind velocity can be calculated. Using this DLS, the wind velocity data from 67.5 m to 527.5 m (20 m apart, 24 altitudes) were obtained with a temporal resolution of about 30 seconds.

The vertical component of measured wind velocity is one or more orders of magnitude smaller than the horizontal components. Hence, this analysis applies only to the horizontal components. We use wind velocity in this paper to refer to the scalar quantity of the horizontal velocity components. In inhomogeneous regions such as dense urban areas, the surrounding urban morphology often varies with direction, causing different wind profiles in each direction. However, in this paper, we discuss the wind profile obtained from the scalar quantities of the horizontal velocity components, without consideration of wind direction.

A total of about 3.5 million steps of data were obtained. Figure 1 shows the data acquisition ratio at each altitude for all observation periods, and for each month. The data acquisition ratio varies with altitude, with low data acquisition at low and high altitudes, and high data acquisition at medium altitudes. For data accuracy, we only used data obtained from time steps that had wind velocity data for all 24 altitudes. Furthermore, a large drop in the data acquisition ratio was found in winter, which we suggest is caused by the decrease in aerosol concentration. The overall data acquisition ratio was 40.9%.

Figure 3 shows the profiles of hourly averaged wind velocities and overall data, plotted on logarithmic axes. Each data point was normalized by z_{nDLS} and U_{nDLS}, which are the altitude with the highest data acquisition ratio (247.5 m) and the wind velocity at z_{nDLS}, respectively. Describing the wind velocity profile with a power law may be reasonable because linear relationships between velocity and altitude are apparent on the logarithmic axes. However, the wind profile varies by the hour. Since the slope of each line in Figure 3 corresponds to the exponent index for the power law (α in Eq. (1)), the value of α changes by the hour.

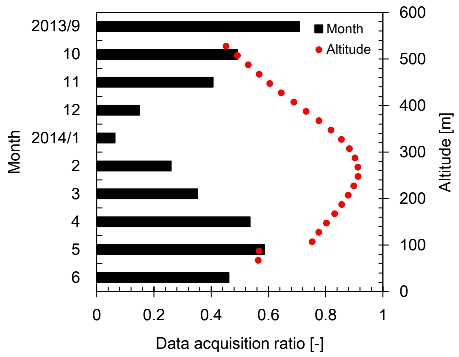


Figure 1: Data acquisition ratio at each altitude for all observation periods, and for each month.

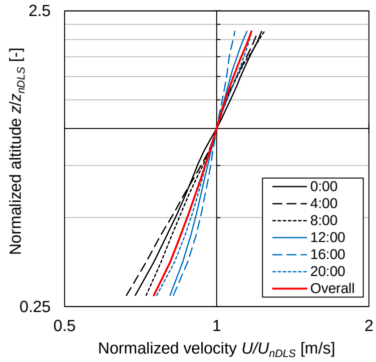


Figure 2: Mean profiles of hourly averaged and overall horizontal wind velocity.

Figure 3 shows the exponent index α derived from the mean profiles of hourly averaged and overall wind velocity data. Each value of α was calculated with a least squares error fit, which minimized the difference between the measured wind velocity at altitude z and $U(z)$ expressed by Eq. (1). The expected value of α for the study area is approximately 0.2 to 0.3, based on the terrain type. The exponent index α derived from the mean wind velocity profiles of all data in the observation period was 0.2, which is within the range. The values of α at night-time and early morning, which were expected to be neutrally stratified or stable, were also within the range. However, the values of α decreased to 0.1 in the daytime, displaying the diurnal variation. This result is consistent with the previous observations (Touma, 1977; Farrugia, 2003), which reported that the value of α decreases under unstable atmospheric conditions. The effect of unstable stratification in the daytime is also suggested as a major factor in causing diurnal variation in our observations.

The fact that α decreases to 0.1 in the daytime can increase the uncertainty in the calculated daytime ventilation. This paper quantifies the uncertainty in the calculated ventilation airflow rate due to the use of a constant value as the exponent index α. The constant value used here is 0.22. In this paper, uncertainties in the calculated ventilation airflow rate for September 2013 and May 2014 in the observation period are presented and analysed in detail. The reasons for selecting May and September are outdoor air during these periods is considered to have high potential for natural ventilation, and the data acquisition ratio is higher than other periods, as shown in Figure 1.

Figure 4 shows diurnal changes in the exponent index and data acquisition ratio for September 2013 and May 2014. Although samples for September 2013 were larger than for May 2014, there are no significant differences between the hours for each month. Diurnal changes in the exponent index for both September 2013 and May 2014 are presented with the same aspect as for the exponent index derived from the profiles of hourly averaged wind velocity for the entire observation period (see Figure 3). The value of α decreased in the daytime in September 2013, displaying a minimum (α = 0.08) at 15:00 and a maximum (α = 0.31) at 23:00. We dealt with data from May 2014 (α = 0.07 at 14:00, and α = 0.39 at 06:00) in the same manner.

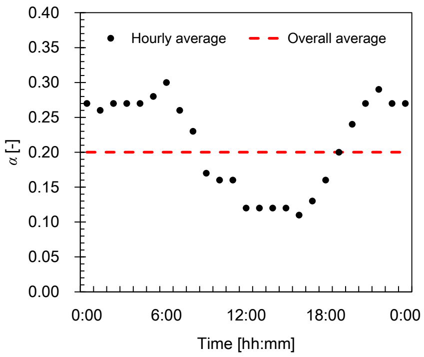


Figure 3: Exponent index derived from profiles of hourly mean and overall mean horizontal wind velocity.

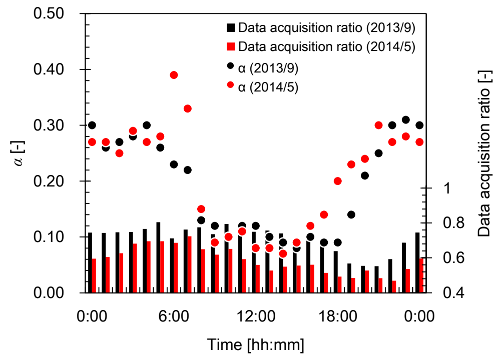


Figure 4: Exponent index derived from profiles of hourly averaged values and the data acquisition ratio in September 2013 and May 2014.

3 RESULTS

Ventilation performance is largely affected by conditions outdoors, such as wind velocity, which is usually obtained from weather data files. These data are modified using Eq. (1), and are used as the approaching wind velocity in building energy simulations. In this section, we present the uncertainties associated with using a constant value as the exponent index in such processes. To achieve this analysis, a 20-story building model with the dimensions 30 m (x) × 20 m (y) × 70 m (z) was designed. Although various weather data files have been generated to estimate building energy consumption, in this study we used the meteorological measurements of September 2013 and May 2014 in Tokyo because the exponent index data used in the analysis are the measured values for those periods. The uncertainties discussed in this section are uncertainties in the wind pressure coefficient on building surfaces, and the resulting uncertainty in the calculated ventilation airflow rate in the model building.

3.1 Relative wind pressure coefficient (C_p) error

In this section, the error of the value of C_p from using a constant value of α is quantified using comparisons with C_p values obtained using time-dependent values of α. The constant and time-dependent values of α used here are 0.22 and the observation result obtained from the DLS, respectively. We used the database from the computational parametric study using CFD as the C_p data source.

This study uses the parametric approach to obtain the C_p database with the designed building model. In the parametric study, cross-comparison of the effects of the diurnal wind profile variation is more easily achieved by observing changes in C_p on the building surface for different values of the exponent index α in the power law.

Five groups, comprising 23 values of α, are setup in five wind directions (0°, 22.5°, 45°, 67.5°, and 90°) and were simulated by the standard k-ε models. The α values were returned from the diurnal changes in α in Figure 4, but with no repetitions.

The building model is a 20-story building model as mentioned above. The size of the computational domain is 300 m (x) × 720 m (y) × 350 m (z). The total number of meshes was about 3,600,000, on a case-by-case basis. Tetrahedral meshes were employed, and finer meshes were arranged in the region near the building and ground surfaces, whereas the meshes were expanded farther away from the solid surface. The minimum mesh size near the building was 0.5 m. The inlet boundary condition was imposed by the power law as shown in Eq. (1), where U_n = 3.16 m/s, which is the averaged wind velocity during the two-month period (September 2013 and May 2014); z_n = 35.3 m, which is the height of the meteorological measurement for the data used; α was set to the above mentioned 23 values. We performed a total of 115 cases of CFD runs (23 values of α × five wind direction).

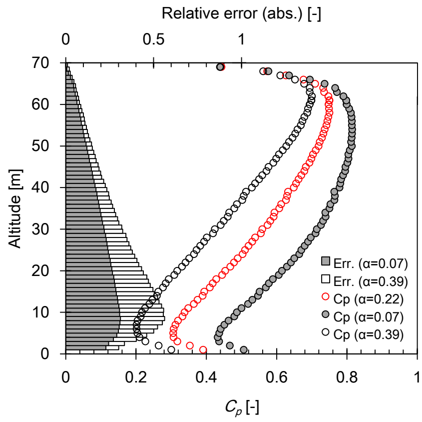


Figure 5: Vertical profile of C_p on windward surface for α = 0.07, 0.22, and 0.39, and relative errors.

Based on the CFD results, Figure 5 shows the vertical profile of C_p for the external wall on the windward side, where the wind is perpendicular to the south side of the building (0°), and the values of α for the inlet boundary are 0.07, 0.22, and 0.39. Each line probe has 69 data points spaced 1 m apart. The bar chart represents the relative error of C_p for α = 0.22 to that for α = 0.07 or 0.39. Using α = 0.22, C_p is underestimated when the value of α decreases under unstable atmospheric conditions. For α = 0.07, the relative C_p error reached 31%. In contrast, using α = 0.22 led to overestimation of C_p when the time-dependent values of α > 0.22. For α = 0.39, the relative C_p error reached 56%. The maximum error appeared at a height of about 10 m. Since the wind direction and the value of α vary by the hour, the calculation error is expected to vary with hour and altitude. Therefore, we estimated the root-mean-square error of ΔC_p ($RMSE_{CP}$) for hours and altitudes as defined in Eq. (4), where ΔC_p is the difference between the value of C_p on the windward and leeward surfaces:

$$RMSE_{CP}(t,z) = \sqrt{\frac{1}{D}\sum_{d=1}^{D}\left(\Delta C_p(d,t,z)_{\alpha=0.22} - \Delta C_p(d,t,z)_{\alpha=\alpha(t)}\right)^2} \tag{4}$$

where D is the day of a given month, $\Delta C_p(d, t, z)$ is ΔC_p for altitude z with the wind direction at hour t on the d^{th} day, obtained from the meteorological measurements for Tokyo, and $\alpha(t)$ is the time-dependent value of α at hour t, obtained from DLS observations. The value of $\Delta C_p(d, t, z)_{\alpha=0.22}$ is calculated using a constant value of α = 0.22, and $\Delta C_p(d, t, z)_{\alpha=\alpha(t)}$ is calculated using the time-dependent value $\alpha = \alpha(t)$.

Figure 6 shows the distribution of $RMSE_{CP}$ for September 2013 and May 2014. The value of ΔC_p was calculated based on the CFD result corresponding to the wind direction and the value of α at each hour. Based on data at 7:00 in September 2013 (see Figure 6(a)), since the value of α obtained from DLS observation was 0.22, $RMSE_{CP}$ was zero for all altitudes. However, at other times, the values of $\Delta C_p(d, t, z)_{\alpha=\alpha(t)}$ are different from ΔC_p $(d, t, z)_{\alpha=0.22}$, because $\alpha(t) \neq$ 0.22, and such errors seem to be a function of altitude and hour. Larger values of $RMSE_{CP}$ were presented in the lower region of the building and in the daytime, i.e., large overestimation or underestimation in the calculated ΔC_p.

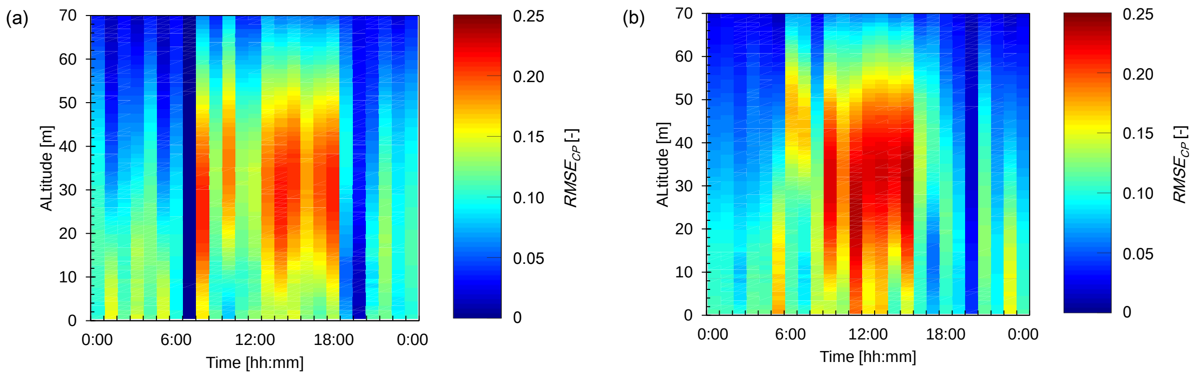


Figure 6: Estimation error of the wind pressure coefficient. (a) September 2013; (b) May 2014.

3.2 Uncertainty in the calculated ventilation airflow rate

In this section, the uncertainties in the calculated ventilation airflow rate are presented and analyzed in detail. To isolate the effects of the wind profile changes (i.e., the effect of the diurnal variation of α on the calculated ventilation airflow rate) from the effects of other parameters (e.g., characteristics of openings in the building), the building model was simplified. Drawing from Cóstola et al. (Cóstola et al., 2009), we designed a building model with the following assumptions: there is only one interior zone on each floor (no internal partitions); all floors have the same area and the same shape; there are only two openings for cross ventilation (no cracks in the external walls); all openings have the same area and the same discharge coefficient; buoyancy-driven ventilation is not considered.
Values of C_p on the face of each opening were obtained from our CFD database. We used surface-averaged C_p values of calculation grids corresponding to each opening. The ventilation airflow rate for each floor (1−20) was calculated using the C_p value obtained from the CFD database, and the approaching wind velocity was represented by a power law.

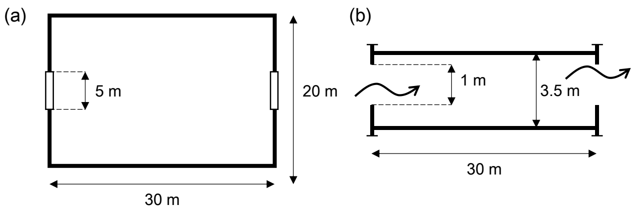


Figure 7: Geometry of a typical floor. (a) Floor plan; (b) cross section.

In this paper, only a situation with a pair of identical openings to achieve cross-ventilation was analyzed (see Figure 7). On each floor, ventilation is ensured by fully opened windows on the external walls, whose dimensions are 5 m × 1 m, and are characterized by a discharge coefficient of 0.7. The ventilation airflow rate was analyzed with the assumption that the pressure and velocity fields around the external walls are not changed by the presence of openings. Of course, depending on the size of the opening, it may significantly alter the pressure and velocity field in its vicinity. However, in this paper, we ignored the influence of the openings, and used the pressure and velocity field, which was determined in the absence of the opening, as boundary conditions for the airflow through the opening. In this case, the ventilation airflow rate (Q_{all}) can be calculated using Eq. (5):

$$Q_{all} = \sum_{f=1}^{20} Q_f = \sum_{f=1}^{20} U_{ref} C_v A \sqrt{\Delta C_p(f)} \tag{5}$$

where f is the floor number. In this paper, we regarded the ventilation airflow rate calculated by time-dependent α, $\alpha(t)$, as the "real" ventilation airflow rate. Therefore, the relative error of the ventilation airflow rate caused by the use of a constant value of α = 0.22 is defined as Eq. (6):

$$E_{ACH}(t) = \frac{Q_{all}\big|_{\alpha=0.22} - Q_{all}\big|_{\alpha=\alpha(t)}}{Q_{all}\big|_{\alpha=\alpha(t)}} \tag{6}$$

Each term on the right side in Eq. (6) is a function of α, which affects $\Delta C_p(f)$ and U_{ref} in Eq. (5). It is clear that $E_{ACH}(t)$ does not depend on the opening characteristics (C_v and A). Despite the fact that the openings in the designed building model are somewhat unrealistic, the dependence of the characteristics vanishes when $E_{ACH}(t)$ is used in the process of estimating the uncertainties of the calculated ventilation airflow rate.

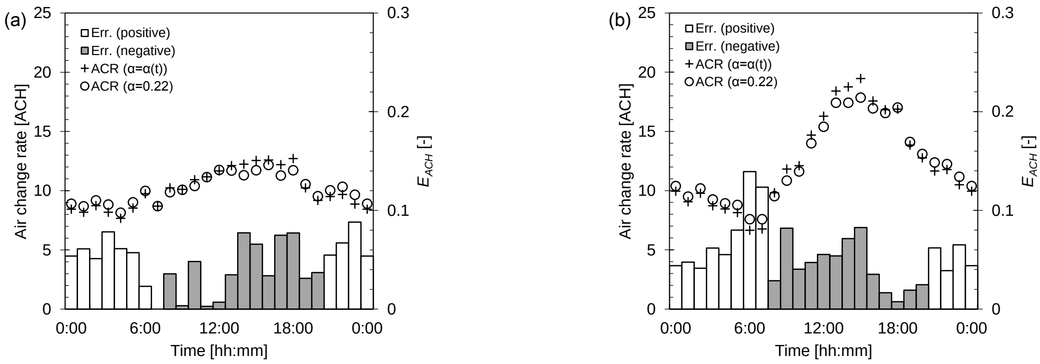


Figure 8: Calculated ventilation airflow rates and relative error. (a) September 2013 and (b) May 2014.

Figure 8 shows the calculated ventilation airflow rate Q_{all} converted to air change rate [ACH] and the relative error $E_{ACH}(t)$ for each hour. The air change rate tends to increase in the daytime because the wind velocities in the meteorological measurement data are higher in the daytime than at night. The magnitude of the relative error $E_{ACH}(t)$ varied by the hour, reaching about 14%, which is as large as the error from selecting C_p data sources. In addition, the air change rate was underestimated in the daytime and overestimated at night. Such results indicate that there may be no significant differences in the estimation of monthly overall ventilation performance in the case of 24-hour ventilation, i.e., underestimation in the daytime is at odds with overestimation at nighttime. However, several studies have investigated either daytime or nighttime ventilation (Choi et al., 2012; Ramponi et al., 2014). When C_p values derived from a constant value of α, determined only by terrain characteristics such as ground roughness, are used in such cases, the C_p error and resulting error of the ventilation airflow rate may be significant.

4 CONCLUSIONS

This paper presents an estimation of the uncertainty in the calculated ventilation airflow rate associated with changes in the wind velocity profile. Changes in the wind velocity profile are represented by a time-dependent exponent index for the power law, which is common in engineering application to describe the wind velocity profile. The ventilation airflow rate was calculated for a 20-story building model. The uncertainties that occurred in the process were quantified. The main conclusions are:

- From the observation of the wind velocity profile using a Doppler LIDAR system, we determined that the values of the exponent index for the power law were 0.2−0.3 in the nighttime and early morning, and decreased to 0.1 in the daytime, displaying a diurnal change. The exponent index derived from the mean wind velocity profiles of all data in the observation period was 0.2.

- The uncertainty in the estimated wind pressure coefficient corresponded to the exponent index. When the value of the exponent index was constant (0.22), the estimation accuracy was relatively high at nighttime, but the uncertainty became large in the daytime. Consequently, the estimation error for the ventilation performance varied by the hour, reaching about 14%, displaying a underestimation in the daytime and overestimation at night. The magnitude of the uncertainty is high, but the usability of these data depends on the problem, e.g., daytime and/or nighttime ventilation.

These results provide boundaries for further research on the wind pressure coefficient database. As a direction for improvement, we suggest classification by local climate and land use. Please refer to the main conclusions of the work.

5 ACKNOWLEDGEMENTS

A part of this study was supported by JSPS KAKENHI Grant Number 25-03368, 24226013, and 26709041. This financial contribution is highly appreciated.

6 REFERENCES

ASHRAE (2011). ASHRAE, ASHRAE handbook: fundamentals.

Choi, W., Joe, J., Kwak. Y., Huh, J. (2012). Operation and control strategies for multi-storey double skin facades during the heating season, *Energy and Buildings*, 49, 454–465.

Cóstola, D., Blocken, B., Ohba, M., Hensen, J. (2009). Uncertainty in airflow rate calculations due to the use of surface-averaged pressure coefficients, *Energy and Buildings*, 42, 881–888.

Farrugia, R. (2003). The wind shear exponent in a Mediterranean island climate, *Renewable Energy*, 28, 647–653.

Hensen, J. (1991). On the thermal interaction of building structure and heating and ventilating system, PhD thesis, Eindhoven University of Technology.

Huang, S., Li, Q., Xu, S. (2007) Numerical evaluation of wind effects on a tall steel building by CFD, *Journal of Constructional Steel Research*, 63, 612–627.

Panofsky, H., Dutton, J. (1985). *Atmospheric turbulence: models and methods for engineering applications*. New York: John Wiley & Sons.

Ramponi, R., Angelotti, A., Blocken, B. (2014). Energy saving potential of night ventilation: Sensitivity to pressure coefficients for different European climates, *Applied Energy*, 123, 185–195.

Touma, J. (1977). Dependence of the Wind Profile Power Law on Stability for Various Locations, *Journal of the Air Pollution Control Association*, 27, 863–866.

AIRTIGHTNESS QUALITY MANAGEMENT APPROACHES IN FRANCE: END AND BIRTH OF A SCHEME. PREVIOUS AND NEW SCHEMES OVERVIEW AND ANALYSIS.

Sandrine Charrier[*1], Jocelyne Ponthieux[2]

1 CEREMA – Direction Territoriale Center-Est
46 rue Saint Théobald – BP128
38081 L'Isle d'Abeau Cedex
France

**Corresponding author:*
Sandrine2.charrier@cerema.fr

2 CEREMA – Direction Territoriale Center-Est
Boulevard Giberstein
BP 141
71405 AUTUN CEDEX
France

ABSTRACT

Since 2006, the French Energy Performance regulation, named RT, has been allowing two ways to justify building airtightness: either with a measurement or with the application of a quality management approach. The quality management approach certification is managed by the French Ministry in charge of construction, for which it set up a specific expert committee to assess quality management approaches. Since 2012, the justification has been compulsory for residential buildings. This obligation led to a more systematic use of certified quality management approaches.

This paper aims at presenting the analysis of the current process, its results and the improvements to be scheduled in the next certification process which will be managed by accredited certification bodies.

The first part of this paper explains the certification process and tools. It presents the yearly follow-up requirements and the control process content. The second part presents and analyses the certified bodies characteristics and the evolution of measured values along the years. This part gives information about the type of bodies implied in French quality management approaches and the way they imply themselves in the yearly application of their approach. The third part deals with the analysis of the whole certification process: certification it-self, yearly follows-up and control campaign. This part shows the results of the processes and states pros and cons of the current process. A last part presents the new scheme for the certification obtainment. Indeed, to cope with the increasing number of applications, starting July 2015, French authorities decided to entrust accredited certification bodies under contract with the government to deliver these quality management approach certifications. An overall analysis of the current process is made, leading to the implementation of the new certification scheme. Both are presented in the paper. Moreover, the first decisions taken by the accredited certification bodies to implement their quality management approaches certification are also given.

The paper concludes on the overall analysis of the quality management scheme that has been implemented since 2006 in France, its pros and cons, and the evolutions decided for the new certification scheme. The current scheme has been improved year after year and is now robust enough to be transferred to private bodies. Moreover, some new developments were needed which could not be implemented by the French authorities, and which will be set up thanks to the new scheme.

KEYWORDS

Airtightness, Quality Management, Certification, Control, Energy Performance Regulation

1 INTRODUCTION

In 2006, the French energy performance (EP) regulation (RT 2005) introduced a significant reward on the overall building energy performance assessment when justifying a better-than-default value for the air permeability of the envelope. Then, in 2012, the RT 2012 compelled residential buildings airtightness to be better than limit values and to be justified.
The French indicator for the building envelope airtightness is Q4PaSurf, which is the airflow rate at 4 Pa divided by the cold surface area (excluding the lowest floor). In the RT 2012, the airtightness level of residential building envelope must not exceed 0.6 $m^3.h^{-1}.m^{-2}$ at 4 Pa for single-family buildings. A better-than-default value can be used, provided it is justified. The French EP regulation gives two options to justify the building airtightness level:

- Either with a systematic measurement, performed by a certified tester;
- Or by the application of a certified quality management approach (QMA) on the building airtightness, that allows to test only a sample of buildings.

The RT 2012 obligation of justifying residential building airtightness level led to a more systematic use of certified QMA.

QMA contents are described by Charrier (Charrier, 2014). The underlying basis of an airtightness QMA is to implement a scheme that lasts from the genesis of the building project to its commissioning and that ensures that the building airtightness will not exceed a limit value. This limit value must be better or equal than regulatory requirements. The QM approach is based on a precise description of "who-does-what-when-and-how". In addition, each step must be traceable and traced.

French authorities are in charge of delivering airtightness QMA certifications. To reach this goal, the French government has implemented a committee (named the "Annexe VII" committee) in charge of analysing the approaches and proposing the certification. This committee has existed since 2007. In 2014, French authorities decided that the QMA certification had to be transferred to private certification bodies, because of the success of the process and the limited State resources to go on managing the certification process. The transfer is planned on July, 2015.

This paper aims at presenting the analysis of the State process, its results, pros and cons after 8 years application, and the next certification process which will be managed by accredited certification bodies.

This paper is organized as follows. Section 2 presents the State certification process since 2007: its organization and gradual improvements. Section 3 presents the way the market answered to the State requirements. Section 4 presents the analysis of the State process: the results and the process pros and cons. Finally, section 5 presents the new certification process, embodied by accredited certification organisms under contract with the government and the limits.

2 STATE CERTIFICATION PROCESS FOR AIRTIGHTNESS QMA: IMPLEMENTED PROCESS AND TOOLS

As presented by Charrier (Charrier, 2014), the Annexe VII committee is composed of a dozen of national experts in charge of analysing the consistency with the authorities requirements, expressed in a ministerial ruling, the Annexe VII of the RT 2012 (JO, 2010). To this end, the committee is in charge of three main deadlines: the certification obtainment, the certification yearly follow-ups, and the certified QMA controls. This part aims at presenting the whole process, organization and tools that the Annexe VII committee implemented.

2.1 Organization and tools for the airtightness QMA analysis

The analysis of airtightness QMA is based on the evaluation of all the files describing the QMA. These files are analysed regarding the authorities requirements. For subjects that are not dealt in the ministerial ruling, the Annexe VII committee proposes precision points, previously discussed and validated. Those precisions are officially diffused via a French authorities web site.

In order to analyse the airtightness quality management approaches, the committee is organized as follows: a couple of experts analyse the same approach, thanks to an evaluation grid. Then, all the committee experts meet to make a final and collective decision. The collective meeting is an important part of the process because the analysis is based on a subjective files analysis. The collective final decision guarantees the treatment homogeneity.

The analysis is based on an evaluation grid described by Charrier (Charrier, 2014). This evaluation grid is also available on the public web site used by French authorities to communicate about their airtightness QMA requirements. This communication enables candidates to be aware of the way the applications will be examined. Gradually, this evaluation grid has been completed by a precise description of what is expected for each criterion. Indeed, each expert could have its own understanding of each criterion, which led to heterogeneous evaluations. Then, as presented in part 3, the committee experts analyse 4 main QMA templates, coming from 4 main studies centers, barely adapted to each builder organisation. To get homogeneous evaluations between similar applications, the evaluation grid has been completed by the repeating comments for every similar application. This grid is often sent to studies centers so that they could improve their QMA templates, for all their clients. Since 2012, so as to meet the increasing number of certifications requests, the Annexe VII committee has had to improve its organisation. Those improvements are described by Charrier (Charrier, 2014). Finally, the Annexe VII committee rule is to propose a certification or a certification denial for each candidate to French authorities, who take the final and official decision.

2.2 Organization and tools for the yearly follow-ups

The certification is valid one year and is renewable yearly if the builder provides a yearly follow-up. Requirements for this follow-up are defined in the ministry rule JO (JO, 2010) and based on the fulfilment of the documents that trace the QMA application.
The yearly follow-up analysis is based on a files analysis, as for the certification. In 2014 and 2015, to strengthen the yearly follow-up, the Annexe VII committee required to submit all the QMA tracing files for two non-compliant buildings. Indeed, the goal is to check whether the certified QMA is actually applied. Moreover, in 2015, the Annexe VII committee restricted the yearly follow-up evaluation period to 6 months, from January to June. Likewise, the

committee Annexe VII restricted the number of evaluations to 2, whereas before there could be up to 4 evaluations. Indeed, in 2014, some renewal had been pronounced in November or December, when builders had to submit their new follow-up file one month after. This shortened delay for follow-ups validation compelled studies centers to submit complete files.

2.3 Control process

In 2011, in order to strengthen the certified QMA reliability, the Annexe VII committee implemented control campaigns which are described by Leprince (Leprince, 2011) and Charrier (Charrier, 2013). Those control campaigns are performed by public, independent and neutral controllers. They are divided into two parts: a qualitative control, aiming at verifying the actual application of the certified QMA on some building; and the quantitative control, consisting in measuring the airtightness value on a sample of buildings. Control campaigns target only a sample of certified QMA. In the first control campaign (2011-2012), the qualitative control only verified if tracing documents were fulfilled or not. In the second control campaign (2014) the file analysis has been strengthened either by analysing each file contents or by being audited *in situ*. Indeed, one major limit to the Annexe VII committee is that evaluations are only based on a files analysis. Sometimes direct oral exchanges could improve the application understanding. 2014 control campaign implemented an *in situ* quality audit, for 4 builders. The current (2015) control campaign strengthened the requirements on the ending of works. Indeed, measurements for previous control campaign were quite often performed on unfinished buildings, at least with impacting airtightness works not completed. Once the control campaign ended, an individual results and decisions report is provided to each controlled builder.

3 MARKET ANSWER TO AIRTIGHTNESS QMA CERTIFICATION

Since 2006, airtightness QMA has been implemented by one main stakeholder: single family houses builder. Indeed, among 101 certified airtightness QMA, only one is applied by an envelope industrialist on single and multi-family dwellings. To be more precise, Figure 1 presents the distribution of the number of certified builders depending on the number of dwellings built per year. The distribution suggests that it remains interesting for builders producing over 50 houses per year, the trade-off being somewhere in the range of 20-50 houses per year.

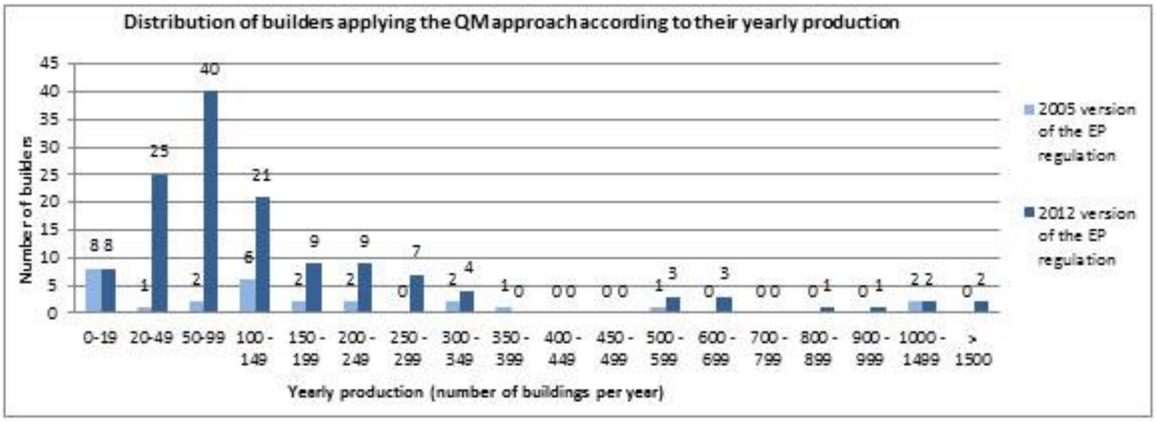


Figure 1: distribution of builders applying airtightness QMA according to their yearly production (number of buildings per year)

Then, Figure 2 shows that 60% of certified builders ask for a limit value equal to the authorities limit value (0,6m^3/ (h.m²)), and 40% ask for a better-than-regulatory limit value, between 0,3 and 0,5 m^3/ (h.m²). Figure 2shows more precisely how the QMA airtightness limit value is distributed with the certified builders yearly production. We can notice that builders with 0,6 or 0,5m^3/ (h.m²) airtightness limit value are more likely builders with a yearly production between 100 and 250 buildings. Builders with 0,4 m^3/ (h.m²) airtightness limit value are more likely builders with a yearly production between 50 and 100 buildings. It must be noticed that such limit value builders are well represented by builders with a yearly production higher than 1000 buildings.

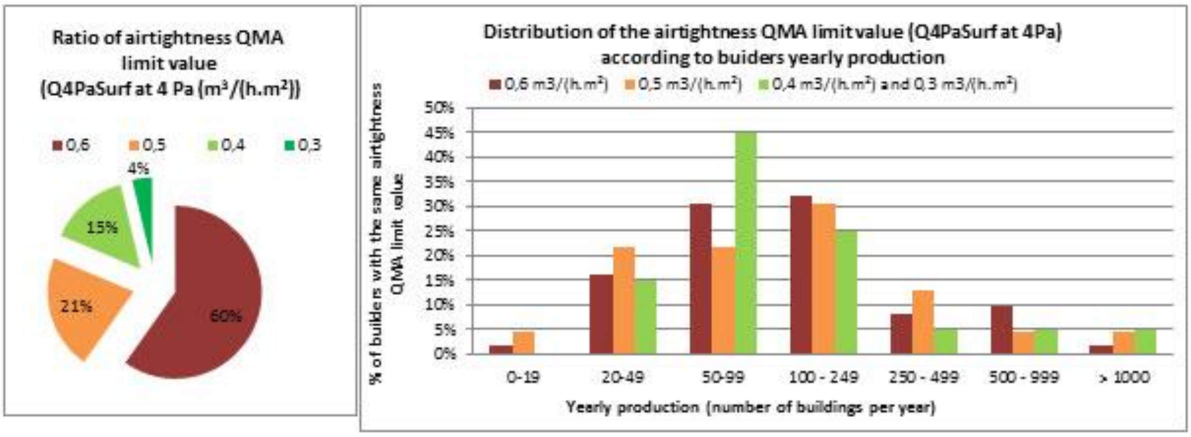


Figure 2: distribution of the airtightness QMA limit value according to builders yearly production

Another market adaptation to the airtightness QMA authorities requests has been the development of airtightness QMA templates by four main studies centers. Indeed, some studies centers decided to help builders in implementing their airtightness QMA and applying it. To this end, they implemented airtightness QMA templates, more or less adapted to each builder organization. Moreover, these studies centers sold in a whole package the yearly *in situ* ISO 19011 audit, and the yearly sample measurement that each candidate has to perform for the certification request. This market answer is controversial. Indeed, on the one hand, it enabled to help builders to implement a QMA and for French authorities to get so many builders requests. But on the other hand, this led to a lack of builders implication in their QMA. Moreover, as studies centers implement and contribute to the application of the QMA, their audit and measurements objectivity might be questionnable.

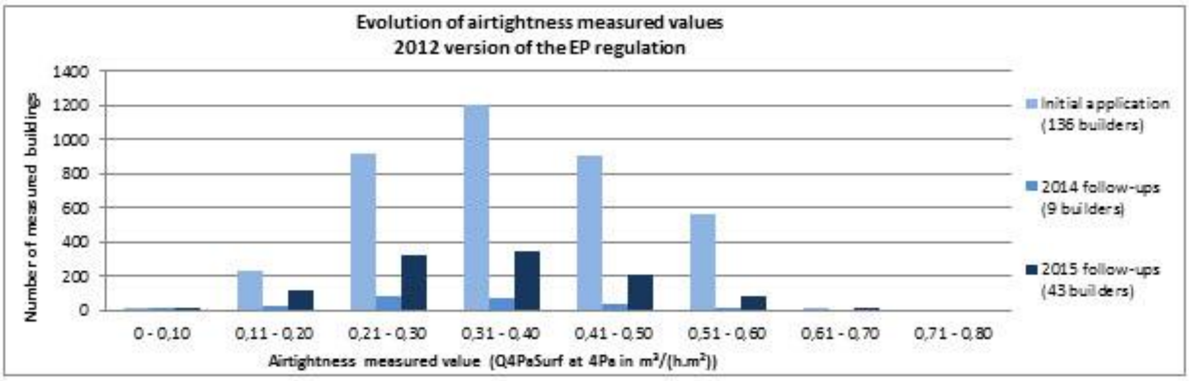


Figure 3: evolution of airtightness measured values

The last market analysis is about the evolution of the airtightness measured values with year. Indeed, the basis of quality management approaches is the ongoing approach, its contents and the results, improvement. As a consequence, measured values improvement was expected year after year, for certified builders.

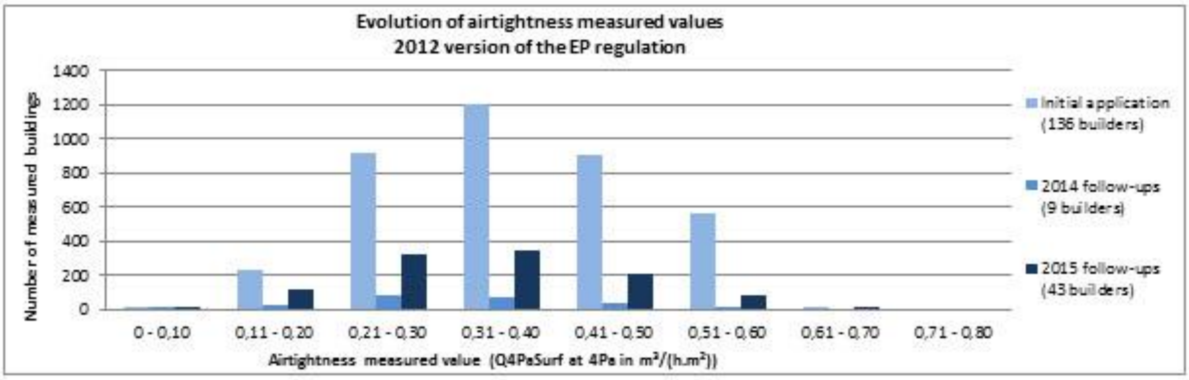


Figure 3 shows that such improvement is not obvious. The measured values improvement has been requested for 2016.

Finally, it must be noticed that, althought French requirements enable ducts airtightness QMA, no certification has been asked for such QMA. This can be explained by low reward in the EP calculation for a better-than-default value and no subsidies to encourage such approaches implementation.

4 STATE CERTIFICATION PROCESS: EFFICIENCY, PROS AND CONS

This part deals with the analysis of the State process efficiency, analysing the 3 main processes: the certification process, the yearly follow-up one and the control campaign one.

4.1 Certification process efficiency

Since 2006, the committee Annexe VII has always been evolving so as to improve its efficiency and to get more homogenous and objective analysis. This evolution is shown in

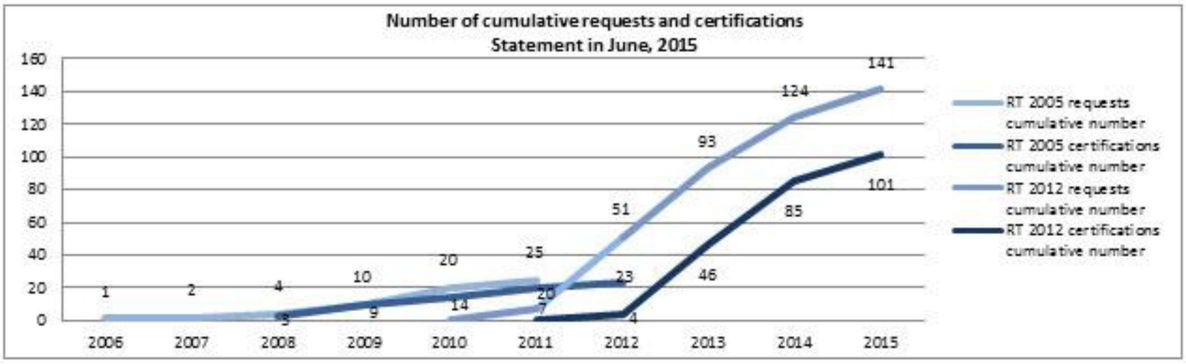


Figure 4 that represents the evolution of the number of certification requests, in RT 2005 and RT 2012, and the number of certifications, year after year. We can notice that the committee annexe VII evolutions have been effective as the number of certifications and requests evolutions have the same slope.

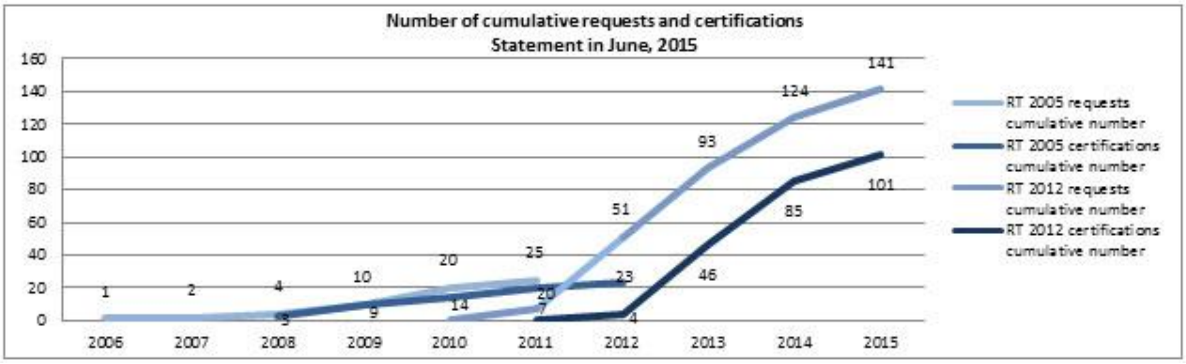


Figure 4: Requests and certifications cumulative number

Then, concerning delays between the request and the State certification, they were stable in 2012 and 2013, around 10 months, thanks to the committee evolutions. The delay is now around 8 to 9 months, for the same reasons.

Another marker of the process efficiency is that QMA encourages applicants to secure a minimum airtightness level. This statement is supported by Figure 5 which shows that both the median value of the airtightness levels and the spread of the distribution are significantly smaller for buildings subjected to a QM approach.

Figure 5: distribution of measured airtightness of houses with and without implementations of a certified QM approach

The feedback of the State certification process can be summed up by the pros and cons of the State process in Table 1.

Table 1: Pros and cons of the State certification process

Pros	Cons
✓ Homogeneous analysis	✓ Limited capacities of French authorities to meet the demand
✓ Analysis objectivity and neutrality, thanks to a double analysis and to a common final decision.	✓ Files analysis limits. No direct exchange with the candidate. Sometimes, misunderstanding between the committee requests and the builder answers.
✓ Improving and efficient process	✓ State certification is cost free, which implies that some builders ask for a certification whereas their QMA is not complete or well applied.

	✓ The audit ISO 9001 is carried out by the studies center that proposed its QMA template. The yearly ISO 9001 audit objectivity is questionable. The same applies for measurements.

4.2 Yearly follow-up process feedback

As described in part 2.2, the yearly follow-up process evolved in 2014 and 2015. In 2014, 11 upon 11 approaches have been renewed. In 2015, 40 upon 49 have been renewed. The number of ended certifications has increased in 2015. Reasons are various:

- 1 did not sent its yearly follow-up files, despite various requests;
- 4 builders asked for the end of the certification, after the control campaign results or because, as small production builder, the cost of the QMA was higher than a systematic measurement;
- 4 builders did not succeed after the two committee analysis.

A feedback of the 2015 follow-up analysis is the limit of files analysis, as described in part 4.1. Another feedback is about the committee limitations to 2 committee analysis per follow-up. This decision compelled studies centers to implement complete follow-up files. Moreover, delay feedback is positive. Indeed, it was adapted to the number of answers expected and no builder has been rejected because of the out of time answer.

4.3 Control campaign process feedback

Results of control campaign process are positive. One of the first feedbacks comes from controlled builders. Indeed, since 2011, a majority of certified builders positively welcome the control campaign. Reasons are double: first, it enables direct exchange with French authorities. Then, it puts forward well applied QMA. This is once more true that results of the control campaign led to a double-label summing-up certificate. This report is once more used as a reward by builders that they do not have other certificate to put forward the certification obtainment.
Then, another feedback is that, very often, state controllers had to perform a measurement on a building with airtightness affecting works unfinished. For instance, those works could be dedicated to the client, but the certified QMA did not precise anything about such work. This really questions the reliability of the sample measurements performed by the studies centers.

Next, the synthesis of the control campaign is summed-up in a double label presented in Figure 6. In both control campaigns, decisions have been hard to take. For instance, which decision to take when the builder succeeded to only one of the two controls, like for instance builder 2 or builder 9? For builders who failed the qualitative control, it has been decided to warn them and either control them once again in 2015, or strengthen the 2015 follow-up analysis. For builders who failed the quantitative control, decision has been taken to strengthen the required sample to test in 2015 (not 5 + 10% of the yearly production but 10 + 10%). For information, studies centers and builders expected a more demanding sample, which could have been half of the production, in order to compel the builder to well apply its QMA and to encourage him to a reduction of the sample the year after, in case of good results. Finally, after the second control campaign, some difficulties appear to take a sanction decision. This might be due to the limited controlled sample (5 to 7 buildings).

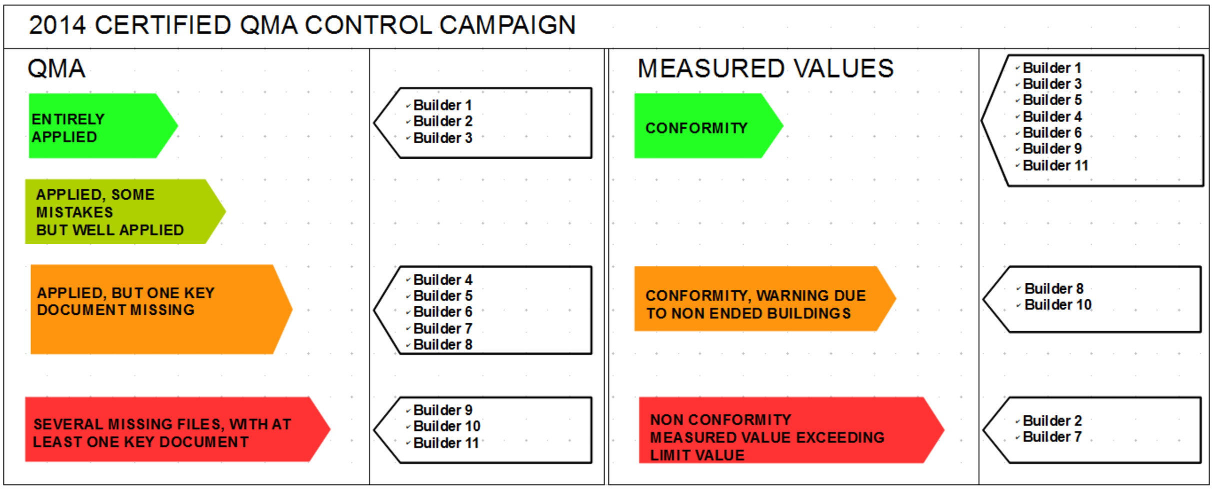


Figure 6: 2014 control campaign results

Finally, some inconsistencies appear between the control campaign and follow-up results. Indeed, the Annexe VII committee pronounced the cancellation of some certification which had succeeded completely during the last control campaign, and vice versa. This strengthened the limits of the files analysis and of the decisions to be taken with the control campaign results, which are unclear today.

As explained in part 1, the French authorities decided to transfer the Annexe VII committee. The next part explains the requirements for certification organisms and the way the annexe VII committee helped to define new rules and requirements.

5 CERTIFICATION ORGANISMS PROCESS: PROCESS AND LIMITS

The certification organisms are dimensioned to manage the number of requests. Moreover, they are used to treat certification requests.

5.1 Scheduled process

JO (JO, 2014) specifies airtightness QMA technical requirements, for any candidate. These technical requirements are exactly the same as previous process ones. JO (JO, 2014) also specifies the requirements for certification organisms. Any certification organism has to propose a certification program, which has to be validated by French authorities and by the accreditation organism. A certification organism can deliver airtightness QMA provided 1/ he is engaged with French authorities, by a contract, 2/ he is allowed, by the accreditation organism COFRAC, to deliver airtightness QMA certifications. The certification organism has to implement three processes: a certification process, a follow-up process, every two years, and a control campaign (every certified builder has to be controlled every 2 years).
This new scheme aims at improving the State process. As a consequence certification organisms will perform ISO 19011 *in situ* audits on their own. It will substitute to the files analysis. The same applies for follow-ups and certification renewal. Moreover, the certification is not free anymore, and we can expect that candidates files will be of better quality. Finally, the control campaign will be applied every 2 years, on all the certified builders. In the previous process, around 10% to 15% were controlled every year.

5.2 New process limits

Because of the certification organisms organisation and in the negotiations with French authorities, exactly the same process could not be transferred to certification organisms and some less demanding point have been accorded.
For instance, the sample has been divided by 2. Indeed, the sample is calculated according to the production of the last 12 months and is asked every 2 years. In the previous scheme, the sample had to be calculated on the last period since when the QMA had been analysed. This presents a huge reduction of the tested sample, which was yet a small part of the production (5 + 10% of the production, if yearly production < 500 buildings).

Then, the neutrality and homogeneity of each analysis is questioned. Indeed, each candidate will be audited by one expert only. The expert will make a report, which will be analysed by a technical director. No committee is organised, which could have guaranteed treatments homogeneity between candidates. No evaluation grid has been submitted to French authorities. French authorities are only involved in a yearly meeting with every contracted certified organism, so as to deal with repetitive or blocking problems. The committee annexe VII had insisted to make jurisprudence but nothing has been retained.

Finally, the control process requires only a measurement control, and not a control on the application of the certified QMA. The success and strength of a QMA rely on the fact that it is a whole process that is applied on the whole life of a project. The process enables an actual quality and improvement. Measurement of the efficiency at the end of the construction is only one step among others. Moreover, the control campaign will be applied on all the certified QMA, every 2 years, which strengthens the control. However, the number of measurements is very low (3 for production > 500 buildings, against 7 in the current control campaign). Finally, the decision taken by certification organisms in case of non-complying measured value is not clear and French authorities did not fix any minimum decision to take (cancellation, sample strengthen).

6 CONCLUSIONS

The paper presents the end and the birth of processes for the airtightness QMA certification. The old process, the State one, has gradually evolved so as to meet the growing demand and to improve the certification reliability. This process has been very efficient, but suffered some deficiencies that French authorities could note solve: limited capacities to meet the demand, limits of files analysis, limits of decisions taken after control campaigns. To solve these deficiencies, the airtightness QMA certification has been transferred to private and accredited certification organisms. Table 2 presents the two processes pros.

Table 2: French authorities and certification organism processes pros

Process	French authorities	Certification organisms
Certification process	✓ Homogeneity	✓ Audit in situ: better understanding of the approach
	✓ Neutrality	✓ Organised to meet the demand
	✓ Jurisprudence	✓ Delays between request and certification (around 4 months)

Follow-up process	✓ Yearly follow-up ✓ Verification of the actual application of the QMA on buildings ✓ Possible consequences: certification cancellation	✓ Audit in situ ✓ Complete analysis
Control Campaign	✓ Control of the application of the QMA and measurements	✓ Every builder is tested within 2 years

If a country wants to implement such airtightness QMA, the two times process seems efficient: first a State authorities certification process, which is improved with experience, and then a private certification organism process.
Next steps for French airtightness QMA are 1/ trying to guarantee continuity between the two processes, 2/ trying to guarantee compliance with regulatory requirements, 3/ trying to get homogeneity between evaluations and between certification organisms and 4/ trying to encourage ducts or ventilation QMA, that are scheduled by EP regulation.

7 ACKNOWLEDGEMENTS

The contribution of the Cerema/ DT Center-Est to the implementation of this quality framework is funded by the ministry of ecology, sustainable development and energy (MEDDE). The responsibility for the content of this publication lies with the authors. It does not necessarily reflect the opinion of the Ministry. All national experts who participate to the Annexe VII committee, all the Cerema controllers that contributed to the quality management control campaigns are gratefully acknowledged for their mobilisation and contributions.

8 REFERENCES

Journal Officiel de la République Française (JO) (2014). *Arrêté du 19 décembre 2014 modifiant les modalités de validation d'une démarche qualité pour le contrôle de l'étanchéité à l'air par un constructeur de maisons individuelles ou de logements collectifs et relatif aux caractéristiques thermiques et aux exigences de performance énergétique applicables aux bâtiments collectifs nouveaux et aux parties nouvelles de bâtiment collectif*. Journal Officiel de la République Française.

Charrier, S., Ponthieux, J., Huet, A. (2014). *The Airtightness Quality Management Approach in France – Assessment after more than Five Years of Operation*. International Journal of Ventilation. ISSN 1473-3315. Vol. 13(2), pp. 125-140

Leprince, V., Biaunier, J., Carrié, F.R., Olivier, M. (2011). *Quality management approach to improve buildings airtightness. Requirements and verification*.32nd AIVC Conference and 1st TightVent Conference. Brussels, Belgium. 12-13 October 2013.

Charrier S., Huet A., Biaunier J. (2013). *Control of airtightness quality management scheme in France: results, lessons and future developments*. 34th AIVC Conference and 3rd TightVent Conference. Athens, Greece. 25-26 September 2013.

Journal Officiel de la République Française (JO) (2010). *Arrêté du 26 octobre 2010 relatif aux caractéristiques thermiques et aux exigences de performance énergétique des bâtiments nouveaux et des parties nouvelles de bâtiments*. Journal Officiel de la République Française.

THERMAL ENVELOPE QUALITY VERSUS NZEB PARAMETERS AND LONG-TERM ECONOMICS: THE ECO-SILVER HOUSE CASE IN LJUBLJANA

Miha Tomšič[*1], Andraž Rakušček[1], Miha Mirtič[1], Luka Zupančič[1] and Marjana Šijanec Zavrl[1]

1 Building and Civil Engineering Institute ZRMK
Dimičeva 12,
1000 Ljubljana, Slovenia
**miha.tomsic@gi-zrmk.si*

ABSTRACT

In 2014 the first multi storey residential building planned and constructed to meet the Passivhaus Institute (Darmstadt) criteria was put in operation in Ljubljana, the capital of Slovenia. This massive-structure building is part of the FP7 EE-Highrise project, aiming to demonstrate nearly zero energy building (nZEB) technologies, an integrated design concept, and advanced systems for sustainable construction. The aim was to test and assess the technological and economic feasibility of innovative energy solutions on a large-scale real project (www.ee-highrise.eu). The construction phases had to be meticulously prepared in order to reach the goals and meet the targets defined during the integrated planning process and selection of most viable design options. This included among other special on-site training and coaching of construction workers, preparation of how-to written protocols, and performing various non-destructive tests and measurements within the established quality assurance scheme.

As two of the most important objectives were to reach the passive house standard according to the PHPP methodology and to possibly comply with the national nZEB boundary values while securing a high level of thermal comfort special attention was put also on the airtightness and avoidance of thermal bridges on the thermal envelope, both in planning and implementation phase. The air exchange rate (n50) strongly influences the energy use and affects the efficiency of ventilation system with heat recovery, conditioning of occupied spaces and overall performance of smart control units in each apartment. A high level of airtightness and absence of thermal bridges are two of the vital preconditions for reaching the nZEB standard after having had other parameters successfully fine-tuned.

Through a step-by step process of repetitive Blower door tests in consecutive construction and installation phases all weak point were identified, solutions for sealing proposed and implemented, corrective measures tested and the final state approved. The same approach was used for identification and remedy of thermal bridges using infrared thermography. The goal was not just to achieve remarkable energy indicators on paper, but on the actual building. The paper describes the main principles and implementation of the above mentioned approach.

Another point of consideration was the long-term economics of selected solutions, i.e. an analytical study of the influence of various building quality parameters (like air exchange rate and thermal bridges). The question was how is the energy use according to different scenarios reflected in the long-term costs. An LCC analysis based on parametric sensitivity studies about the influence of individual thermal envelope features was conducted to find out which of them is the most significant. The aim was twofold, namely to see what is the impact level of the quality of specific work procedures and, indirectly, skills of workers on operational costs, and to see if

common guidance for future similar projects can be extracted from the results. The paper explains selected findings and point out the main lessons learnt to be considered on the way towards nZEB standards and sustainable buildings.

KEYWORDS

Integrated design, quality control, airtightness, nZEB, costs.

1 INTRODUCTION

The Eco-Silver House (ESH) is a multi-residential high-rise building with 128 flats located in the city centre of Ljubljana, Slovenia (Figure 1). It is part of the FP7 EE-Highrise demonstration project, aiming to demonstrate new nearly zero energy building (nZEB) technologies, an integrated design concept and systems for sustainable nZEB construction in order to test and assess the technological and economic feasibility of innovative energy solutions within a real high-rise multi-residential building project (www.ee-highrise.eu). The ESH was designed as two rays forming a shape of an inverted letter »L«. The net total area covers a surface of 23.455 m^2 over 17 floors (12.870 m^2 net treated floor area). The building is a private investment built for the market and its concept aims to correspond to the specific needs of potential buyers looking for an intelligently controlled and eco-oriented building in a passive standard (according to the Passivhaus Institute criteria).

Figure 1: Eco-Silver House – the design concept and the completed building (Design: Akropola; Photo: Milan Tomažin).

During the design and construction phase the focus of the demonstration project was on implementation of integrated energy design and quality assurance (QA) protocols. The scope of the on-going post-occupancy research is monitoring of the energy performance indicators, comfort parameters and users' satisfaction as well as assessment of sustainable building indicators.

Among other actions substantial efforts were put on avoidance of thermal bridges and securing a high level of airtightness, which both positively contribute to energy indicators of the building, to operational efficiency and proper functioning of certain mechanical equipment, and of course to easier control of indoor comfort parameters.

2 THE MAIN ECO-SILVER HOUSE FEATURES

The energy concept of the building allows cost-independent performance of individual apartment units where excellent insulation of thermal envelope and dynamic shading as well as the possibility of intelligent control of indoor conditions ensure that each apartment can function like an independent passive unit. The U values of the building envelope are 0,17 W/(m^{2}K) for walls and 0,14 W/(m^{2}K) for the flat roof, while the U value of a triple glazed (U_g = 0,58 W/(m^{2}K)) window of standard dimensions is 0,83 W/(m^{2}K). The airtightness level (n_{50}) of building sectors measured with Blower door test is between 0,45 h^{-1} and 0,59 h^{-1}, i.e. bellow the design value of 0,6 h^{-1}.

The building is connected to the energy efficient municipal district heating, with wood biomass co-burning and cogeneration. In addition to the main district heat station each apartment has its own heat substation for space heating and domestic hot water preparation, while electricity is used for operation of mechanical ventilation with heat recovery with a system efficiency of 0,85 and for an air to air inverter heat pump for preheating or precooling of the inlet air. The heat is emitted via radiators and in larger apartments also partly via floor heating. Cooling needs are negligible in the standard usage profile. A multi-split air conditioning system can be installed in larger flats on request and depending on the particular usage pattern.

The fundamental principles of sustainable design of the building are reflected in the demonstration project through comprehensive planning of energy efficiency features with respect to passive house standard criteria and national nZEB criteria. The important features include utilisation of renewable energy sources, very good thermal insulation, high-performance air-conditioning system with heat recovery, dynamic sun protection, intelligent control and management of electric and mechanical devices, machinery and tools, and a significant share of ecological materials. Building owners use rainwater stored in a roof tank (60 m^3 of storage covers a 10-day sanitary water demand and brings 500 m^3 of fresh water savings), a micro solar power plant (34 kWp) on the roof, green roof, and e-mobility with car sharing (Figure 2). The building was completed in 2014, and in 2015 it entered the post-occupancy monitoring period.

Figure 2: Eco-Silver House – PV power plant on the flat roof and the intelligent control unit in each apartment (Photo: Borut Slabe).

3 NATIONAL nZEB CRITERIA

Slovenia implemented minimum requirements for energy-efficient buildings in the 2010 Building Codes based on the recast EPBD. In the 2008 Regulation (PURES 2008) an intensive reduction of transmission losses through the building envelope as well as new requirements on the obligatory 25% use of renewable energy sources in the final energy use were introduced. The 2010 Building Codes (PURES 2010) built on the 2008 version and placed the focus also on the calculation of primary energy and CO_2 indicators, set additional minimum requirements for the primary energy for heating, limited the heating and cooling needs, both in terms of useful and primary energy, and added many new minimum requirements for energy systems.

From January 2015 more severe minimum requirements for maximum energy needs for heating have entered into force according to the long term perspective integrated in the rules of the PURES 2010 Building Code. Minimum requirements are expressed using performance-based requirements, energy-related requirements, and detailed technical requirements for building components and systems. By the end of 2015 the Building Codes are planned to be revised with respect to the outcome of the national cost optimal study and amended and upgraded with further details accompanying the national definition of nZEB.

The climate in Slovenia is very diverse, which results from many intertwined factors. This is visible from the distribution of degree-days on the map in Figure 3. It is determined among other by the geographic location, type of terrain, orientation of mountain ridges and proximity to the sea. There are three predominant types of climate in certain areas but their effects are intertwined: the eastern Slovenia has a temperate continental climate, the central part a sub-Alpine (Alpine in the mountains) one and west of the Dinaric-Alpine barrier there is a sub-Mediterranean climate. Over 90% of inhabitants live in the regions with more than 2800 degree days.

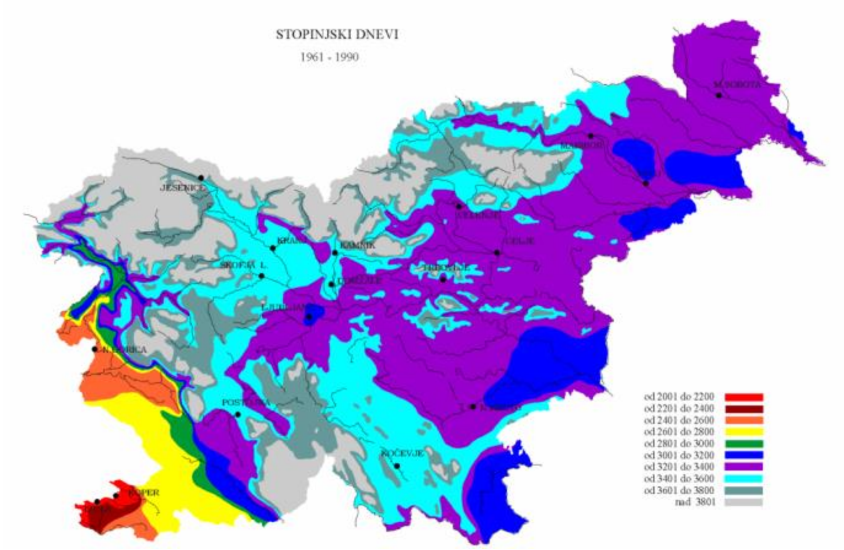


Figure 3: Spatial distribution of degree-days in Slovenia *(Mekinda - Majaron, 2002)*.

The national definition of nZEB *(Ministry of Infrastructure, 2014)* was formulated based on the cost-optimal study of minimum energy efficiency requirements *(Šijanec Zavrl et al, 2014)* with consideration of building tradition, availability of technologies for nZEB and with respect to the Slovenian climate.

Table 1: Minimum requirements for nearly zero energy buildings (nZEB) in Slovenia

Building Type	**Minimum requirements for nZEB Primary Energy Q_p (kWh/(m^{2}a)) and Share of Renewable Energy (%)**		
	New building Max. Q_p kWh/(m^{2}a)	**Major renovation Max. Q_p kWh/(m^{2}a)**	**Min. share of renewable energy %**
Single family house	75	95	50
Apartment building	80	90	50
Non-residential building [a]	55	65	50

a. Appliances and equipment not included.

The National definition of nZEB is based on a cost optimal study for reference buildings where the primary energy as a core nZEB performance indicator is complemented with the criterion of achievable target of 50% share of renewables in the final energy use, selected with consideration to the nZEB acceptable technologies and available renewable energy sources. In future the use of RES will be increased due to growing share of RES in district heating systems that are subject to comply with 2020 energy efficiency targets set in the Energy Act. In addition to that, the nearly zero or very low amount of energy required is achieved by further limitation of energy need for heating to a maximum value between 25 kWh/(m^{2}K) and 15 kWh/(m^{2}K), with respect to the shape factor and climate on the location. Although not directly prescribed the very high energy performance of nZEB will be demonstrated with nZEB building ranked in class A1, A2 or B2 based on the building heating needs.

The nZEB definition provided minimum requirements for primary energy (for all energy use according to Directive EPBD Annex I, including lighting in residential building and excluding appliances and other energy use but EPBD-related) for new buildings as well as for major renovation, for single family houses, apartment buildings and for non-residential/office buildings (Table 1). An nZEB action plan with the national definition of nZEB was accepted by the government in April 2015.

4 ENERGY INDICATORS ACHIEVED

The Eco-Silver House, as built, fulfils the commonly accepted passive house standard characteristics with a PHPP annual heating demand of 14 kWh/(m^{2}a) and a total primary energy use for building operating systems including electrical household appliances of 106 kWh/(m^{2}a). According to the national energy performance certificate ESH is ranked in energy performance class A1 with a standard annual heat demand of 8 kWh/(m^{2}a).

Figure 4 shows the total delivered energy amounting to 51 kWh/(m^{2}a) and the primary energy to 71 kWh/(m^{2}a). The latter number is reduced to 55 kWh/(m^{2}a) when exported energy from the PV power plant is considered. These are values calculated using the prescribed national method and national primary energy conversion factors for various energy carriers. The ESH presents a new standard of the energy-efficient apartment building in the Slovenian construction sector, meeting the 2015 definition of nZEB, which set the maximum allowed primary energy for apartment buildings to 80 kWh/(m^{2}a). The graphs below show also a comparison of the ESH indicators with the “state-of-the-art” ones, which would be reached using a standard approach according to current regulation.

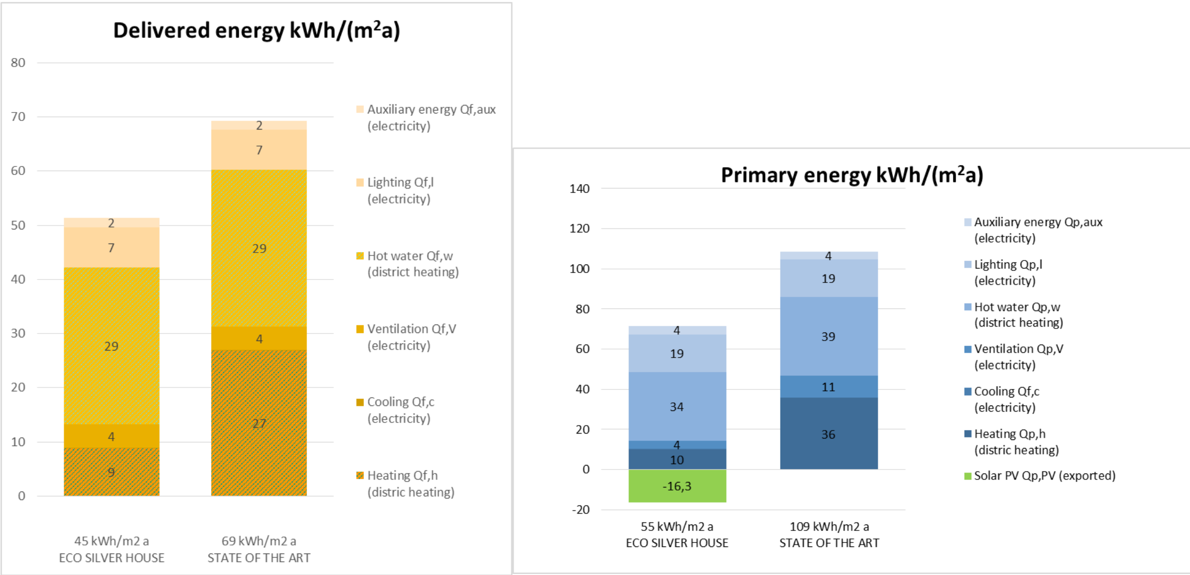


Figure 4: Delivered and primary energy for the ESH demonstration building and a (reference) state of the art new building in Slovenia.

5 QUALITY CONTROL AT THE CONSTRUCTION SITE – AN EXAMPLE FOR AIRTIGHTNESS AND (CONVECTIVE) THERMAL BRIDGES

Already before the start of the actual construction work it was clear that due to special demands and high targets of this project special attention will have to be put upon the quality of individual work tasks. The design process therefore included numerous sessions where the details of individual stages were discussed and defined. One of the key segments was to ensure provision of adequate knowledge and upgraded skills to construction workers to enable them to meet the set targets. Quality assurance and quality control had to be integrated in all phases from design to the finalization of the building (Figure 5).

Figure 5: ESH (FP7 EE-Highrise) QA steps from paper to the construction site.

To ensure the optimal results the demonstration project accommodated pilot trainings for on-site workers, where various approaches were tested. The QA process for nZEB during the construction phase covered: training of on-site working teams; video streaming of best practice applications of nZEB technologies; special coordination meetings of on-site workers, contractor, designers, investor and QA team; selected quality control on site (Figure 6). During the construction phase the weak points of the installation of nZEB technologies were identified (i.e. continuity of the thermal envelope, sealing of penetrations, installation of air-tightness layers within the building volume, installation of windows according to the RAL guideline). The QA was based on the plan-do-check-act principle where bottlenecks in nZEB-related skills were identified and missing experience integrated within corrective actions.

The integration of the FP7 demonstration EE-Highrise project and the understanding of relatively rigid performance and feedback of the construction sector studied in detailed in the Build Up Skills Slovenia project provided added value for both – reaching nZEB targets in demonstration as well as smooth provision of better skilled workforce.

The benefits of implementation of QA are reflected not only in the technical aspects but also in improved processes and better skilled workforce as well as in profound understanding of barriers specific to various occupations and workers' profile. The theoretical background of nZEB planning was integrated into construction process by defining of various protocols, where comprehensive knowledge and skills in the whole production chain (investor, designer, contractor, quality control, technology provider etc.) are essential for a success of such demonstration project.

The importance and benefits of consecutive quality testing were expressed through constant improvements of the results. The workers implementing newly acquired knowledge were made aware of needed improvements of certain techniques and proper use of specific materials in real time, so it was to a large extent a learning-by-doing process.

Figure 6: Preparation and execution of Blower door tests.

The gradual but steady improvement of airtightness levels was visible through repeated Blower door tests, where findings of each session served workers as a guidance on how to remedy the performance (example in Table 2 and Figure 7).

Table 2: Air exchange rate (n_{50}) from Blower door tests done during the initial external quality control at demonstration phase, during internal control by the building contractor and during the final external quality control at the building completion

INITIAL EXTERNAL

Airtightness zone – apartment or sector description	Apartment or sector code	Area (m^2)	Date	n_{50} (h^{-1})
3-bedroom	C-1-1	69.59	21.11.2013	1,04
2-bedroom	C-1-2	57,82	21.11.2013	0,88
3-bedroom	C-1-3	81,40	21.11.2013	1,92
4-bedroom	C-1-4	98,90	21.11.2013	1,36
Sector C-1	C-1-1, 2, 3, 4	401,00	12.5.2014	0,77
Sector C-1	C-1-1, 2, 3, 4	401	24.7.2014	0,58

INTERNAL

Sector description	Sector code	Area (m^2)	Date	n_{50} (h^{-1})
C-1-1, 2, 3, 4	C-1	401	12.5.2014	0,77
C-2-1, 2, 3, 4	C-2	401	30.4.2013	0,65
C-3-1, 2, 3, 4	C-3	401	21.11.2013	0,68
C-7-1, 2	C-7	278	21.11.2013	0,88
C-8-1, 2	C-8	229	21.11.2013	1,92
C-9-1	C-9	165	21.11.2013	0,86

FINAL EXTERNAL

Flats in sector	Sector code	Area (m^2)	Date	n_{50} (h^{-1})
C-1-1, 2, 3, 4	C-1	401	24.7.2014	0,58
C-2-1, 2, 3, 4	C-2	401	21.7.2014	0,58
C-3-1, 2, 3, 4	C-3	401	23.7.2014	0,59
C-4-1, 2, 3, 4	C-4	401	23.7.2014	0,59
C-5-1, 2, 3	C-5	350	24.7.2014	0,56
C-6-1, 2, 3	C-6	347	24.7.2014	0,57
C-7-1, 2	C-7	278	24.7.2014	0,60
C-8-1, 2	C-8	229	23.7.2014	0,60
C-9-1	C-9	165	23.7.2014	0,56

Figure 7: Airtightness sectors C-1 (apartments C-1-1, C-1-2, C-1-3, C-1-4), B-1, A-1 covering three staircases of the 1st floor of the residential part.

6 BUILDING ENVELOPE QUALITY AND ITS FINANCIAL ASPECTS

A high-end project in performance terms featuring also demonstration elements as described above has a specific financial framework, but need not be high-end in costs as well. This is especially true when looking at the long-term impact of implemented energy efficiency solutions affecting operational costs. A number of particular ESH features and solutions were checked for their impact, but there are also other, non-measurable elements to be recognised such as indoor comfort, well-being and satisfaction of the dwellers. On the other hand the work force involved in the construction acquired new knowledge and skills, which makes them competent for implementation of further nZEB (or similar) works and highly competitive on the construction market.

Here we present only one example of different scenarios, the one related to the manner of installation of windows. Installation according to the RAL guideline was prescribed and implemented, which is indeed more costly than the standard procedure. But, this upgrade strongly influences the airtightness of the thermal envelope and consequently the energy demand for heating. It also supports proper functioning of certain mechanical systems.

We analysed three airtightness scenarios with corresponding investments:

- The business as usual approach considered no special attention given to installation of windows, and an airtightness of $n_{50} = 3{,}00\ h^{-1}$ (a conservative value based on long-term experience from the national building stock).
- The basic integrated design approach considered on-site training without Blower door tests and use of moderate class sealing materials and techniques. According to the results of preliminary Blower door tests (see the excerpt in Table 2 above) an assumption was made for an average building envelope airtightness of $n_{50} = 1{,}50\ h^{-1}$.
- The advanced integrated design approach considered on-site training with consecutive Blower door tests performed at selected project milestones (pilot apartment, confirmation of adequate sealing, internal control by the building contractor, and final external quality control) and use of state-of-the-art sealing materials and techniques, which as a result brought the airtightness level down to $n_{50} = 0{,}55\ h^{-1}$.

The surplus costs for RAL installation depend in practice also on the scope of the operation and on particular investor-contractor arrangements. The total installation length in ESH was 5.782 m, and the surplus cost for moderate class sealing materials and techniques was estimated to 6 EUR per meter, and to 15 EUR per meter for the state-of-the-art sealing performance.

The comparison of results based on data from Table 3 is shown in the spider chart below (Figure 8), which compares parameters such as payback period, CO_2 emissions, primary energy for heating and DHW.

Table 3: Input data and results for evaluated scenarios.

Window installation (total perimeter) length	5782 m
State-of-the-art cost surplus (compared to business as usual) per unit length $n_{50}=0{,}55\ h^{-1}$	15 €/m
Moderate class cost surplus (compared to business as usual) per unit length $n_{50}=1{,}50\ h^{-1}$	6 €/m
Training costs	22.500 €
Blower door plus internal QA	15.000 €

Scenario	**Airtightness n_{50} (h-1)**	**Primary Energy for Heating and DHW per TFA (kWh/(10*m²a))**	**Annual CO_2 emissions per TFA (kg/(10*m²a))**	**Payback period in years**
Business as usual ($n_{50}=3{,}00\ h^{-1}$)	3,00	5,1 kWh/(m²a)	1,7 kg/(m²a)	0,0
Basic ID approach ($n_{50}=1{,}50\ h^{-1}$)	1,50	4,0 kWh/(m²a)	1,3 kg/(m²a)	5,0
Advanced ID approach ($n_{50} = 0{,}55\ h^{-1)}$	0,55	3,5 kWh/(m²a)	1,1 kg/(m²a)	8,3

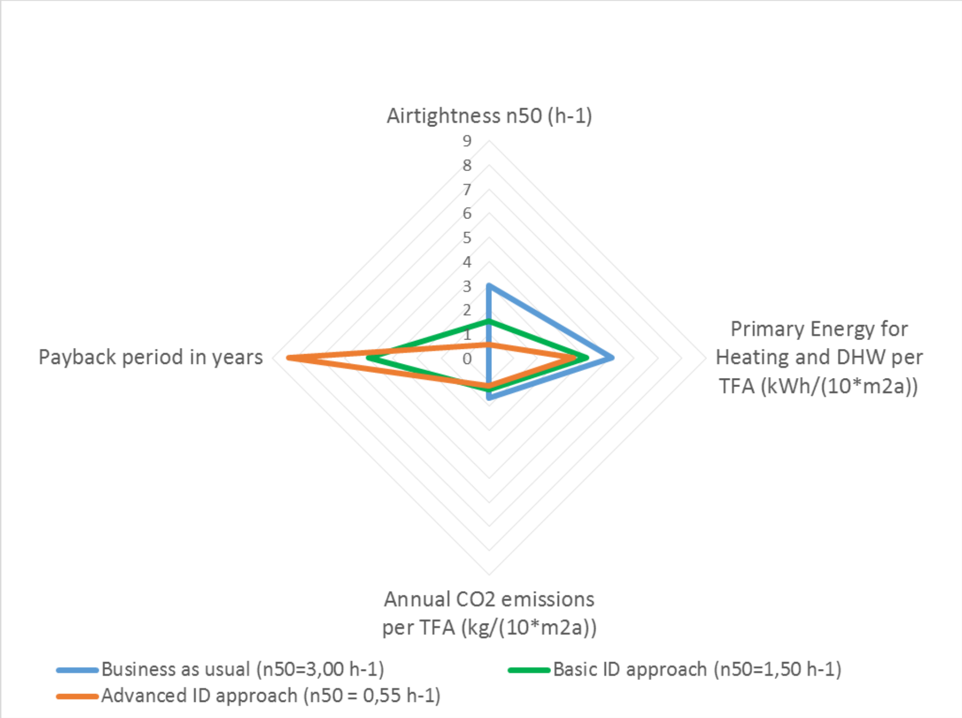


Figure 8: The comparison of parameters for different scenarios.

7 CONCLUSIONS

The Eco-Silver House/FP7 EE-Highrise project clearly showed the necessity of an integrated design approach including a QA scheme when rigorous targets are set for energy indicators. The key problem of nZEB lies not in availability of products and technologies, but in knowledge, competences and skills of planners and contractors, accompanied by understanding and at least basic technical knowledge of the investor. The situation can be more critical when smaller markets are in question where specialisation of the construction sector is less developed also for economic reasons.

The demonstration part of the project also showed that such actions cannot be satisfyingly implemented without clearly defined quality assurance protocols prepared in advance. Training of workers and monitoring of construction stages with guidance on remedy procedures in case of inadequate results are vital parts of nZEB construction.

The state-of-the-art procedures (i.e. protocols, training, monitoring, repeated work tasks) described in the paper did consequently affect the construction price, but not to an extent which would bring the initial decision in question. The activities proved to be sensible both from the aspect of energy indicators and cost parameters. For example, the presented installation of windows according to RAL guidelines with all due materials and procedures contributed to reaching very favourable energy indicators including complying with the national nZEB requirements, while the payback period for these additional efforts including QA-related costs remains notably short and points out the cost-effectiveness of the approach.

It has to be stressed that airtightness and particularly installation of windows was just one of many segments and their impacts considered within the project, and that the interaction of various interventions was studied in advance, too, in order to achieve well-balanced results related to energy, economy, and indoor comfort.

8 ACKNOWLEDGEMENTS

The demonstration part of the Eco-Silver House project is supported by the European Commission within the 7th Framework Programme FP7 EE-HIGHRISE (FP7-2011-NMP-ENV-ENERGY-ICT-EEB) (2013-2015). The authors acknowledge the significant effort of the investor and design team at Akropola in development of the project as well as the contribution of contractor's (HTZ) group in the demonstration part of construction.

9 REFERENCES

Mekinda – Majaron, T. (2002) From: *Klimatografija Slovenije*, http://www.arso.gov.si/vreme/poro%C4%8Dila%20in%20projekti/dr%C5%BEavna%20slu%C5%BEba/Stopinjski_dnevi_in_trajanje_kurilne_sezone.pdf

Ministry of Infrastructure (2014). http://www.energetika-portal.si/fileadmin/dokumenti/publikacije/an_snes/an_snes_slovenija.pdf

Šijanec Zavrl, M., et al (2014). Strokovne podlage za določitev stroškovno optimalnih ravni za minimalne zahteve glede učinkovitosti z uporabo primerjalnega metodološkega okvira in strokovne podlage za Nacionalni akcijski načrt za skoraj nič energijske stavbe za obodbje od leta 2012 do leta 2020. Ljubljana: Gradbeni inštitut ZRMK.

THE ZERO PRESSURE PARADOX

Dr.ir. N.J. Bink[*1], Peter Lok[1] and ir. W.V. Struik[1]

1 ACIN instrumenten B.V
Handelskade 76
Rijswijk, the Netherlands
**Corresponding author: nj.bink@acin.nl*

ABSTRACT

The zero pressure compensation method has proven to be the best method to measure air flow rates accurately although it also has be shown that the accuracy depends on the type of air terminal device and how and where the pressure to be compensated is measured in the instrument. Although the principal of the zero pressure method implies universal applicability, in practice this does not seem to be case. This has lead us to develop the 'extended' zero pressure method

KEYWORDS

Zero-pressure method, Powered Flow Hood, compensation

1 INTRODUCTION

To balance ventilation systems accurate measurements of supply and return flows at a large spectrum of air terminal devices (ATD's) are needed. Measuring supply flows is challenging because of the uneven velocities coming from different parts of the registers. Passive flow hoods and anemometers can be extremely inaccurate because they typically mistake turbulence for additional flow and tend to read high. Return flows are much easier to read but if return pressures are low, the presence of a passive flow hood can increase the pressure in the return and force flow to divert to other registers where it is unmeasured causing low readings.
A Powered Flow Hood (PFH) is supposed to solve both of these problems by reducing the pressure next to the register to ´zero´ by a powered fan whose speed is adjusted automatically. This measuring method is therefore often referred to as the zero pressure compensation method. It has proven to be the best method to measure flow rates accurately ((Caillou, 2014), (Stratton, Walker, & Wray, 2012), (Wray, Walker, & Sherman, 2002) and (Knights & Gilbert, 2015)).
In theory 'zero pressure compensation' refers to an amount of compensation as if the PFH is no longer 'visible' for the flow. But of course it **is** present. In practice, the pressure to be compensated is difficult to measure and on top of that the needed pressure compensation for supply and return flow is fundamentally different. We refer to this tension between theory and practice as the 'zero pressure paradox'.
In order to understand better what is needed for accurate measurements independent of ATD, flow direction and flow intensity, fundamental research using Computational Fluid Dynamics (CFD) and measurements on a variety of ATD's have been carried out. This resulted in an

enhanced understanding of the complexity of different flows encountered by a PFH and what ´zero pressure´ in fact should be.

2 ZERO PRESSURE METHOD

The zero-pressure method was originally developed in the early seventies by Bitter & Detzer (Bitter & Detzer, 1974) as a means to accurately measure air flow quantities at inlets and outlets.

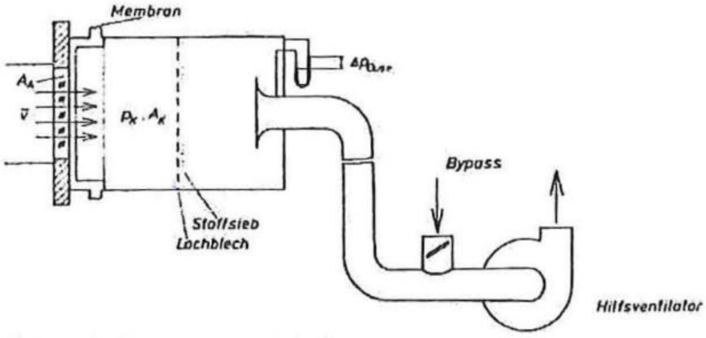


Figure 1: Original schematic for the zero-pressure method (Bitter & Detzer, 1974).

Figure 1 shows a schematic of the setup to apply the zero-pressure method at an outlet. The outlet is fully covered by a big box which is connected to a duct and a fan. When introducing the device, the pressure in the box will rise due to its induced resistance and consequently the flow will decrease. Powering the fan decreases the upstream pressure until the pressure measured in the box is compensated for. The supply flow from the ATD is now supposed to be as if no flowhood is present and can be measured accurately. The same principle applies for return flow.
Next the influence on a ventilation system of a non-powered flow hood will be discussed

2.1 System dependent impact of an intrusive device

Figure 2 shows a schematic of a hypothetical ventilation system, where air is sucked into the system by a fan on the left hand side. The air will flow through the main branch and exits over branch 1 and branch 2.
The fan characteristic can be given in a fan performance curve, showing the pressure difference dp_T - the flow quantity Q_T relation assuming a constant fan speed.

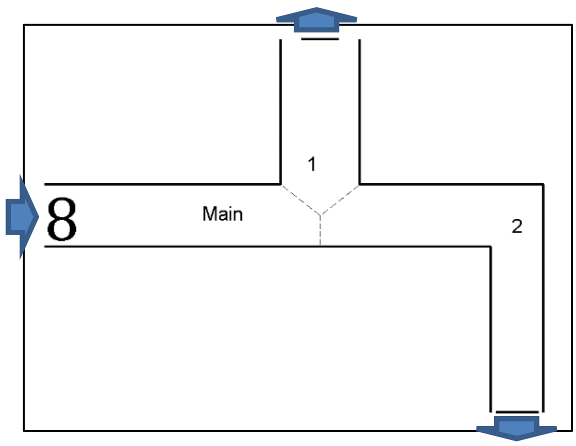


Figure 2: System Layout

The resistance curve of the independent branches can be estimated by equation (1)where Q is the air flow quantity in the branch and R the resistance factor. This resistance factor relates to the flow parameters and duct geometry (Rolloos, 1975), but will be assumed only to depend on the geometry in this example.

$$dp = R \cdot Q^2 \tag{1}$$

Figure 4 and Table 1, Situation 1, show the equilibrium state of the undisturbed system providing 250 m^3/hr at a total pressure difference of 78.2 Pa.

Table 1: Equilibrium of the undisturbed ventilation system (1).
Equilibrium with an intrusive device at branche 1 (2).

Parameter	**Flow [m^3/hr]**		**Pressure [Pa]**	
Situation	1	2	1	2
Main Branch	250	228	31.3	25.9
Branch 1	153	120	46.9	57.8
Branch 2	97	108	46.9	57.8
Total System	250	228	78.2	83.7

With a device at the outlet of branch 1, an additional resistance is induced reducing both the flow in branch 1 and the total flow (Table 1, Situation 2).

The flow reduction depends on the resistance of the intrusive device relative to the resistance of the measured branch and the total system. In particular in high performance buildings, where high efficiency, low noise and therefore low resistance are key, the flow reduction due to the use of flowhood can be considerable. Due to the large variety of ATD's and ventilation systems it is not possible to account for the resistance of an intrusive instrument by a single calibration factor (Roper, 2013). The zero pressure method seems to be the best alternative.

Some design challenges for a PFH to be dealt with will be discussed in the following section.

2.2 Design challenges for an easy-to-use PFH

The first measuring systems applying zero pressure compensation were quite big.

Figure 3: two zero pressure measuring systems, one from 1978 and one as it is being built today (top left)

These days a PFH should be easy to use, lightweight and compact. These are conflicting demands for proper zero pressure compensation.
Air supply flow will expand fast after exiting the ventilation system, causing a quick equalization of the surrounding static pressure to environmental pressure. When limiting the space at the exit with a flowhood, the flow is unable to expand freely and a non-homogeneous static pressure field is formed. This pressure field strongly depends on the flow pattern and

magnitude. Even in a PFH with a big hood (Presser, 1978) the pressure to be compensated to zero pressure depended on the measuring location in the hood.

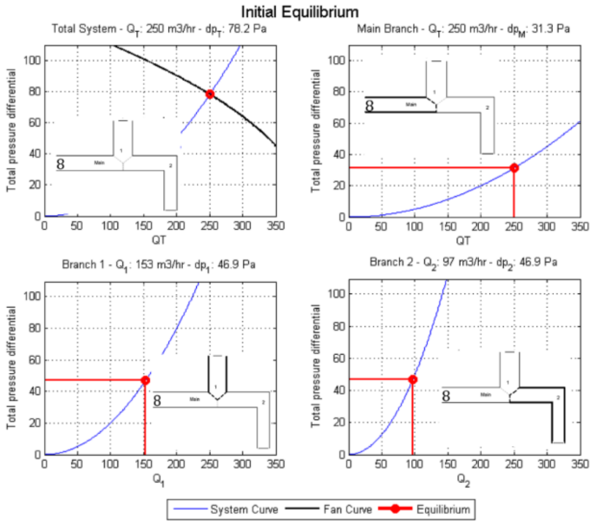


Figure 4: System equilibrium, undisturbed. From top left clockwise: total system curve and fan curve; main branch system curve; Branch 2 system curve which is steep indicating a high resistance duct and/or valve and branch 1 system curve which is less steep, therefore indicating a lower resistance and receiving a larger part of the total flow. The pressure differences over branch 1 and branch 2 are equal being the difference between the pressure in the main branch at the split and the environmental pressure.

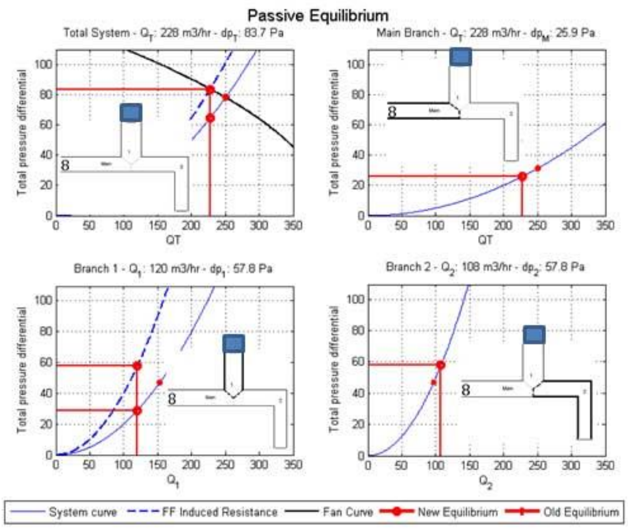


Figure 5: System equilibrium with intrusive device. The additional resistance induced by the intrusive device is visible by the steeper total system curve, resulting in a new equilibrium with the fan curve with a decreased flow and an increased pressure difference. The main branch system curve does not change, therefore the equilibrium shifts over this line to a lower air flow. A similar process applies to branch 2 where the equilibrium shifts over the system curve to a higher value due to an increased pressure difference at the splitting point. Finally, the additional resistance as induced by the device resulting in a steeper resultant system curve of branch 1 plus the device, thereby lowering the flow and increasing the pressure drop

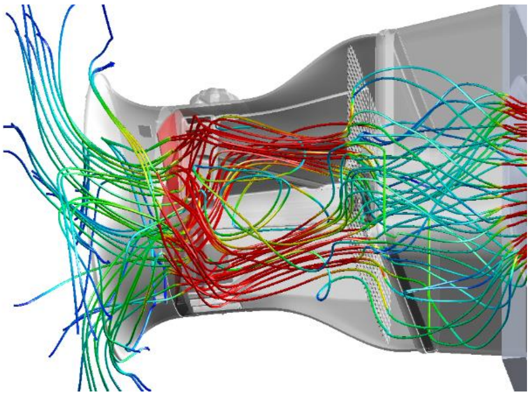


Figure 6: The non-homogeneous pressure field in a small and easy-to-handle PFH. Which pressure should be compensated for?

3 RETURN- AND SUPPLY FLOW COMPARED

Until now, supply and return flow have been treated the same way and no clear distinction is made. To show that these two flows are fundamentally different, the 1-dimensional pressure gradient will be investigated using a simple model.

3.1 Pressure Diagrams

The set-up for the model consists of a single duct with a fan in the center and a valve at both ends of the ducts. The fan induces a flow from left to right and measurements take place on the right hand side of the duct for supply flow (Figure 8) and on the left hand side for return flow (Figure 10).
Figure 9 and Figure 11 show the influence of a non-powered flow hood on the static pressure in the system.
When looking at supply flow (Figure 9), the theoretical pressure measured in front of the instrument is equal to the pressure to be compensated to restore the original condition. With a small PFH this pressure is hard to determine due to the non-homogeneous pressure field in the PFH. An additional practical problem in a small PFH is the large fluctuations of the pressure at higher flows.
In case of return flow, the non-homogeneity of the pressure field is less important since the flow is disturbed by the ATD downstream of the PFH. However, the pressure measured differs greatly from the pressure to be compensated as shown in Figure 11. This difference is equal to the dynamic pressure. The dynamic pressure can be estimated using the conditions measured with a 'passive' PFH.
When measuring return flow, theoretically, 'zero-pressure' means an equalization of the sum of static and dynamic pressure (the total pressure) with respect to the atmospheric pressure. This is fundamentally different from supply flow.

3.2 Supply flow challenges

Figure 7 shows a non-homogeneous pressure field resulting from a typical ATD. In order to compensate the resistance of the PFH, the question to be asked is: at which point should zero-pressure be applied? Since an instrument should be accurate independent of the ATD or flow measured, determining the perfect location for the pressure measurement proved to be impossible since it will differ for every ATD and air flow quantity. Therefore an 'extended' zero pressure method is developed that uses the pressure in the 'passive' PFH in relation to the passive flow amount for a wide variety of ATD’s at a large range of air flow quantities.

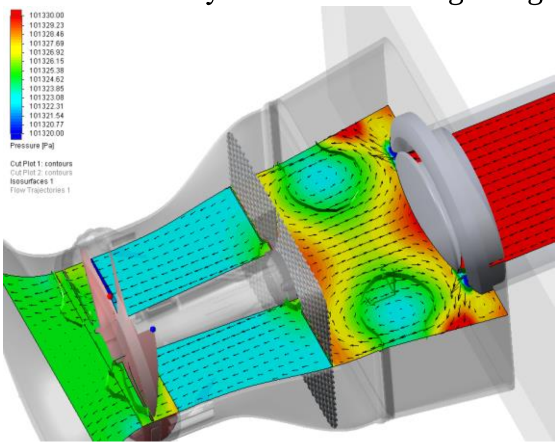


Figure 7: Valve dependent flow and pressure patterns at the pressure measurement area.

4 CONCLUSIONS

To summarize the main findings, the following conclusions can be drawn.

- The effect of a flow hood on low resistance ventilation systems is large and must be compensated for.
- The use of a compact PFH induces several challenges:
  - Supply and return flow are fundamentally different due to non-zero dynamic pressure.
  - Return flow measurement is independent of the ATD and system measured, though an invariable error is made being equal to the dynamic pressure of the flow.
  - Supply flow measurement strongly depends on the ATD measured introducing pressure variations around the point of measurement.
- The ‘Zero-pressure paradox’: in theory it seems straightforward to compensate the pressure caused by an intrusive device, in practice it is not.
- With the ‘extended’ zero pressure method the pressure to be compensated can be more accurately determined both for supply and return flows.

5 REFERENCES

Bitter, H., & Detzer, R. (1974). Lufhstrommessung am Auslaß mit der Nullmethode. Klima + Kälteingenieur(10), 425-428.

Caillou, S. (20. March 2014). Measurement of airflow rates at air terminal devices: an overview (PowerPoint Presentation). Indoor Air Quality and Ventilation Laboratory - Belgian Building Research Institute (BBRI).

Knights, C., & Gilbert, A. (2015). Domestic ventilation Systems - A BSRIA Guide. BSRIA Limited. Von www.bsria.co.uk/goto/57015-2 abgerufen

Presser, K. H. (1978). Volumenstrommessung an Luftdurchlässen nach dem Kompensations-Verfahren. HLH, 29(Nr. 2), 59-68.

Rolloos, M. (1975). Het Inregelen van Luchthoeveelheden bij Ventilatiesystemen. Verwarming en Ventilatie, no. 12, 658-670.

Roper, M. (2013). Flow Measurment for Domestic Ventilation Fans. BSRIA Limited.

Stratton, J. C., Walker, I. S., & Wray, C. P. (2012). Measuring Residential Ventilation System Airflows: Part 2 - Field Evaluation of Airflow Meter Devices and System Flow Verification. Lawrence Berkely National Laboratory, Environmental Energy Technologies Devision. California Energy Commision.

Wray, C., Walker, I., & Sherman, M. (2002). Accuracy of Flow Hoods in Residential Applications. Lawrence Berkely National Laboratory, Environmental Energy Technology Division Indoor Environment Department, Berkely, CA 94720.

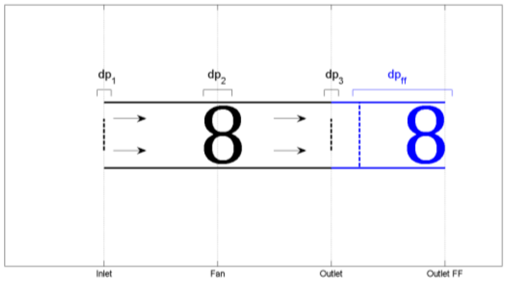


Figure 8: Layout supply flow

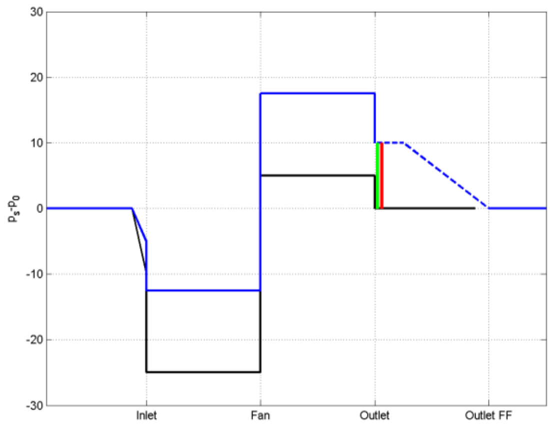


Figure 9: Supply flow pressure diagram. The black line represents the static pressure in the system without a PFH (FF); the blue solid line shows the static pressure including a PFH and the blue striped line the static pressure inside the FF. The red bar represents the pressure measured and the green bar the pressure to be compensated

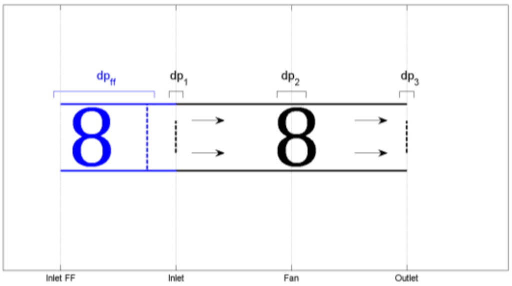


Figure 10: Layout return flow

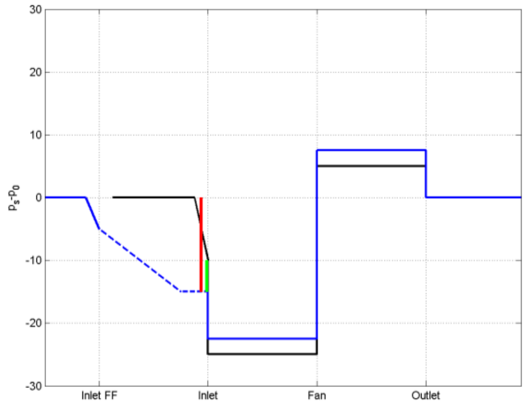


Figure 11**:** Return flow pressure diagram. The black line represents the static pressure in the system without a PFH (FF); the blue solid line shows the static pressure including a PFH and the blue striped line the static pressure inside the FF. The red bar represents the pressure measured and the green bar the pressure to be compensated.

OPTIMIZATION OF THE AIRTIGHTNESS AND THE FLOW RATE OF AIR IN NEARLY ZERO ENERGY BUILDINGS

Prof. ir. Patrick Ampe[1], ing. Anthony Tetaert[*2], ing. Leo Van Cauter[3], ir. Hilde Witters[4]

1 Ghent University
Valentin Vaerwyckweg 1
9000 Ghent, Belgium
Presenting author

2 Ghent University
Valentin Vaerwyckweg 1
9000 Ghent, Belgium
**Corresponding author: Anthony.Tetaert@UGent.be*

3 Ghent University
Valentin Vaerwyckweg 1
9000 Ghent, Belgium

4 Ghent University
Valentin Vaerwyckweg 1
9000 Ghent

Note: the contact addresses may be re-arranged

ABSTRACT

The control of heat losses, inwards/out, in nearly zero energy buildings is of high importance. The transmission losses through the building envelope are easily reduced using larger amounts of insulation. Calculation of the impact of this action on the total energy demand of the building, is quite standard. It's however much more difficult to determine the efficiency of actions to increase the airtightness of the building and the influence of the ventilation system. A valid model for these calculations is important, as the calculation of intelligent ventilation systems is much more complex due to constant air flow rate changes, which depend on the occupancy. The flow rates fall back to the hygienic minimum when no occupancy is detected, thereby reducing heat losses and lowering fan consumption. The investment cost for the ventilation system management however, increases. This paper focusses on the optimization of these two objectives using a case study.

As first result a cost optimal solution for the project is achieved, lowering the energy consumption and reducing the investment cost of the building, due to the adjustment of the ventilation parameters and the building parameters. As second result the cost optimization, combined with dynamic simulation, learns that no extreme airtightness is needed to achieve a nearly zero energy building.

KEYWORDS

Optimization, ventilation, airtightness, monitoring, dynamic simulation

1 INTRODUCTION

Since 2006 there's a lot to do about diminution of the emission of greenhouse gasses. The Energy Performance of Building Directive (EPBD), formalized its directive 2002/91/EC in

2003. In this directive the European commission imposes all member states to evaluate the energy performance of buildings in order to encourage the reduction of the energy consumption for heating, cooling, domestic hot water, lighting, ventilation and auxiliaries. Moreover, the directive specifies that the calculation should happen on basis of a methodology which may be differentiated at regional level. It is suggested that cost-optimal measures have to be taken to achieve levels of high energy performance in order to lower the energy consumption of buildings at minimal cost. Since the financial crisis of 2007-2008, not only the European Union is concerned about this topic, but countries all around the world make efforts to optimize their buildings for energy use, construction cost and occupant comfort. Bambrook (Bambrook et al., 2011) simulated a simple model for a detached house in Sydney with IDA-ICE and TRNSYS in order to determine construction solutions in which there is no energy need for heating and cooling. A multi-objective optimization has been done to minimize the energy consumption and the net present cost, which are conflicting objectives. By doing so, the heating and cooling demand in the studied house was reduced with 94% compared to the local BASIX requirements. However in Finland, Alanne (Alanne et al., 2007) was also confronted with the optimisation problem. A study has been conducted to select the appropriate energy supply system for residential buildings, using value tree analysis for multi-criteria decision problems. A common conclusion for both studies was that there are a lot of uncertainties that should be taken into account and that several contradictory objectives have to be evaluated simultaneously. Also Diakaki (Diakaki et al., 2008) in Greece and Magnier (Magnier et al., 2010) in Canada did research on multi-objective optimizations to improve the energy efficiency of buildings at the lowest possible cost with the help of genetic algorithms and dynamic building simulations. It seems however that the quest for a singular solution in multi-objective optimization problems is difficult to determine, since every model that has been made is simplified and a lot of iterative calculations have to be executed.

A lot of research has already been carried out by authors all over the world about the optimization of insulation measures in walls, floors and roofs. Daouas (Daouas et al., 2010) investigated the optimal thickness of the insulation for building walls in the Tunisian climate. Complex Finite Fourier Transform and life cycle cost analysis were joint together to compute the most profitable thickness of the insulation layer for which the sum of the energy cost due to air-conditioning and the insulation is minimised. The Complex Finite Fourier Transform is used by Ozel (Ozel et al., 2007) as well and several formulas were deduced to derive the optimum insulation thicknesses for walls with different layer structures, glazing areas, orientations and in different outdoor climate conditions (Özkan et al., 2011, Yu, 2009). Researchers in Spain (Perlova et al., 2015, Ruiz et al., 2014) also conducted several studies to determine the optimal orientation of a building and to optimize the concept design of zero energy buildings in general. In Finland as well viability studies and optimizations are carried out (Saari et al., 2012) in order to pinpoint the energy-efficiency of measures. At last in China (Huang et al., 2012), Genetic Algorithms and Pareto-analysis are used to optimize the indoor conditions of buildings. It's clear form this list that building optimization has recently become a frequently discussed topic.

It's however difficult to evaluate the optimality of airtightness measures in an optimization procedure, because the contribution of these measures is difficult to cypher. A lot of uncertainties need to be taken into account. In Belgium (Langmans et al., 2010, Van Den Bossche et al., 2012) and Norway (Relander et al., 2012) a lot of research has been done to conclude which interfaces and joints in the building envelope are most critical when it comes to airtightness. Databases were set up so building models can be expanded and airtightness can be evaluated as well. Mostly though, the authors conclude that the developed databases cannot be used without caution, because the sum of all leakages of all joints almost never corresponds with the measured value for v_{50} during pressurisation tests. On top of that, the way pressurisation tests are validated is prone to errors, uncertainties and criticism (Okuyama et al., 2012, Fernández-Agüera et al., 2011) and alternatives are tested (Hassan, 2013). It's of high interest to evaluate the airtightness on a correct way, so the taken measures can be evaluated properly and the simulation and evaluation models can be improved (Jokisalo et al., 2009, Nabinger et al., 2011, Montoya et al., 2010). In a lot of works, repetitive pressurization tests are performed to know the effect of taken measures. In this paper, the repetitive pressurization tests are conducted to know which level of airtightness has already been achieved and when to stop doing further interventions.

In buildings with a high energy performance, the losses due to leakages of air have to be restricted to a minimum. To support this strategy one of the most common solutions is to reduce the in- and outgoing airflow to the hygienically necessary ventilation rate. This implicates a control of the airflows by installation of a system for mechanical supply or exhaust or both. Furthermore, if the system is controlled by occupancy registration (CO_2, presence, humidity…), the losses of heat through air transport can be diminished further. Meanwhile it's important to ensure a good and healthy indoor climate (Laverge et al., 2011, Laverge et al., 2013, Cho et al., 2015, Turner et al., 2013, Sherman et al., 2011). The flow rates and the choice for the ventilation system need to be optimized in order to increase the energy performance of the building with a minor investment (Santos et al., 2012, Chineret al., 2014, Laverge et al., 2013).

This paper aims at explaining an optimisation method to balance the investment costs and energy savings for demand-driven ventilation systems and measures to improve the airtightness of buildings. First the general method for optimization of building parameters is explained and the way to decide which energy saving measure is the most cost-optimal is illustrated with an example. After exemplifying the general procedure, the derivative procedures for ventilation systems and air infiltration are clarified and an optimization between several ventilation systems can be achieved. Beside a cost optimal level for the infiltration flow rate v_{50} is determined. In the final chapter of this paper, a dynamic simulation with TRNSYS is carried out to demonstrate the importance of hour to hour simulation for demand-driven ventilation systems.

2 METHODOLOGY

When energy consumption in buildings needs to be reduced, it's common practice to rely on existing concepts, like Passive House, Trias Energetica, Geothermal Home… All these existing concepts have in common that they suggest to overdo some measures that can be taken to achieve better energy performances. Passive House for example is well known for the big amounts of insulation in walls, floors and roofs and for they high levels of airtightness of the building envelope in general. Triple glazed windows are often used and a ventilation system for mechanical supply and exhaust is in almost all cases indispensable to reach the Passive House requirements. Due to these steps to decrease the energy demand for heating and cooling, the generating systems can be rather limited in power. In the Geothermal Home concept on the other hand, the share of generating techniques for heating and cooling is much larger than the share of the passive measures discussed above. The author questions whether blind application of these building concepts answers the questions of cost optimality in every way and in every situation. Each building is different after all and each house, each office building, each utility building should be considered a whole, with respect for all measures that can be taken to intervene in the energy performance, passive or active.

In order to achieve optimal levels of insulation thickness, airtightness, glass-window profile combinations, ventilation systems and of course the generating systems, the objectives for the optimization need to be determined. Since it is important to lower the energy consumption and the investment cost of the building at the same time, these two objectives will be important during the optimization process. It is however difficult to designate the combination of building parameters which corresponds with the most energy efficient solution and the least expensive combination of parameters at the same time. Reducing the energy use and lowering the investment cost are opposing objectives and lowering one of them almost inevitably brings along an increase in the other. To solve this problem, the author appeals to common sense: *'If it is more expensive to invest in measures to lower the yearly energy consumption by 1 kilowatt-hour than to invest in the necessary generating techniques to yearly produce that 1 kilowatt-hour in a sustainable way, then the investor should choose to invest in the generating techniques. The opposite story is true as well.'* It is momentous though to complement the statement by explaining why the sustainable way of energy production is of that great importance. For this kind of optimization procedure, the investigators aspire the zero energy level. When attaining this level of energy efficiency, theoretically the resident will not have to pay any energy bill at the end of a year. Overproduction of electricity during summertime by photovoltaic systems will balance the higher energy needs during wintertime. Knowing this, no yearly energy consumption has to be taken into account and no yearly energy cost (predicted for the next 30 years) needs to be derived.

This results in the practicing ratio $\Delta IC_T/\Delta E$, which has to be computed for all possible measures. In the ratio ΔIC_T is the additional cost for the taken measure and ΔE is the additional energy saving, caused by the taken measure. The ratio $\Delta IC_T/\Delta E$ has to be as low as possible, since that means a large reduction of the energy consumption can be realised at a relatively low investment cost.

Table 1: Determining the most cost optimal solution from a list of energ saving measures for buildings.

Energy saving measure	Additional energy saving after measure [kWh]	Additional cost of measure [€]	Ratio $\Delta IC_T/\Delta E$ [€/kWh]
Increasing the airtightness of a building from v_{50} = 6 m³/h.m² up to v_{50} = 2.5 m³/h.m²	1,321.86	469.10	0.355
Insulation of the cavity in a wall with PUR-insulation	2,563.15	18,401.20	7.179
Changing glazing from U= 2.3 W/m².K up to 1.1 W/m².K	2,277.50	4,450.89	1.954
Installation of a photovoltaic solar system with a peak power of 6,120 Wp	4,862.51	9,200.00	1.892
Installing a demand-driven ventilation system for natural supply and mechanical exhaust	3,476.42	3,500.00	1.007
Placing a heat pump and floor heating instead of an electrical floor heating	17,409.30	18,258.20	1.049

Table 1 shows an exemplary list of measures that can be taken for an exemplary detached house in Belgium. The costs have been obtained by asking quotes in local companies and the energy calculations were conducted in accordance with the applicable standards. It is clear from this table that the lowest value for the ratio $\Delta IC_T/\Delta E$ is 0.355, which corresponds to the improvement of the airtightness of the building. Only €469.1 has to be spend and already 1,321.86 kWh can be saved every future year.

2.1 Investment cost calculations

As explained earlier in this paper, only the investment cost for the energy saving measures has to be computed. Since all energy production is considered to be sustainable, no yearly energy bill is expected, provided that the investment costs for renewable generation is included with the cost of the generation system. To make a proper estimation of the investment cost for ventilation systems and airtightness measures, the research team relied on price offers from several companies. In case of ventilation systems, the calculation is easily done, since the fan cost, window lattices, piping, correct adjustment and labour cost are the only parameters to be set. It's fair enough to take all related costs from the price offer from the installation company and to sum them. A total and correct investment cost for every single ventilation system can be found that way. Table 2 shows a list of ventilation systems with the investment cost and the necessary system parameters. Note that the investment costs are VAT-exclusive and only applicable for the project the list of systems was made for.

Table 2: List of ventilation systems with their respective investment costs

System Name	Investment cost	m-factor for placement quality	Reduction factor for demand control	Heat recovery efficiency
System C	€1,500.0	1.50	1.00	n/a
Dantherm HCV5	€5,696.0	1.30	0.60	0.90
Systemair SAVE VSR 300	€4,592.0	1.25	0.54	0.90
Systeem C+ EVO II / Healthbox II	€2,394.8	1.22	0.72	n/a
Systeem C+ EVO II / Healthbox Smartzone	€2,644.8	1.22	0.45	n/a
Systeem C+ / Xtravent EcoModus Compact	€1,941.7	1.22	0.94	n/a

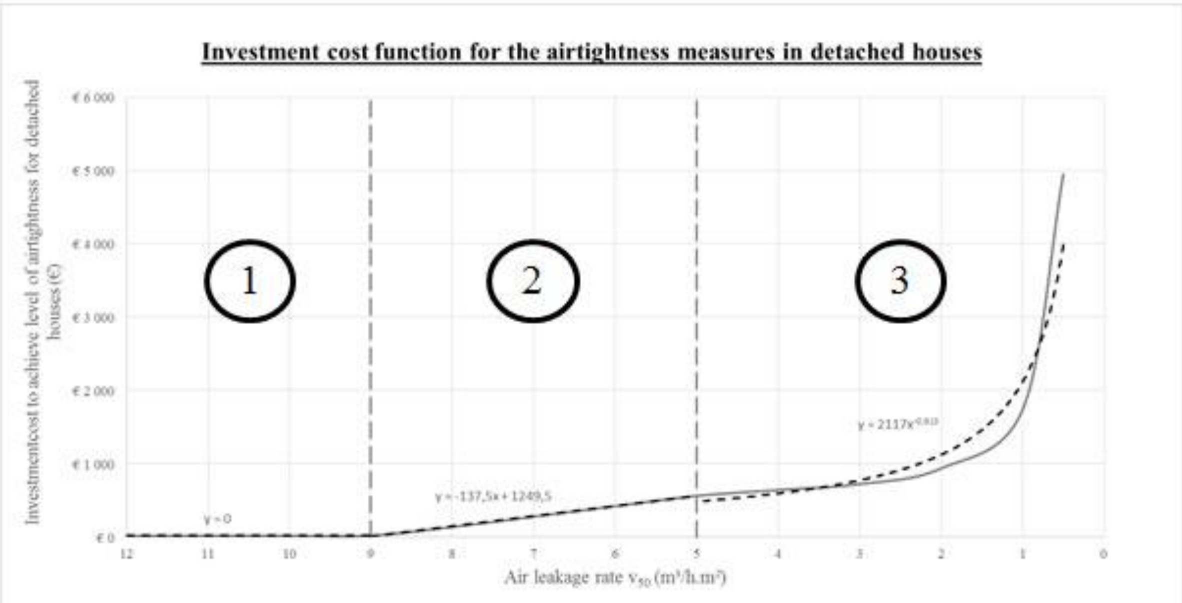


Figure 7: Investment cost function for the airtightness measures in detached houses

As far as the airtightness, the price setting is far more complicated. In the first place, it's difficult to allocate the appropriate labour time (and cost) to the airtightness improving measures since it's sometimes enough to just pay attention while construction and since the increase of the airtightness can be a secondary effect of other energy saving measures. Besides that, it's almost impossible to predict the effect of the measures on the level of airtightness. As frequently stated in the literature, one can rely on component leakage models to dimension the impact of a single measure on a single building component and sum the impact for all measures. Literature however also states there is an uncertain part in the airtightness of buildings which cannot be predicted using the component leakage method and the component leakage models often differ as a result of anomalies in the installation methods.

To be able to make a proper estimation of the effect and the investment cost of the measures to improve the airtightness, the research team contacted different companies in order to obtain the necessary information. It was possible to determine some rules of thumb with regard to the price function of airtightness. Figure 1 illustrates that three cost zones can be distinguished. Zone 1 is the zone where no investment cost for the airtightness improvement have to be accounted. No special efforts have to be done to achieve a level of airtightness corresponding with an air leakage rate v_{50} equal to approximately 9 m³/h.m². From that point on, investments need to be made to increase the airtightness of the building. In zone 2, between the air leakage rates v_{50} equals 9 m³/h.m² and v_{50} equals 5 m³/h.m², the graph shows a linear relation with the investment cost. Better values for v_{50} can be reached, but enhancing the level of airtightness over v_{50} equal to 5 m³/h.m², will involve an investment following a power function. It is shown in figure 1 that every additional investment in the increase of the airtightness, becomes more costly and thus less profitable. Knowing what has been explained earlier in this paper, the ratio $\Delta IC_T/\Delta E$ for an improvement from v_{50} = 9 m³/h.m² up to v_{50} = 8 m³/h.m² is lower than the ratio $\Delta IC_T/\Delta E$ for an improvement from v_{50} = 3 m³/h.m² up to v_{50} = 2 m³/h.m², and in this way more profitable.

2.2 Energy saving calculations

The procedure to calculate the energy savings achieved with the implementation of one of the suggested measures is in accordance with the Belgian requirements and norms. The investigators rely on the calculation methods suggested by the Belgian government in their energy regulations. The Energy Decree of November 19th, 2010 describes a structure of formulas to be used in the normalised calculation method for the energy performance of residential buildings. The optimization however only concerns ventilation and infiltration. To calculate the energy losses due to exhaust of comfortable warm air from inside the building to the outside environment, following equations need to be used.

$$Q_{V,heat,seci,m} = H_{V,heat,seci} \times (18-\theta_{e,m}) \times t_m \qquad (1)$$

In equation (1) $Q_{V,heat,seci,m}$ [MJ] stands for the monthly heat loss due to ventilation in a building expressed in MJ, $H_{V,heat,seci}$ [W/K] is the heat transfer coefficient through ventilation in a building calculated with equation (3), $\theta_{e,m}$ [°C] is the monthly average outdoor temperature for Belgium and t_m [Ms] stands for the length of the studied month, both shown in table (3). The indoor temperature of the building is assumed 18°C by which means the temperature differences between rooms and between day time and night time is taken into account. It's a fair average for the overall indoor temperature of the building during an entire day. To transform the value for $Q_{V,heat,seci,m}$ in MJ to a usable value for the optimization in kWh, equation (2) is used.

$$E_{V,heat,seci,m} = Q_{V,heat,seci,m} / 3.6 \qquad (2)$$

with $E_{V,heat,seci,m}$ the monthly heat loss due to ventilation in a building, expressed in kWh. In order to derive the value of the heat transfer coefficient $H_{V,heat,seci}$, equation (3) is used. The formula shows that both hygienic ventilation and in/exfiltration through the building envelope are included in the calculation.

$$H_{V,heat,seci} = H_{V,inf/exfilt,heat,seci} + H_{V,hyg,heat,seci} \tag{3}$$

In equation (3), $H_{V,inf/exfilt,heat,seci}$ is the monthly average due to in/exfiltration for heating calculations and $H_{V,hyg,heat,seci}$ the monthly average due to hygienic ventilation for heating calculations. Both variables in equation (3) are calculated below.

Monthly average characteristic	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Outdoor temperature [°C]	3.2	3.9	5.9	9.2	13.3	16.2	17.6	17.6	15.2	11.2	6.3	3.5
Length of month [Ms]	2.6784	2.4192	2.6784	2.5920	2.6784	2.5920	2.6784	2.6784	2.5920	2.6784	2.5920	2.6784

2.2.1 *Heat transfer coefficient for in/exfiltration*

The heat transfer coefficient for in/exfiltration of air through the building envelope is the coefficient that represents the airtightness of the building. The coefficient is calculated using equation (4).

$$H_{V,inf/exfilt,heat,seci} = 0.34 \text{ x } \tilde{V}_{inf/exfilt,heat,seci} \tag{4}$$

With

$$\tilde{V}_{inf/exfilt,heat,seci} = 0.04 \text{ x } \tilde{v}_{50,heat} \text{ x } A_{T,E,seci} \tag{5}$$

In equation (5), $\tilde{v}_{50,heat}$ symbolises the air leakage rate per unit of area, with a pressure difference of 50 Pa and $A_{T,E,seci}$ is the total area of the building envelope components through which heat transfer due to transmission occurs.

2.2.2 *Heat transfer coefficient for ventilation*

To derive the heat transfer coefficient for hygienic ventilation, equation (6) should be applied.

$$H_{V,hyg,heat,seci} = 0.34 \text{ x } r_{preh,heat,seci} \text{ x } \tilde{V}_{hyg,heat,seci} \tag{6}$$

In formula (6), $r_{preh,heat,seci}$ is a reduction factor for the effect of preheating, which contains the efficiency of the heat recovery system. $\tilde{V}_{hyg,heat,seci}$ is the necessary hygienic ventilation air flow, required for the building. This flow can be calculated easily in accordance to the regulations, but in the calculation structure of the Belgian government, an easy formula is presented. For this paper, equation (7) is used.

$$\tilde{V}_{hyg,heat,seci} = \left[0.2 + 0.5 \text{ x } e^{\left(\frac{-V_{seci}}{500}\right)}\right] \text{ x } f_{reduc,vent,heat,seci} \text{ x } m_{heat,seci} \text{ x } V_{seci} \tag{7}$$

In this formula, V_{seci} (m^3) is the total volume of the building, $f_{reduc,vent,heat,seci}$ a reduction factor for ventilation, based on the demand-control, whether or not provided and m_{heat} a multiplication factor, depending on the ventilation system itself and on the quality of the execution. The value of the m_{heat}-factor is always between 1 and 1.5. As seen in table 2, both $f_{reduc,vent,heat,seci}$ and m_{heat} are system dependent variables which are included in the database.

3 CASESTUDY

To verify the calculation method on optimization and to determine the efficiency of the different measures that are taken, it is important to rely on a practical case. Only that way, the procedure can be validated.

3.1.1 Description of the case study

The project that has been studied is a detached house in Moorslede in Belgium. It's a two-storey home with a flat roof and a simple, rectangular floor plan, which is shown in figure 2. The house consists of an open living room and kitchen on ground level and three bedrooms and two bathrooms on the first floor level. The building is constructed out of prefabricated 'cross laminated timber'-panels. The insulation layer of the building envelope is placed around the building and the outside of the walls are finished with wooden battens.

3.1.2 Optimization procedure

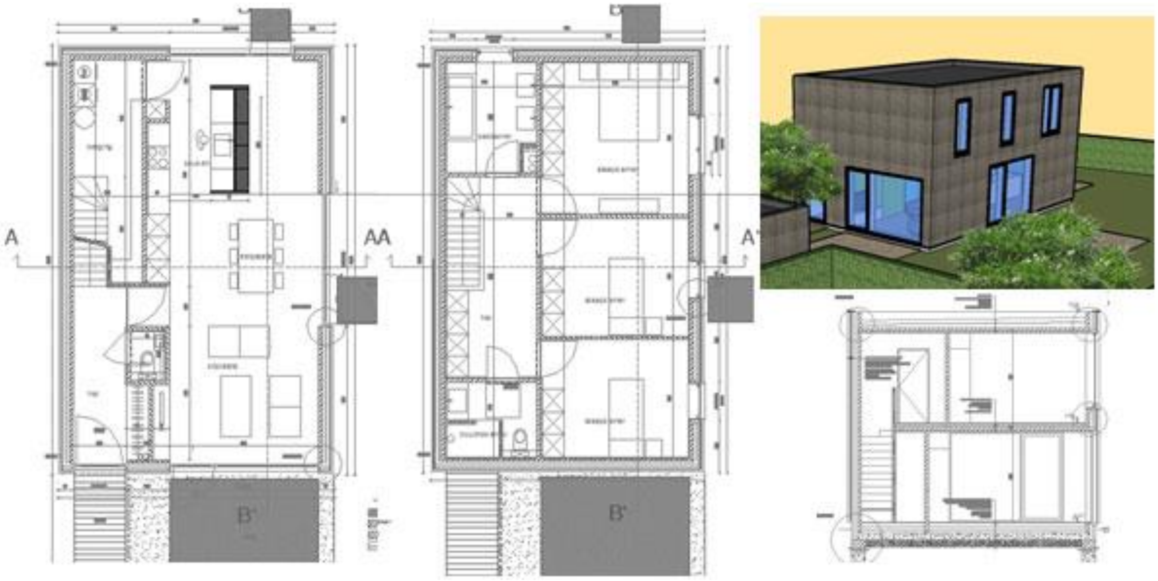


Figure 8: Case study project in Moorslede, Belgium: Floor plans, section, visualisation.

In the optimization procedure a lot of variables were left open. All insulation materials were known, but no insulation thickness was determined for the ground floor, the walls and the flat roof. An optimum U-value and g-coefficient for the windows needed to be set as well and the generation system for heating and domestic hot water was also not determined yet. As for the ventilation system, a list of possible systems was provided and for the airtightness optimization, the investment cost function, illustrated above, was used.

Iterative application of the optimization procedure was started. For each iteration a decision was made between the increase of the insulation thickness in a building component by one centimetre, the change of the window type, the improvement of the airtightness by 0,5 $m^3/h.m^2$, the amelioration of the ventilation system and the change of the generation system.

The optimization process resulted in the following optimum combination of parameters:

11 cm excelsior in the walls, 12 cm mineral wool on the roof, 8 cm expanded polystyrene on the ground floor, windows with a U-value of 1.14 W/m².K and a g-value of 0.28, combined with a heat pump. The optimum airtightness level was achieved at 3.0 m³/h.m² and from the list the ventilation system 'Systemair SAVE VSR 300' was selected. To verify the correctness of the optimization results, a scatterplot was generated consisting of 100 random combinations of variables and none of these combinations was more optimal than the suggested combination.

3.1.3 Results for in/exfiltration rate and ventilation

It's clear from above that the optimization procedure is a rather theoretical concept. The practical application of the optimization results is sometimes more complicated.

As told before, ventilation system 'Systemair SAVE VSR 300' was elected to be the most optimal ventilation system for the studied building. It's easy to order the ventilation system and to get it installed in the house. However, it's of great importance to appeal to professional installers who have the necessary skills to commission the system. Any minor deviation can make the system less optimal than calculated and thus sometimes a wrong choice. Decent craftsmanship is still one of the most important requirements during the building process.

In the case of airtightness, literature has shown that it's impossible to predict the impact of measures improving the airtightness of a building. Too much uncertainties play a role in the overall airtightness level of a building. Since the optimization procedure stated that an air leakage rate of 3.0 m³/h.m² is most optimal, it's wise to follow a roadmap to seal the leaks. After every step, a pressurization test has to be done to verify the impact of the measure that has been carried out. In this case a pressurisation/depressurisation test was conducted after every step using a Blower-door.

To achieve an higher level of airtightness, the following steps need to be completed. Once the desired airtightness level has been reached, the next steps don't have to be completed anymore. The roadmap below is applicable for massive wood-constructions like the one presented as case study. Other steps need to be taken when air tightening a masonry construction or a wood frame construction. Furthermore, all steps and all optimal combinations only apply on the studied project and need to be adjusted for other cases, even if the same type of construction is used.

Step 1: taping the floor seams
Step 2: taping the seams of the CLT-panels
Step 3: airtight installation of windows, doors and wall and roof ducts
Step 4: airtight installation of power outlets and light switches
Step 5: air tightening of chases for electricity and water
Step 6: taking all other possible measures to meet the Passive House standards.

It's fair to stop taking measures, when the desired value for v_{50} has been reached and when not reached yet, a thermographic camera can be of assistance to point out the leakages in the building that aren't sealed yet.

3.1.4 Dynamic simulations with TRNSYS

When studying the optimization of systems and building variables, it's important to keep in mind the comfort for the occupants. As explained in the methodology for this research, a calculation has been carried out based on monthly average outdoor temperatures. In table (3) it is shown that the average temperature for July for example is 17.6 °C in Belgium. It's obvious that during summertime the outdoor temperature exceeds this average for a large number of hours. Using the average outdoor temperatures is an acceptable method to determine the energy consumption for an entire year, since peaks are not important for that matter. Nevertheless, high outdoor temperatures and consequently big amounts of energy entering the building, can cause overheating for several hours. To predict the number of hours the rooms are overheated, dynamic simulation is inevitable. The temperature of all rooms in the case study building are simulated.

Dynamic simulations can also be very useful to compute the value of $f_{reduc,vent,heat,seci}$, since this value depends on the operative time of the system. After all, the system only has to work when there's a ventilation demand. In order to evaluate whether the ventilation system has to be running or not, occupation schedules, indoor temperature monitoring, humidity evaluation… can be added to the model. The dynamic simulation makes it possible to activate and deactivate the ventilation system every time step and a truthful answer can be given concerning the energy consumption. In the dynamic model, the reduction factor for demand-control is no longer an estimation, but depends on a lot of in time variable parameters.

As for the case study, results have been extracted from the dynamic simulation model about the risk of overheating and the monthly heating demand.

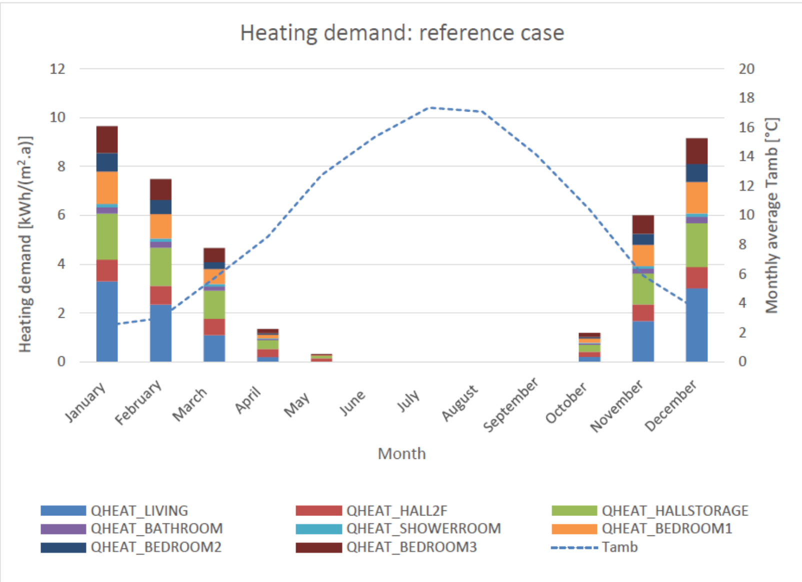


Figure 9: Monthly heating demand of the reference building per m² net floor area.

4 CONCLUSIONS

In this paper it's been clarified how to calculate the optimal airtightness level and how the select the most cost optimal ventilation system from a list. When implementing the results of the calculation, a stepwise application of air tightening measures is in order. To make sure no unnecessary steps are taken, after each step, a pressurisation test has to be performed on the building. In that way the cost optimal solution can be pursued. As for the ventilation system, the characteristics of the system are of great importance. The quality of installation and the reduction factor for demand control have to be determined correctly before the optimization procedure starts and a yearly based optimization study can be carried out.

In order to minimize the risk of overheating, the author recommends a dynamic simulation with a time step of one hour. That way, the number of hours for which the indoor temperature in a room exceeds the maximum acceptable temperature, can be counted for an entire year. Several measures to decrease the risk can be tested by including the measures in the dynamic simulation model. Meanwhile, the demand-control can be programmed in the model and a correct energy consumption can be derived, without uncertain estimation of some variables.

5 ACKNOWLEDGEMENTS

The authors would like to thank company Korlam for the cooperation and for the confidence, while providing the studied case. We also want to thank the architects group Architecten Groep III for the fruitful cooperation.

6 REFERENCES

Alanne, K., Salo, A., Saari, A., Gustafsson, S. (2007). Multi-criteria evaluation of residential energy supply systems. Energy and Buildings 39, 1218-1226

Bambrook, S.M., Sproul, A.B., Jacob, D. (2011). Design optimisation for a low energy home in Sydney. Energy and Buildings 43, 1702-1711

Chiner, E., Sancho-Chust, J. N., Landete, P., Senent, C., Gómez-Merino, E. (2014). Complementary Home Mechanical Ventilation Techniques: SEPAR Year 2014, Archivos de Bronconeumologia 50(12), 546-553

Cho, W., Song, D., Hwang, S., Yun, S. (2015). Energy-efficient ventilation with air-cleaning mode and demand control in a multi-residential building. Energy and Buildings 90, 6-14

Daouas, N., Hassen, Z., Ben Aissia, H. (2010). Analytical periodic solution for the study of thermal performance and optimum insulation thickness of building walls in Tunisia. Applied Thermal Engineering 30, 319-326

Diakaki, C., Grigoroudis, E., Kolokotsa, D. (2008). Towards a multi-objective optimization approach for improving energy efficiency in buildings, Energy and Buildings 40, 1747-1754

EN (2003). Directive 2002/91/EC of the European Parliament and of the Council of 16 December 2002 on the Energy Performance of Buildings, Official Journal of the European Communities, L1/65-L1/71

Fernández-Agüera, J., Sendra, J. J., Domínguez, S. (2011). Protocols for measuring the airtightness of multi-dwelling units in Southern Europe. Procedia Engineering 21, 98-105

Hassan, O. A. B. (2013). An alternative method for evaluating the air tightness of building components. Building and Environment 67, 82-86

Huang, H., Kato, S., Hu, R. (2012). Optimum design for indoor humidity by coupling Genetic Algoritm with transient simulation based on Contribution Ratio of Indoor Humidity and Climate analysis. Energy and Buildings 47, 208-216

Jokisalo, J., Kurnitski, J., Korpi, M., Kalamees, T., Vinha, J. (2009). Building leakage, infiltration, and energy performance analyses for Finnish detached houses, Building and Environment, 377-387

Langmans, J., Klein, R., De Paepe, M., Roels, S. (2010). Potential of wind barriers to assure airtightness of wood-frame low energy constructions, Energy and Buildings 42, 2376-2385

Laverge, J., Janssens, A. (2013). Optimization of design flow rates and component sizing for residential ventilation. Building and Environment 65, 81-89

Laverge, J., Pattyn, X., Janssens, A. (2013). Performance assessment of residential mechanical exhaust ventilation system in accordance with Belgian, British, Dutch, French and ASHRAE standards. Building and Environment 59, 177-186

Laverge, J., Van Den Bossche, N., Heijmans, N., Janssens, A. (2011). Energy saving potential and repercussions on indoor air quality of demand controlled residential ventilation strategies. Building and Environment 46, 1497-1503

Magnier, L., Haghighat, F. (2010). Multiobjective optimization of building design using TNSYS simulations, genetic algorithm, and Artificial Neural Network. Building and Environment 45, 739-746

Montoya, M. I., Pastor, E., Carrié, F. R., Guyot, G., Planas, E. (2010). Air leakage in Catalan dwellings: Developing an airtightness model and leakage airflow predictions, Building and Environment 45, 1458-1469

Nabinger, S., Persily, A. (2011). Impacts of airtightening retrofits on ventilation rates and energy consumption in a manufactured home. Energy and Buildings 43, 3059-3067

Okuyama, H., Onishi, Y. (2012). Reconsiderationg of parameter estimation and reliability evaluation methods for building airtightness measurement using fan pressurization. Building and Environment 47, 373-384

Ozel, M., Pihtili, K. (2007). Optimum location and distribution of insulation layers on building walls with various orientations. Building and Environment 42, 3051-3059

Özkan, D. B., Onan, C. (2011). Optimization of insulation thickness for different glazing areas in buildings for various climatic regions in Turkey. Applied Energy 88, 1331-1342

Perlova, E., Platonova, M., Gorshkov, A., Rakova, X. (2015). Concept Project of Zero Energy Building. Procedia Engineering 100, 1505-1514

Relander, T., Holøs, S., Thue, J. V. (2012). Airtightness estimation - A state of the art review and an en route upper limit evaluation principle to increase the chances that wood-frame houses with a vapour- and wind-barrier comply with the airtightness requirements. Energy and Buildings 54, 444-452

Ruiz, P. A., Martín, J. G., Lissén, J. M. S., Sánchez de la Flor, F. J. (2014). An integrated optimisation method for residential building design: A case study in Spain. Energy and Buildings 80, 158-168

Saari, A., Kalamees, T., Jokisalo, J., Michelsson, R., Alanne, K., Kurnitski, J. (2012). Financial viability of energy-efficiency measures in a new detached house design in Finland. Applied Energy 92, 76-83

Santos, H. R. R., Leal, V. M. S. (2012). Energy vs. ventilation rate in buildings: A comprehensive scenario-based assessment in the European context. Energy and Buildings 54, 111-121

Sherman, M. H., Walker, I. S. (2011). Meeting residential ventilation standards through dynamic control of ventilation systems. Energy and Buildings 43, 1904-1912

Turner, W. J. N., Walker, I. S. (2013. Using a ventilation controller to optimise residential passive ventilation for energy and indoor air quality. Building and Environment 70, 20-30

Van Den Bossche, N., Huyghe, W., Moens, J., Janssens, A., Depaepe, M. (2012). Airtightness of window-wall interface in cavity brick walls. Energy and Buildings 45, 32-42

Yu, J., Yang, C., Tian, L., Liao, D. (2009). A study on optimum insulation thicknesses of external walls in hot summer and cold winter zone of China. Applied Energy 86, 2520-2529

AIRTIGHT DUCT SYSTEMS
[A SIMPLE WAY OF IMPROVING A BUILDING'S ENERGY EFFICIENCY WITHOUT INCREASED INVESTMENT]

Toni Nicolas Salame[1], Rodrigo Sanz[2], Santiago Pascual[3]

1 Industrias Gonal, S.L.
Can Macia, 2
Les Franquesas-Spain

2 Industrias Gonal, S.L.
Can Macia, 2
Les Franquesas-Spain

3 Industrias Gonal, S.L.
Can Macia, 2
Les Franquesas-Spain

ABSTRACT

Against the background of increased global demands for energy efficiency, property owners should raise the standards of ductwork systems for ventilation, heating and air conditioning. This would not only save energy, but also mean lower installation costs, shorter assembly times and better air quality thanks to less leakage.

The importance of energy-efficient buildings will increase in the future, not only due to rising electricity prices, but also due to increased environmental awareness. One example of this is the Kyoto Agreement, which indirectly forces countries to review their energy use with the aim of reducing carbon dioxide emissions. A current example of the latter is the energy directive from the EU, which basically increases the demands on energy planning and on the energy performance of buildings. One way of satisfying the stricter energy rules can be to make demands of airtight ductwork systems for ventilation.

KEYWORDS

Ductwork, Ventilation, air tightness, rectangular duct, leakage,

1. INTRODUCTION

Ductwork systems for ventilation, heating and air conditioning can be divided into four air tightness classes. The most airtight class is D, and the least airtight duct system is categorised in the A class. Most modern systems rarely achieve air tightness class B. Thus, there is much potential for improvement here. In Europe, more airtight systems would mean an annual energy saving of approx.10 TWh, which is comparable to the annual production of three nuclear power plants.

2. ADVANTAGES

2.1 Lower investments costs

At present ventilation ducts are usually rectangular or circular. There are ductwork systems with integral sealing systems that can guarantee the highest air tightness class, D. A ventilation system often consists of a mixture of circular and rectangular ducts, in which the latter often find it more difficult to satisfy strict demands on air tightness. If we strive to ensure that as much of the duct system as possible consists of ductwork systems with integral sealing systems of air tightness class D, the whole system can ultimately achieve, savings to the order of 10 TWh.
We should therefore as far as possible replace standard ductwork with ductwork systems with integral sealing systems. The total cost of purchasing material and assembly is lower than a standard solution, meaning that the payback is immediate. This also means that no payback calculation is necessary.

Figure 10: Round metallic ductwork

Figure 2: Round thermoplastic ductwork

2.2 Shorter assembly time

If you have an integral ductwork sealing system, the assembly time is approx. 80% shorter, because no tapes and or sealants needed. The conclusion is therefore that the total investment for an airtight class D system is lower than for the poorer quality class A system. System planners should strive for a solution that is integral ductwork sealing system, and in some cases this may also produce new, innovative alternatives and close interaction with the architect about the installation areas.

Figure 3: Rectangular thermoplastic ductwork assembly

2.3 Smaller fans

In order for the right flow (and air conditioning effect) to reach the ventilated areas, the fans must transport the total flow, i.e. not only the air that passes to and from the various areas in the building, but also the air expelled out of or drawn into the ventilation system. A system that leaks a lot thus requires larger, more expensive fans and more space in the building.

2.4 Improved health and internal environment

Besides the purely financial benefits of more efficient ventilation based on a integral sealing ductwork system, there can also be a positive effect on people's health. This is because a higher-quality duct system creates the conditions for buildings being well ventilated with the lowest possible energy use. Much current research indicates that the air quality in our buildings has a major impact on people's health and well-being. Improved air quality reduces absence through illness and increases productivity. And if the air is supplied in an energy-efficient way, this reduces the impact on the external environment.

3. APPLICATION

3.1 Act now

Stricter demands on duct systems thus produce nothing but positive effects. We should therefore define the requirements for air tightness as soon as possible, as early as the planning stage of a property. Otherwise we can expect unnecessarily high and escalating energy costs!
System planners should strive for a solution for air tightness, and in some cases this may also produce new, innovative alternatives and close interaction with the architect about the installation areas.

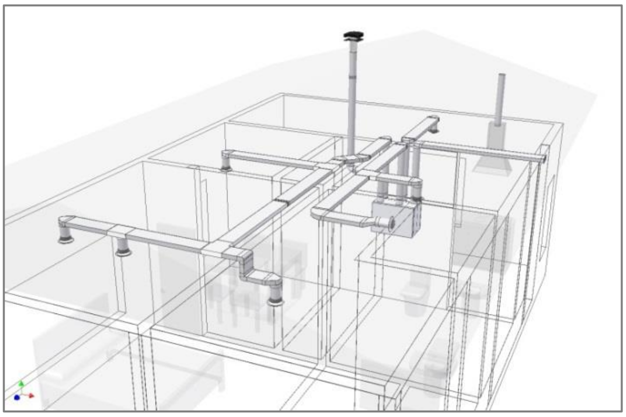


Figure 4: Rectangular thermoplastic ductwork application

3.2 Why is air tightness important?

If the ventilation system is not airtight, the leakage must be compensated with an increased fan flow. This requires an over-dimensioning of the unit's parts such as fans, filters, heater/cooling-coil batteries, heat exchangers, etc. All of this causes increased energy consumption and hence increased costs and a greater impact on the environment than necessary. It is also not entirely uncommon for the leaking air to cause a disturbing hissing sound.

3.3 Air tightness to the surroundings

Leakage is a function of the system's size and of the pressure difference. A large system leaks more than a small one. Higher pressure also entails greater leakage. Leakage is specified in the unit l/s. The leak factor incorporates the system's size by specifying the leakage per unit of area. This is calculated as the system's total leakage

divided by its total casing area, and is written (l/s)/m². One complication is that it is not the system's area that leaks, rather its joints and connections etc., which is why systems with unusual ratios between components and ducts can produce peculiar results. The pressure difference is retained as a separate parameter. The diagrams that are used

3.4 Different types of air tightness

Air tightness is divided into the classes A, B, C and D, where class D is the most airtight. It is three times as airtight as class C, which in turn is three times as airtight as class B, and so on. Classes A–C were first defined in the publication EUROVENT 2/2.

The classification is intended for entire ventilation systems. In other words, it is not meant to be applied to individual products.

Testing air tightness normally takes place at a 400 Pa pressure difference. This is not a requirement, however, so testing may take place at any suitable pressure difference(s).

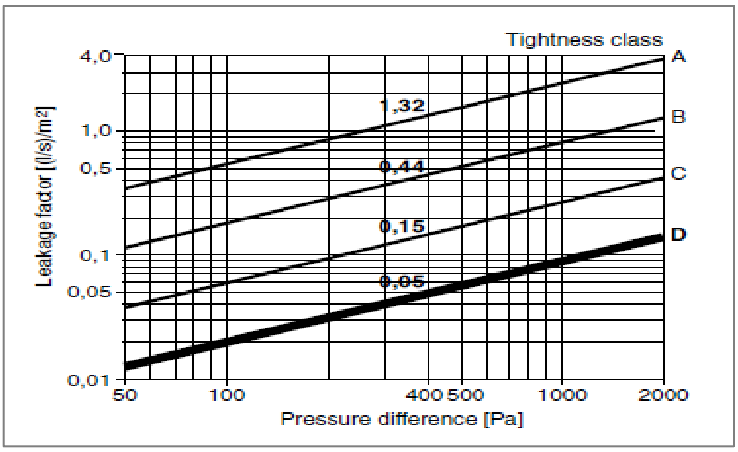


Figure 6: Tightness class according to EN 12237

3.5 Reduced comfort

If the leakage is not compensated with an increased flow, the consequence will be reduced comfort combined with poorer air quality and the wrong temperature. Furthermore, the planned air flows will not be achieved.

3.6 What can go wrong?

The most common causes of leaks are incorrect assembly, reduced product quality and products that have been reused.

4 PRODUCT

4.1 General

Inaccessible points are often leakage points. This is due either to the fact it has not been possible to access these points to screw in properly, or that the product must be specially built at the installation location in order to be assembled.

Components that have been reused or moved/turned often have old, non-airtight holes created by screws or adhesive tapes. These holes cannot be seen from the outside of the system, but require an internal inspection in order to be discovered.

Components that are “produced” or adjusted/converted at the actual installation location often leak.

Quite naturally, components that are used outside of their intended function have difficulty remaining airtight. For example, where 60° bends are used where there is a 45° change of direction.

You should stipulate that the manufacturer of the products you use reports which air tightness class they satisfy – this means less worry for you.

4.2 Installation

Air tightness class D places high – very high – demands on the installation and on the fitters. Follow our installation instructions.

Screw holes must be sealed if components and ducts have been reused or moved. Otherwise the result will be leaking screw holes.

Installation using blind rivets that are not airtight must be avoided. The rivets’ pull-stems can otherwise fall out of the rivets, resulting in leaking holes. This happens particularly with low-price blind rivets.

Self-tapping screws are occasionally tightened so hard that they lose grip and simply spin around. They are then not so airtight.

Knives are often used as a guide when aligning the components in the ducts. There is then a risk of causing indentations on the ends of the components, resulting in leakage.

5 CONCLUSION

It is important to minimise the leakage in the system. Leakage has different consequences, and a majority of them in the end lead to increased costs:

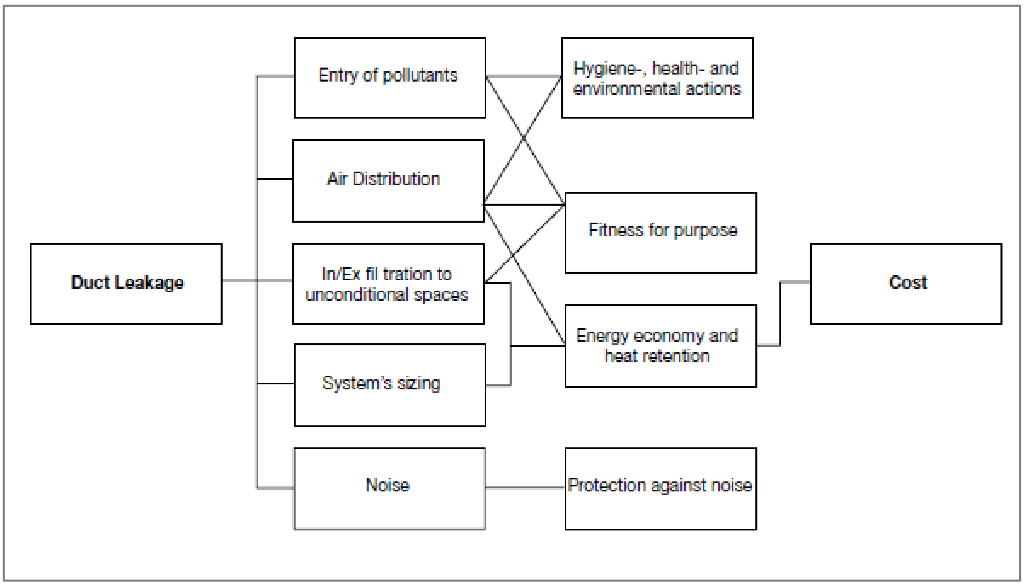


Figure 7: Leakage consequences

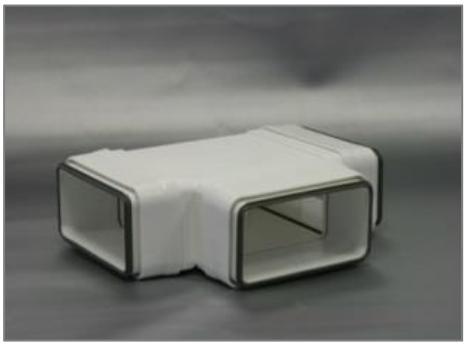


Figure 8: Rectangular thermoplastic ductwork

Figure 9: Rectangular thermoplastic ductwork

Checklist for an Adequate Ventilation System

To construct a proper ventilation system you should be able to tick all the boxes in the checklist below:

Good indoor climate

- Lack of draft
- Low noise level
- Appropriate temperature

- Good air quality

Low energy use

Simple adjustments

Low life cycle cost (total cost)

- Material cost
- Installation cost
- Heating/cooling cost
- Fan electricity cost

Easy to operate and maintain and equipped with detailed instructions.

6 REFERENCES

Carrié F. R. et al, pp 235–243, ELSEVIER Energy saving potential in Europe, Duct leakage in European buildings: status and perspectives, Energy and Buildings 32 (2000),

Sick Leave – IAQ Risk of Sick Leave Associated with Outdoor Air Supply Rate, Humidification, and Occupant Complaints. Indoor Air 2000:10. ISSN 0905-6947

Wargocki P., Wyon D.P, Sundell J.,Clausein G. and Fanger O. The effects of outdoor air supply rate in an office on perceived air quality, Sick
Building Syndrome (SBS) Symptoms and Productivity, Indoor Air, 10, pp 222-236.

Malmstrom, T.G., A Review of International Literature Related to Ductwork for Ventilation Systems, Technical Note AIVC 56, INTERNATIONAL ENERGY
AGENCY, 2002

Tsal et al 1998, T-method duct design: Part IV - Duct
leakage theory, ASHRAE Transaction.
UK SAP (UK Standard Assessment Procedure for rigid duct), 14/February/2011
BS EN 13141-6:2004, to define the duct configurations
Directive 2002/91/EC of the European Parliament and of the Council, Dec 16th 2002

CEN standard CE-XX-YY

ESTIMATING THE AVERAGE AIR CHANGE RATE FOR THE HEATING SEASON

Dr. László Fülöp[*1], György Polics[2]

[1] *Faculty of Engineering and Information Technology, University of Pécs Boszorkány u.2. Pécs, Hungary*
**Corresponding author: e-mail address: fulop@mik.pte.hu*

[2] *Faculty of Engineering and Information Technology, University of Pécs Boszorkány u.2. Pécs, Hungary*
e-mail address: Gyorgy.Polics@mik.pte.hu

ABSTRACT

Ventilation of buildings and homes is a key issue both from comfort and energy aspects.
However to determine the average ventilation air flow or the Air Change Rate (ACH) for a heating season by tests in case of natural ventilation, involve certain difficulties.

Essential requirements when testing a physical phenomenon:
1) The test itself must not influence the tested phenomena.
2) The test must be repeatable delivering the same result each time.

Tracer gas test does not influence the tested phenomena but repeated tests do not deliver the same result due to the constantly changing ambient condition e.g. temperature and wind speed affecting the ACH tremendously.
Repeated BlowerDoor tests deliver the same result assuming that the wind speed during the tests remains below the threshold, but the ACH evinced by the test is much higher than the normal due to the higher than normal natural test pressure difference.

The aim of the research introduced by the paper was to elaborate a method to specify the average ACH for a heating season based on field tests.

Test method:
1) Testing the leakage by BlowerDoor test at +50 and -50 Pa pressure difference, and performing a series of tests at various pressure differences. Plot the correlation curve against logarithmic scale.
2) Performing a series of tests to measure the real ACH values by tracer gas technology throughout a whole heating season during various ambient conditions.
3) Searching cross correlation between the temperature difference, the wind speed and the real ACH.
4) Searching correlation between real ACH values and the BloweDoor test results.

The result of this approximation is a multivariate equation visualised as a surface chart. This equation with the parameters provides a more sophisticated method than the currently available ones to estimate the average ACH for a heating season. The method developed is suitable to estimate the average ACH for various climate zones.

KEYWORDS

Air Change Rate (ACH), Tracer gas test, BlowerDoor test, Heating season avg ACH

1 INTRODUCTION

Ventilation of buildings and homes is a key issue both from comfort and energy aspects. However to determine the average ventilation air flow or the Air Change Rate (ACH) for a heating season by tests in case of natural ventilation, involve certain difficulties.

Essential requirements when testing a physical phenomenon:

1) The test itself must not influence the tested phenomena.
2) The test must be repeatable delivering the same result each time.

Tracer gas test does not influence the tested phenomena but repeated tests do not deliver the same result due to the constantly changing ambient condition e.g. temperature and wind speed affecting the ACH tremendously.

Repeated BlowerDoor tests deliver the same result assuming that the wind speed during the tests remains below the threshold, but the ACH evinced by the test is much higher than the normal due to the higher than normal natural test pressure difference.

The aim of the research introduced by the paper was to elaborate a method to specify the average ACH for a heating season based on field tests.

2 BACKGROUND

Ventilation represents a significant proportion of the thermal energy requirement of houses and buildings. The overwhelming majority of the existing building stock has natural ventilation. When auditing a house or a building with natural ventilation the ventilation air flow is estimated that may lead to significant inaccuracy.

In order to achieve an acceptable accuracy in energy audit ventilation air flow should be measured. There is need for a procedure to specify the ventilation air flow for the heating season by a test or series of tests performed in a short period of time.

Driving forces of natural ventilation:

2.1 Stack effect

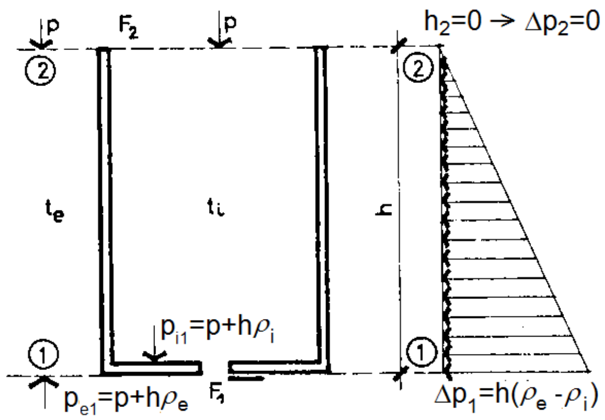


Figure 1: Pressure difference as the function of the height and the temperature difference

$$\Delta p = h \cdot g \cdot (\rho_e - \rho_i) \quad [\mathrm{kg/(m{\cdot}s^2)}]\ \ [\mathrm{Pa}] \tag{1}$$

Where

h : height [m]

g : gravitational acceleration [m/s^2]

ρ : air density [kg/m^3]

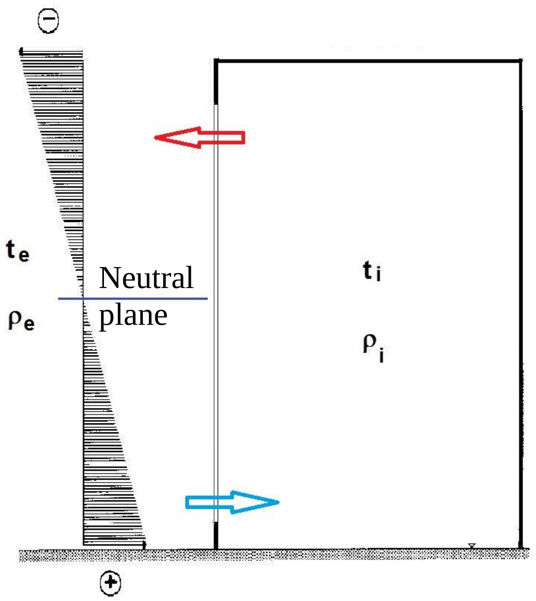


Figure 2: Pressure difference distribution in case of an opening on the facade

2.2 Wind effect

In case of obstacles airflow speed changes that results dynamic pressure. Dynamic pressure is proportional to the square of the velocity. In case the velocity drops to zero:

$$p_d = \frac{\rho}{2} \cdot v^2 \quad [\mathrm{kg/(m{\cdot}s^2)}]\ \ [\mathrm{Pa}] \tag{2}$$

Where

ρ : air density [kg/m^3]

v : velocity [m/s]

3 DISCUSSION

Since the driving force of the natural air flow is the function of both the temperature difference as well as the wind speed the natural air flow (or ACH) can be characterised by bivariate function:

$$n = a \cdot \Delta T + b \cdot v^2 + c \quad [1/h] \tag{3}$$

Where

n: ACH

ΔT : temperature difference [K]

v : wind speed [m/s]

c = 0

Search for the minimum of the function by Least Squares method i.e. minimize the sum of squared deviations:

$$F(a,b,c)=\sum_{i=1}^{N}\left(a\cdot\Delta T[i]+b\cdot v^{2}[i]+c-n[i]\right)^{2} \quad (4)$$

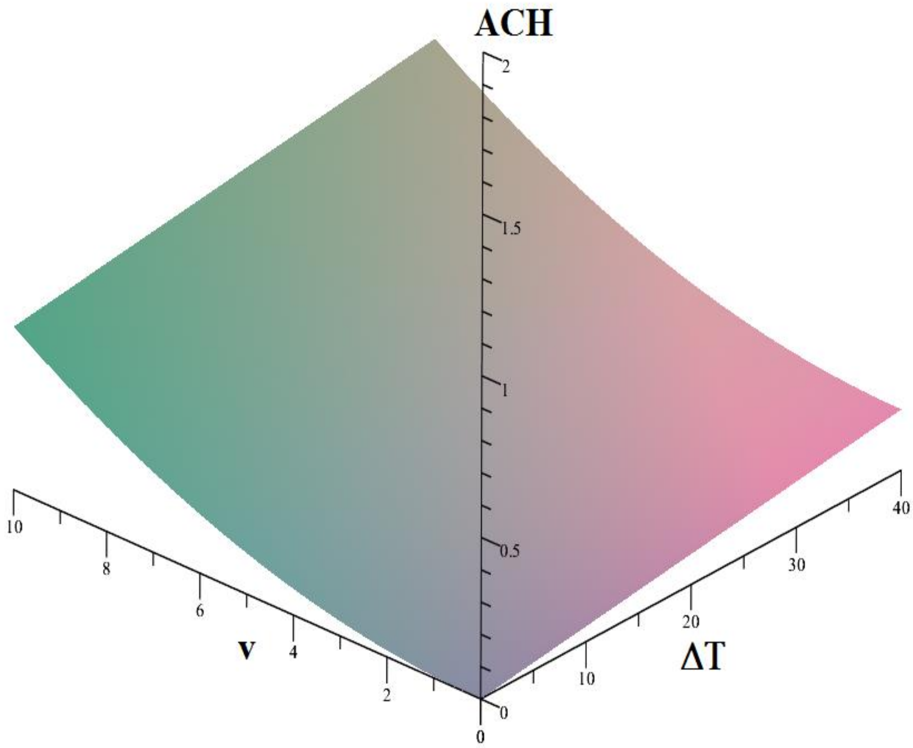


Figure 3: ACH as the function of the wind speed (v) and the temperature difference between the room and the ambient (ΔT)

4 TEST METHOD

Test method:

1) Testing the leakage by BlowerDoor at +50 and -50 Pa pressure difference, and performing a series of tests at various pressure differences. Plot the correlation curve against logarithmic scale.
2) Performing a series of tests to measure the real ACH values by tracer gas technology throughout a whole heating season during various ambient conditions.
3) Searching cross correlation between the temperature difference, the wind speed and the real ACH.
4) Searching correlation between real ACH values and the BloweDoor test results.

The result of this approximation is a multivariate equation visualised as a surface chart. This equation with the parameters provides a more sophisticated method than the currently available ones to estimate the average ACH for a heating season. The method developed is suitable to estimate the average ACH for various climate zones.

5 TEST RESULTS

5.1 BlowerDoor test results

The test room leakage is shown in Fig.4. as a function of the pressure difference between inside and outside.

$$\dot{V} = C \cdot \Delta p^n \quad \left[m^3 / h\right] \tag{5}$$

Where

$\dot{V}$: air leakage at given pressure difference [m^3/h]

Δp : pressure difference [Pa]

C : Constant

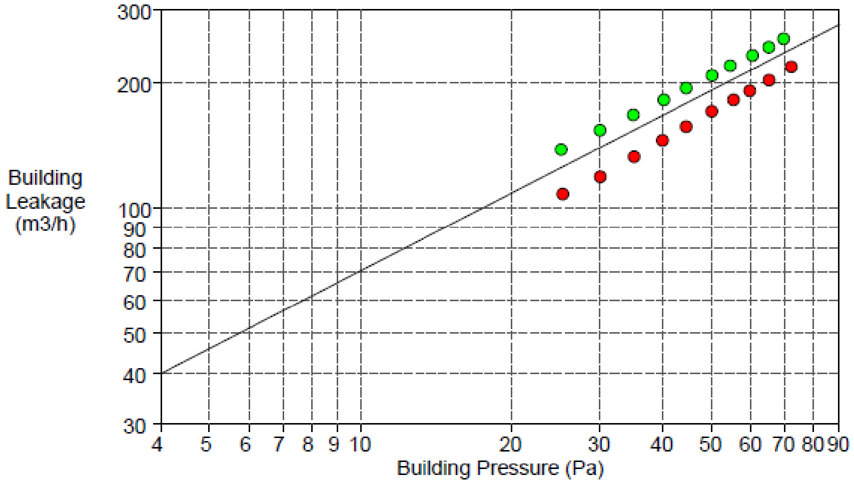


Figure 4: Test room leakage as a function of the pressure difference

5.2 Tracer Gas test results

The method used: Concentration Decay

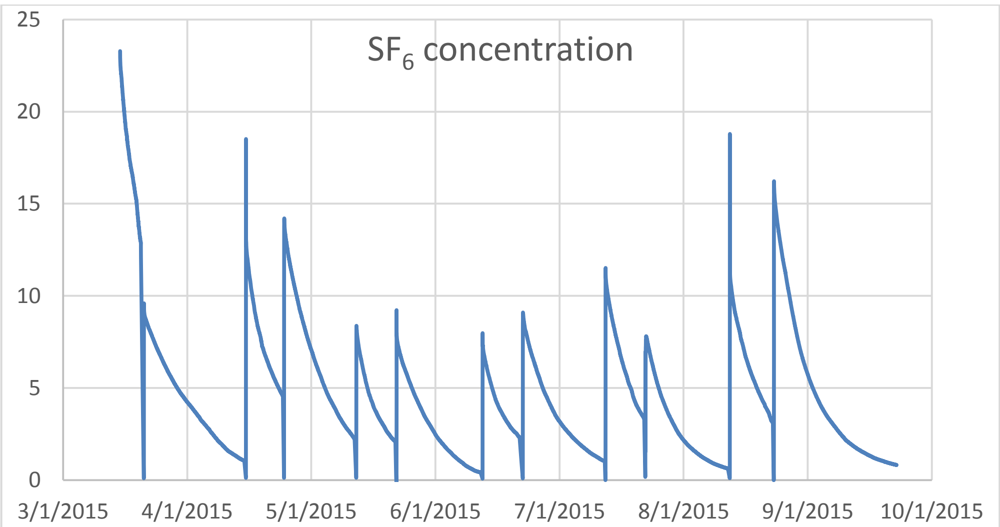


Figure 5: Example of Tracer Gas tests

Evaluation of the Concentration Decay test results:

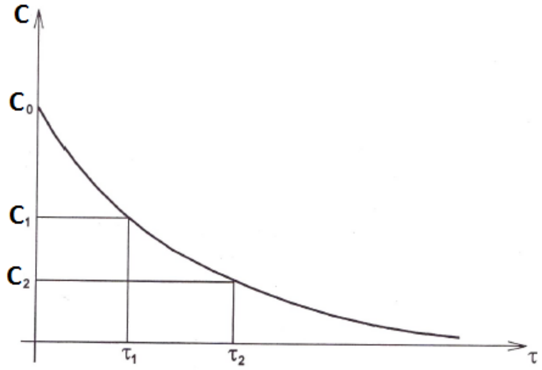


Figure 6: Evaluation of Tracer Gas Concentration Decay test results

$$n = \frac{\ln(C_{r1}) - \ln(C_{r2})}{\Delta\tau} \left[\frac{1}{h}\right] \tag{6}$$

Where

n: ACH [1/h]

τ: Time

$C\tau$: Concentration at time τ [ppm]

5.3 Summary of the test results

Table 1: Summary Table of the Test Results

#	ACH [1/h]	Δt [°C]	v [m/s]	t_i [°C]	t_e [°C]	ρ_e [kg/m3]	p_{din} [Pa]	ρ_i [kg/m3]	Stack Effect [Pa]	Δp TOTAL [Pa]
1	0,139	14,8	0,45	23,54	8,71	1,253	0,126	1,19	1,659	1,785
2	0,093	16,9	0,41	23,61	6,74	1,262	0,107	1,19	1,9	2,007
3	0,136	16,6	1,45	23,81	7,21	1,26	1,318	1,189	1,865	3,183
4	0,127	20,4	1,17	23,98	3,62	1,276	0,874	1,189	2,316	3,19
5	0,19	19,7	1,41	24,01	4,36	1,273	1,266	1,188	2,229	3,495
6	0,142	20	0,99	24,13	4,1	1,274	0,62	1,188	2,273	2,893
7	0,176	20,2	1,66	23,75	3,57	1,276	1,76	1,19	2,298	4,057
8	0,14	22,3	0,57	24,19	1,89	1,284	0,212	1,188	2,551	2,763
9	0,118	24,3	0,76	25,22	0,93	1,289	0,375	1,184	2,779	3,154
10	0,146	21,4	0,41	25,02	3,63	1,276	0,106	1,184	2,424	2,53
11	0,144	21,6	0,58	24,58	2,99	1,279	0,215	1,186	2,457	2,671
12	0,151	6,6	0,39	24,97	18,39	1,211	0,092	1,185	0,708	0,8
13	0,127	9,5	1,01	25,37	15,86	1,222	0,618	1,183	1,031	1,649
14	0,076	4,5	0,29	24,32	19,82	1,205	0,049	1,187	0,483	0,532
15	0,238	12,5	2,78	25,69	13,14	1,234	4,773	1,182	1,371	6,144
16	0,149	13,3	1,41	24,98	11,7	1,24	1,227	1,185	1,462	2,69
17	0,173	11	0,69	25,37	14,42	1,228	0,29	1,183	1,193	1,483
18	0,149	7,5	0,53	24,97	17,43	1,215	0,172	1,185	0,815	0,987
19	0,235	11,5	0,68	25,76	14,26	1,229	0,281	1,182	1,252	1,533
20	0,193	13,1	2,26	23,6	10,52	1,245	3,172	1,19	1,454	4,625
21	0,135	10,2	0,77	23,52	13,32	1,233	0,364	1,19	1,123	1,488
AVG:	**0,151**	**15,14**	**0,98**	**24,49**	**9,36**	**1,251**	**0,858**	**1,187**	**1,697**	**2,555**

5.4 Results of the regression analysis

$$\text{ACH} = n = a \cdot \Delta T + b \cdot v^2 + c \quad [1/h] \tag{3}$$

Constants of the regression analysis:

$$ACH = 0{,}009461005 \cdot \Delta\,\mathrm{T} + 0{,}010126854 \cdot \mathrm{v}^2 + 0 \tag{7}$$

The equation above provide an acceptable approximation for the test results but not yet validated against a large number of test at various sites.

6 CONCERNS

The spread of the data is too large the method to be proved with full confidence.

Possible reasons:

1) The BlowerDoor set of equipment was installed in the door. As a consequence the air leakage through the gaps of the door is not included in the BlowerDoor test, but is in the Tracer Gas tests. The only other opening is the window.
2) Wind direction is not recorded, assumed that it is perpendicular to the surface. There any many turbulences around the test façade.
3) The properties of the air flow through the gaps might change as a consequence of the variation of the air speed. Might be turbulent at high speed during BloerDoor test but laminar during normal conditions.
4) The properties of the air gaps might change (become wider or narrower) as a consequence of the higher than normal pressure.

7 CONCLUSIONS

The overall aim of the research has been fulfilled: to elaborate a method to calculate the natural ACH based on BlowerDoor test by taking the temperature difference and the wind speed into the account. Applying the method the average ACH can be specified on the basis of the average ambient temperature and the average wind speed of the region for certain period of time e.g. heating season. This method is simple but reasonably refined. The ACH derived by the method increases the reliability of the energy balance calculations.

However the parameters and constants are not finalised yet. Further tests are required to achieve more reliable values.

8 ACKNOWLEDGEMENTS

Acknowledgement for Dr. Hámori Ildikó Perjésiné associate professor, Head of Mathematics Department, Faculty of Engineering and Information Technology, University of Pécs, HU for her support in mathematics and Maple.

Acknowledgement for László Budulski, engineer, HVAC Department, Faculty of Engineering and Information Technology, University of Pécs, HU for his contribution in testing and data processing.

Acknowledgement for EU Hungary-Croatia IPA (Instrument for Pre-Accession Assistance) Cross-border Co-operation Programme for funding the equipment procurements.

9 REFERENCES

Rex W. Moore, CIH, CSP Boelter & Yates, Inc. presented by Catherine E. Simmons, CIH Park Ridge, Illinois "*Tracer gas test methods to diagnose ventilation-related indoor environmental quality problems,*" Presentation. 19.06.2006

Innova AirTech Instruments, *Ventilation measurements and other tracer-gas applications,* http://www.innova.dk.

Max Sherman "*The use of blower-door data*" Energy Performance of Buildings Group, Energy and Environment Division, Lawrence Berkeley Laboratory, University of California, Berkeley, California, March 13, 1998

Dipl.-Ing. Stefanie Rolfsmeier, “*Air Tightness in Passive Houses*”. BlowerDoor GmbH, Energie- und Umweltzentrum 1, D-31832 Springe

EN-832 Standard

DIN13829 Bestimmung der Luftdurchlässigkeit von Gebäuden

EXPERIMENTAL ANALYSIS OF MICROSCALE TRIGENERATION SYSTEMS TO ACHIEVE THERMAL COMFORT IN SMART BUILDINGS

Parantapa Sawant[*1] and Jens Pfafferott[2]

1 Institute for Energy System Technologies
Am Güterbahnhof 1a
77652 Offenburg, Germany
* parantapa.sawant@hs-offenburg.de

2 Offenburg University of Applied Sciences
Badstrasse 24
77654 Offenburg, Germany

ABSTRACT

The transformation of the building energy sector to a highly efficient, clean, decentralised and intelligent system requires innovative technologies like microscale trigeneration and thermally activated building structures (TABS) to pave the way ahead. The combination of such technologies however presents a scientific and engineering challenge. Scientific challenge in terms of developing optimal thermo-electric load management strategies based on overall energy system analysis and an engineering challenge in terms of implementing these strategies through process planning and control. Initial literature research has pointed out the need for a multiperspective analysis in a real life laboratory environment. To this effect an investigation is proposed wherein an analytical model of a microscale trigeneration system integrated with TABS will be developed and compared with a real life test-rig corresponding to building management systems. Data from the experimental analysis will be used to develop control algorithms using model predictive control for achieving the thermal comfort of occupants in the most energy efficient and grid reactive manner.
The scope of this work encompasses adsorption cooling based microscale trigeneration systems and their deployment in residential and light commercial buildings.

KEYWORDS

Adsorption Cooling, Experimental Investigation, Grid Integration, Microscale Trigeneration, Predictive Control, Thermal Comfort in Buildings

1 INTRODUCTION

Trigeneration systems or combined cooling, heating and power (CCHP) systems are a technological extension of combined heat and power (CHP) or cogeneration systems. Here the waste heat of the cogeneration process is used to produce cooling, usually in thermally activated chillers. Due to the fact that an energy cascade is developed and a single fuel source is utilised for multiple energy conversions the exergy efficiency of trigeneration systems is higher than individual conversion units. Ghaebi *et al.* (Ghaebi et al., 2010) demonstrated this is in their in-depth exergy and thermoeconomic analysis of a trigeneration system. Wu *et al*. (Wu et al., 2006) and Jrade *et al*. (Jrade et al., 2014) have also discussed the benefits of trigeneration in their reviews and the progress it has made in the last few decades in terms of different prime movers, chillers, operation strategies and government policies supporting its further development and implementation.

Trigeneration systems are frequently deployed in small (<1MWel) and medium scale (1-10 MWel) applications in industries and commercial complexes. Here, absorption based thermal chillers that utilise high temperature waste heat (>100°C) are most dominant. However in the past decade studies, e.g. Zhai et.al. (Zhai and Wang, 2008), have shown promising results of microscale (<20kWel) adsorption cooled trigeneration systems in terms of grid flexibility and demand side management for smaller buildings having less cooling load. The capability of adsorption chillers to work with low temperature waste heat (60–80°C) increases their integration potential with other renewable sources and thermally activated building structures (TABS). The focus of this research is only on adsorption cooled microscale trigeneration systems and hereon referred to as simply microscale trigeneration systems.

2 MULTIPERSPECTIVE ANALYSIS

Although microscale trigeneration systems have the potential to play a vital role in the building energy sector their deployment is hindered due to interdisciplinary scientific and engineering challenges as summarised in figure 1.

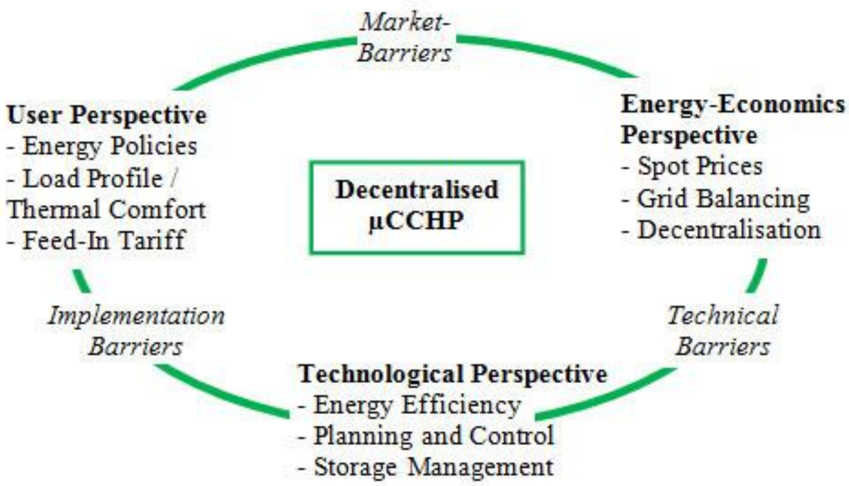


Figure 1: Multiperspective analysis of decentralised µCCHP system

Scientific challenges arise due to supply and demand mismatch for power and the heating/cooling loads especially in a smart building setup with combination of other renewable sources. Here, energy storages and integrated systems demonstrate the potential to enhance the system's flexibility for optimal scheduling. However, this increases the system's complexity and accordingly optimised operation strategies considering multiple perspectives (energy-economical, technological and user based) must be developed.

Various optimisation approaches for CCHP systems focusing on system operational costs, primary energy consumption and emission reduction have been published in recent years. For instance, Chandan *et al*. (Chandan et al., 2012) proposed a non-linear programming (NLP) method by making use of forecasted electric and thermal demands for the optimal control of a CCHP plant with a thermal storage. Rong *et al*. (Rong et al., 2005) proposed an enhanced linear programming method specialised to optimise operating costs of producing any three energy commodities simultaneously. Kavvadias *et al.* (Kavvadias et al., 2010) and Piacentino *et al*. (Piacentino et al., 2008) followed a multi-objective approach based on the cost and emissions reduction taking into account the energy load and price variations on an hourly basis. Zhao *et al*. (Zhao et al., 2015) demonstrated the application of model predictive control (MPC) using a NLP method to develop optimal scheduling of active and passive energy systems in buildings based on time sensitive electricity prices. Although no single method has been established to offer the best possible solution, MPC is the forerunner for recent optimisation studies because it facilitates application of powerful programming algorithms such as nonlinear, mixed integer linear and mixed integer nonlinear algorithms.

Implementation of such complex scheduling strategies further leads to engineering challenges such as planning, standardising, operation and control of these systems. Thus, real life test-rig research is necessary to develop, validate and implement algorithms that will enhance the integration of microscale trigeneration systems into smart grids for residential and light commercial building applications.

Majority laboratory research focuses on analysis and improvements of the adsorption chiller or the thermal storages on a component level. For example Huangfu *et al*. (Huangfu et al., 2007) performed an experimental investigation for analysing the performance of the adsorption chiller under varying heating conditions. Wang *et al*. (Wang et al., 2009) pointed out the adsorption deterioration as a non-negligible factor adversely affecting the service life of adsorption chillers. Schmidt *et al.* (Schmidt et al., 2014) introduced the concept of heat recovery between the adsorption and desorption cycle of the chiller using stratified storages. However, some researchers such as Kong *et al*. (Kong et al., 2005) developed a comprehensive database for the operating parameters of an entire microscale trigeneration system in a test facility with a natural gas engine and a two bed adsorption chiller. Becker *et al*. (Becker et al., 2013), Angrisani *et al.* (Angrisani et al., 2012), Henning *et al.* (Henning et al., 2010) and a few others have also performed similar real life experimental investigations of the system as a whole in laboratory environments. Although the potential of these systems was established in these studies a characteristic deduction has been the need for improved control strategies, integration of other building heating/cooling technologies and standardisation of system planning and operation.

Raimondo et al. (Raimondo et al., 2013) demonstrated the ability to utilise water at higher temperatures (such as those achieved by thermal chillers) for space cooling in thermo-active building system (TABS) during field tests supported by dynamic simulations. In an extensive field test and measurement study Kalz et al. (Kalz et al, 2014) recorded and analysed data for thermal comfort and energy efficient cooling of 42 office buildings in Europe. It was stated that TABS system can achieve energy efficient cooling solutions for buildings and in most cases an average room temperature of 22,5-25,5°C was achieved. However for higher thermal comfort requirements and fluctuating user profile additional backup systems are necessary. Another field test project (Fraunhofer, 2015) will focus on providing an integrated energy and building concept considering aspects on renewable energy integration and energy efficiency.

Thus due to the economic and environmental benefits it is becoming more meaningful to deploy microscale trigeneration systems in combination with TABS for decentralised grids and buildings. However, for analysing the interdependency of the integrated components in such systems and developing optimal scheduling it is necessary to perform laboratory tests.

This study will therefore address the issue of developing control algorithms and testing them on a real life test-rig to realise the potential of adsorption cooled microscale trigeneration systems integrated with TABS in an energy efficient and grid reactive manner.

3 PROPOSED INVESTIGATION

A flow chart of the proposed plan to investigate the optimal scheduling of a trigeneration system is shown in figure 2. Analytical models for the system components will be developed using basic physical energy balance equations. The focus will be on making robust models to facilitate the use of MPC. Simultaneously an experimental lab is being set-up. The lab is described in more detail in the following section of the paper. A mathematical optimisation problem will be developed using load profile of building, spot prices, weather forecasts and feed-in tariff data as inputs. The constraints on optimisation will be the thermal comfort of occupants, security of power supply and the physical constraints of the components. An MPC solver will be implemented to provide optimal scheduling strategies which will be then implemented in the real life test-rig. Feedback from the lab will be used for validating and further developing the models.

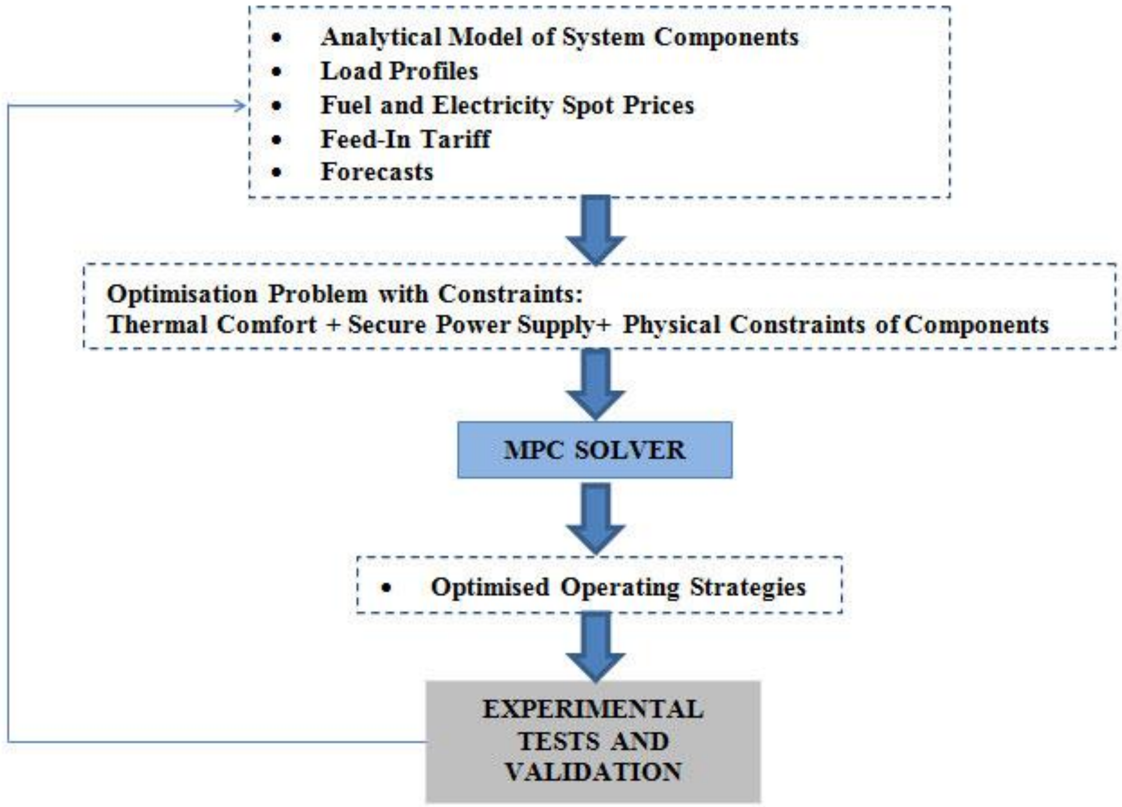


Figure 2: Flowchart of proposed research

4 THE EXPERIMENTAL SET-UP

A microscale trigeneration lab is being set-up to facilitate thermo-electric load management for the smart microgrid concept of the institute. The specifications for main components are shown in Table 1 below.

Table 1: Specifications of main components in the microscale trigeneration lab

Components	Abbreviation	Specification
Combined Heat and Power Unit	CHP	5kWel; 10kWth; 66%eta_ th; 30%eta_el; Fuel Oil
Hot Storage	HOT/STRG	1500 L; with 6kWel Heat Coil
Adsorption Chiller	AdCM	12kW cooling max.; 0,65 COP max.
Cold Storage	COLD/STRG	1450 L
Mechanical Chiller/ Heat Pump	MC/HP	12,9kW (cooling nom.); 16,7kW (heating nom.); 3,75kWel
Cooling Tower	CT	29kW(cooling max.); 24.000 m³/h fan airflow; 0,54kWel

The trigeneration lab and the climate chamber are shown in figure 3. The trigeneration lab will integrate with the TABS for heating and cooling application in the climate chamber.

Figure 3: Trigeneration unit and climate chamber (CC) at the Institute of Energy System Technologies (INES)

A multiple regression analysis based control algorithm is developed by Schmelas *et al.*, (Schmelas et al., 2015) for the TABS in the climate chamber. It controls the system to maintain a temperature profile in the test chamber and correspondingly a thermal comfort profile corresponding to building standards, e.g. (figure 4).

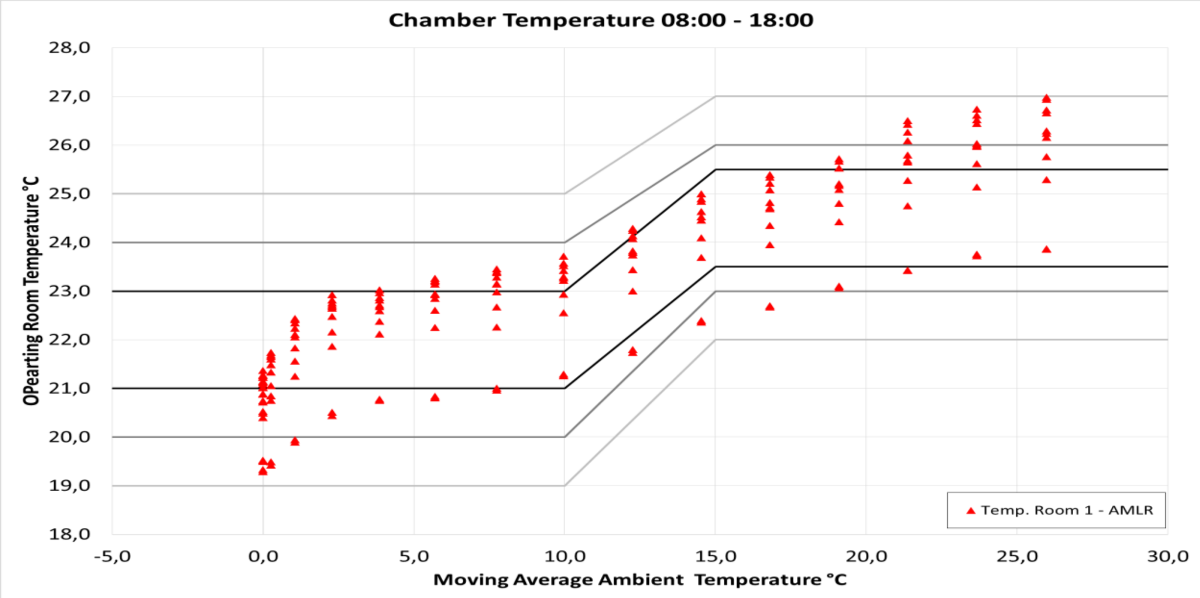


Figure 4: Thermal comfort categories

4.1 Modes of Operation

Different operation modes of the trigeneration system are possible. The basic winter or summer season based operation modes are described in Figures 5 to 8 below.

Winter Electricity Production: Electricity from CHP will be fed to the grid. Heat will be stored and distributed from the hot storage tank as shown in figure 5.

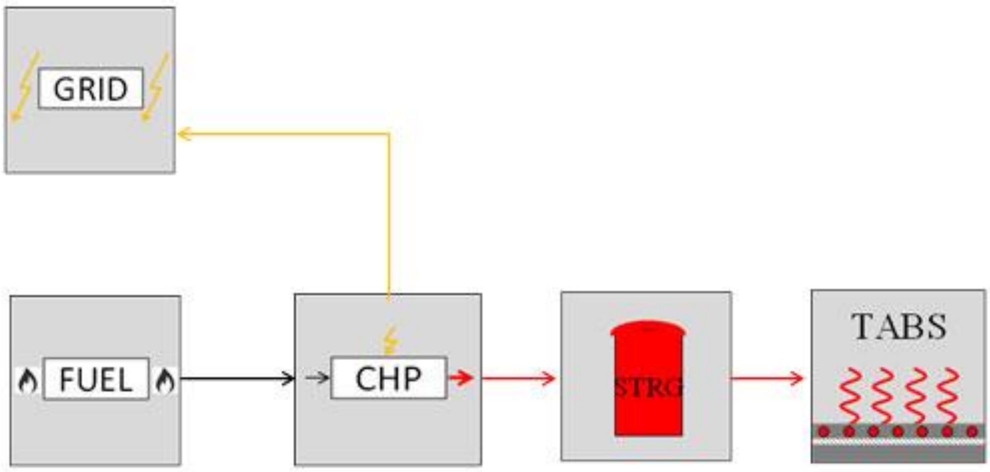


Figure 5: Winter electricity production mode

Winter Electricity Consumption: Electricity will be taken from the grid to operate the reversible heat pump and heat will be stored and distributed from the hot storage tank as shown in figure 6

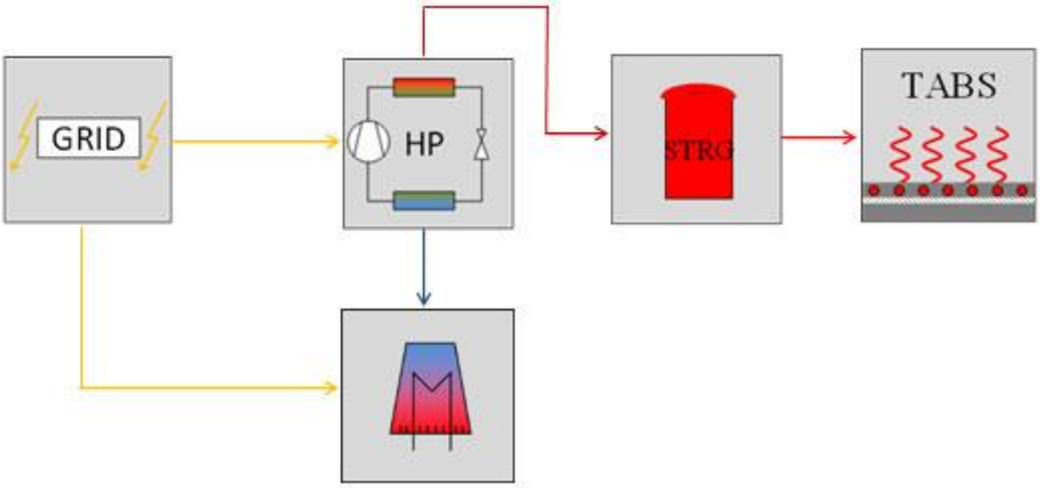


Figure 6: Winter electricity consumption mode

Summer Electricity Production: Electricity from CHP will be fed to grid and heat provided to the adsorption chiller. The chilled water from adsorption chiller will be stored and distributed from the cold storage tank as shown in figure 7

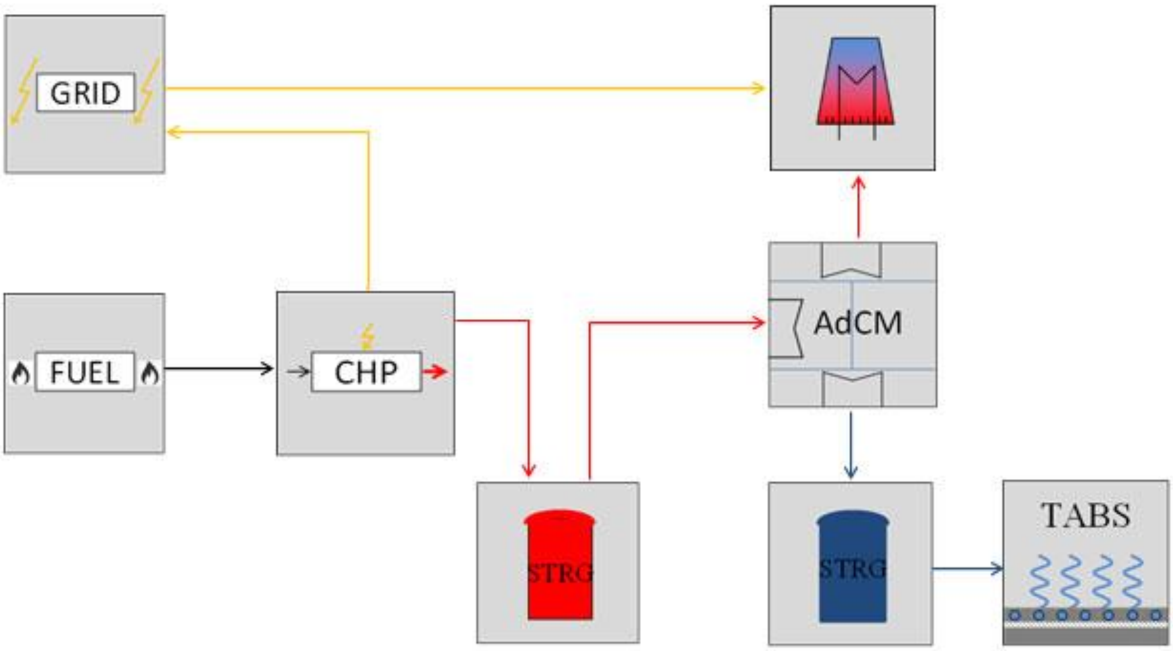


Figure 7: Summer electricity production mode

Summer Electricity Consumption: Electricity from grid will be used to run electrical chiller and chilled water will be stored and distributed from the cold storage tank as shown in figure 8

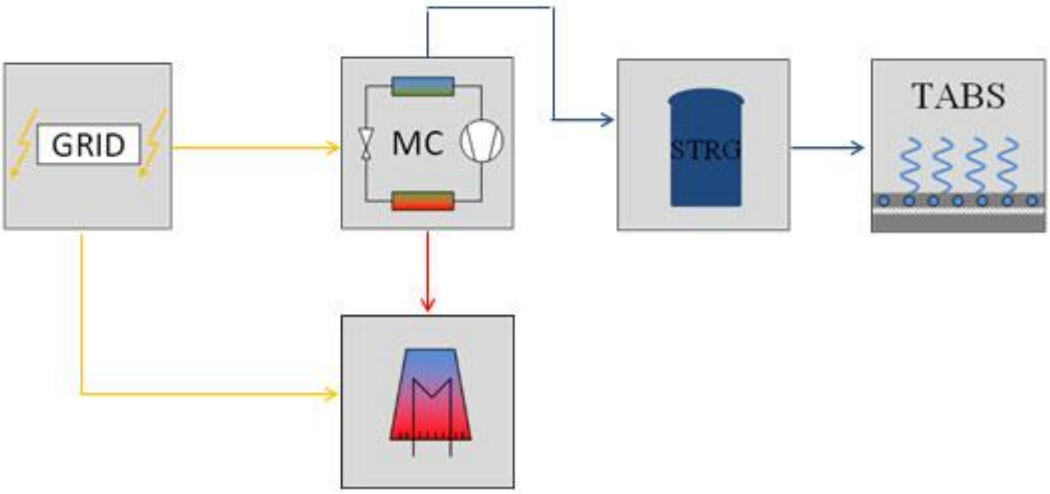


Figure 8: Summer electricity consumption mode

Further advanced operation modes will be analysed to increase overall system efficiency. For instance a step cooling cascade will be developed between the two chillers; where in the thermal chiller will serve as heat sink for the electrical chiller. Here, the higher quality cooling power from the electrical chiller will be supported by waste heat from the CHP and also in turn increase the operating time of the thermal chiller. The possibility to deploy the thermal chiller as a heat pump and the adsorption-desorption exergy balance through stratified storage will also be analysed. The MPC solver will decide on the optimal operation mode in pre-defined time steps for the process shown in figure 9 below:

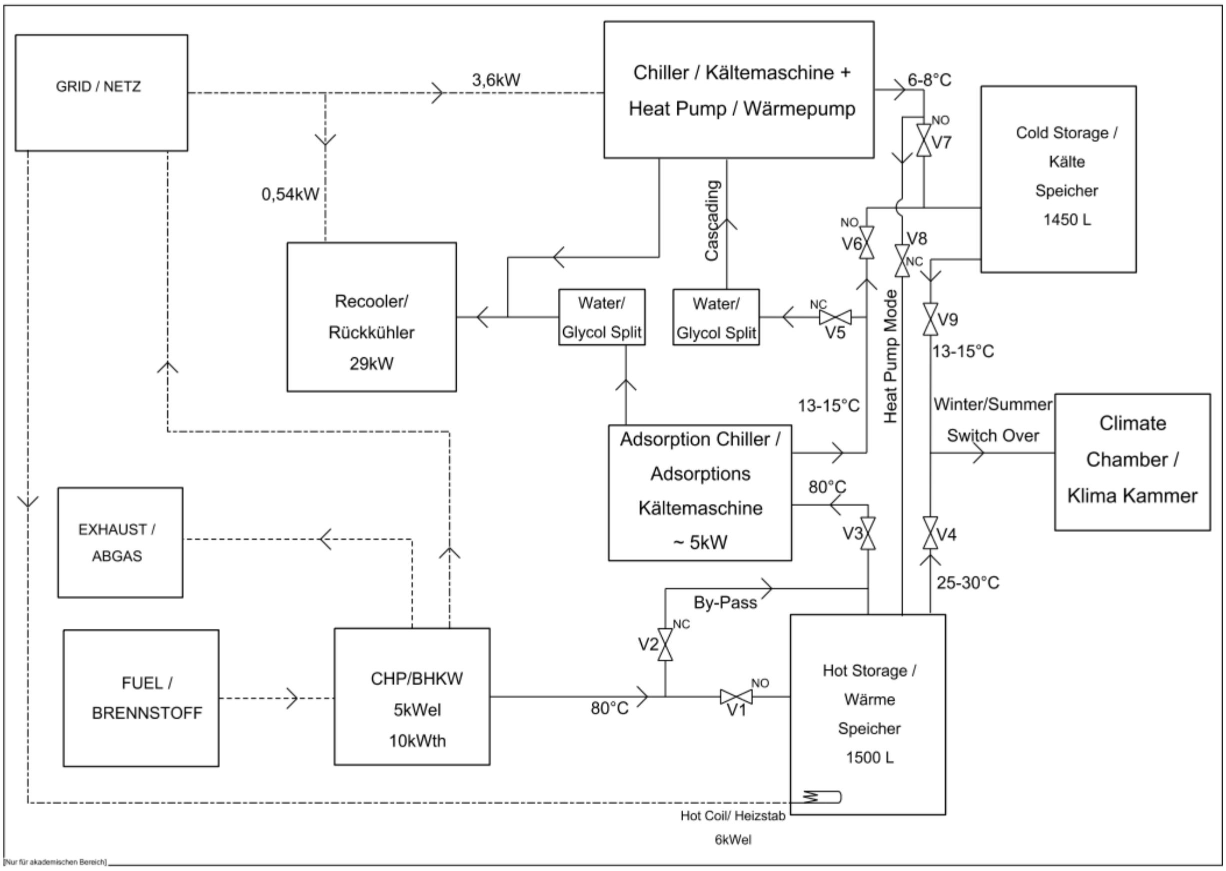


Figure 9: Basic process flow diagram of the trigeneration process

The modi operandi is summarised in terms of the actuating valves in Table 2 below:

Table 2: Valve status for different operation modes

Operation Modes	Status of Valves	
	ON	OFF
Summer		
Electricity Production	V1, V3, V6, V9	V2, V4, V5, V8, V7
Electricity Consumption	V7, V9	V1, V2, V3, V4, V5, V6, V8
Advanced System Design	V1, V3, V5, V7, V9	V2, V4, V6, V8
Winter		
Electricity Production	V1, V4	V2, V3, V5, V6, V7, V8, V9
Electricity Consumption	V4, V8	V1, V2, V3, V5, V6, V7, V9

The process and instrumentation diagram for the test-rig is shown in figure 10 below. The three primary circuits hot water, cold water and recooling are highlighted with red, blue and green respectively.

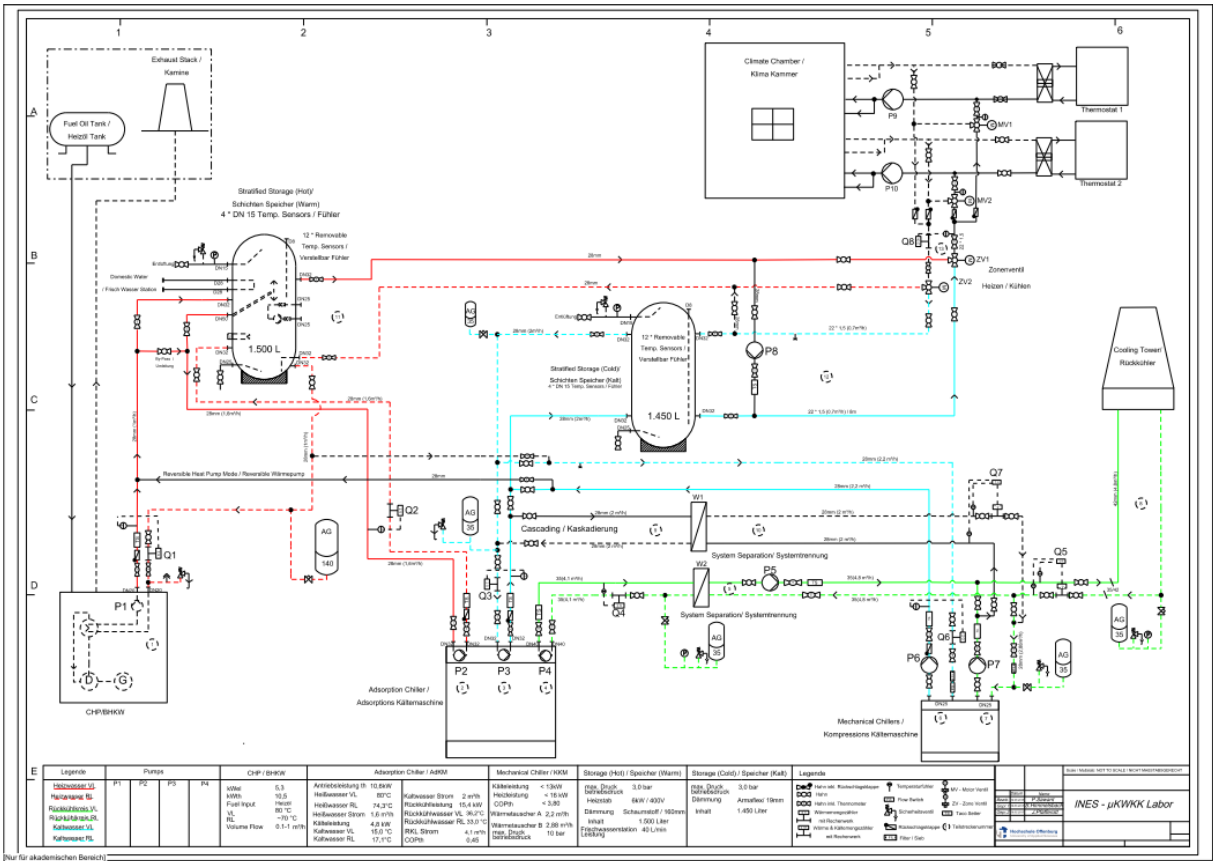


Figure 10: Process and instrumentation diagram of the trigeneration lab

5 CONCLUSIONS

The need for real life experimental analysis of an integrated microscale trigeneration system supported by analytical modelling and simulation is established. An overview of the laboratory set-up and proposed investigation is given. The test facility will help understand the flexibility potential of trigeneration systems for better thermo-electric demand side management in terms of the crowd energy concept and decentralised microgrids.

6 REFERENCES

Ghaebi, H., Amidpour, M., Karimkashi, S., & Rezayan, O. (2011). Energy, exergy and thermoeconomic analysis of a combined cooling, heating and power (CCHP) system with gas turbine prime mover. *International Journal of Energy Research*, *35*(8), 697-709.

Wu, D. W., & Wang, R. Z. (2006). Combined cooling, heating and power: a review. *progress in energy and combustion science*, *32*(5), 459-495.

Jradi, M., & Riffat, S. (2014). Tri-generation systems: Energy policies, prime movers, cooling technologies, configurations and operation strategies. *Renewable and Sustainable Energy Reviews*, *32*, 396-415.

Zhai X.Q., Wang R.Z. (2008) A review for absorbtion and adsorbtion solar cooling systems in China, *Renewable and Sustainable Energy Reviews,* 13 (2009) 1523–1531

Chandan, V., Do, A. T., Jin, B., Jabbari, F., Brouwer, J., Akrotirianakis, I., ... & Alleyne, A. (2012, June). Modeling and optimization of a combined cooling, heating and power plant system. In *American Control Conference (ACC), 2012* (pp. 3069-3074). IEEE.

Rong, A., & Lahdelma, R. (2005). An efficient linear programming model and optimization algorithm for trigeneration. *Applied energy*, *82*(1), 40-63.

Kavvadias, K. C., & Maroulis, Z. B. (2010). Multi-objective optimization of a trigeneration plant. *Energy Policy*, *38*(2), 945-954.

Piacentino A, Cardona F. An original multi-objective criterion for the design of small-scale polygeneration systems based on realistic operating conditions. *Appl Therm Eng*, 2008, 28(17–18), 2391–404.
Zhao, Y., Lu, Y., Yan, C., & Wang, S. (2015). MPC-based optimal scheduling of grid-connected low energy buildings with thermal energy storages. *Energy and Buildings*, *86*, 415-426.

Huangfu, Y., Wu, J. Y., Wang, R. Z., Kong, X. Q., & Wei, B. H. (2007). Evaluation and analysis of novel micro-scale combined cooling, heating and power (MCCHP) system. *Energy Conversion and Management*, *48*(5), 1703-1709.

Wang, D. C., Li, Y. H., Li, D., Xia, Y. Z., & Zhang, J. P. (2010). A review on adsorption refrigeration technology and adsorption deterioration in physical adsorption systems. *Renewable and Sustainable Energy Reviews*, *14*(1), 344-353.

Schmidt, F., Joshi, C., Schwamberger, V., & Taheri., H. (Feb, 2014). Stratisorp-systementwicklung: Schichtspeicher Adsorptionswärmepumpe für Heiz- und ühlanwendungen in Gebäuden, Project Report, Karlsruher Institut für Technologie (KIT), Research Group Energy and Building Technology. (In German)

Kong, X. Q., Wang, R. Z., Wu, J. Y., Huang, X. H., Huangfu, Y., Wu, D. W., & Xu, Y. X. (2005). Experimental investigation of a micro-combined cooling, heating and power system driven by a gas engine. *International Journal of Refrigeration*, *28*(7), 977-987.

M. Becker, B. Ander, K. Sturm T. Patel and J. Braun. Regenerativ betrieben, innovative Kraft-Wärme-Kälte-Kopplungsanlage (InnoKKK), in FORETA Ergebnisse der Forschungsverbundes Energieeffiziente Technologien und Anwendungen, Verlag Attenkofer, Munich 2013, pp. G-1-G-26 (In German)

Angrisani, G., Rosato, A., Roselli, C., Sasso, M., & Sibilio, S. (2012). Experimental results of a micro-trigeneration installation. *Applied Thermal Engineering*, *38*, 78-90.

Henning, H. (July 2010), PolySMART-Publishable Final Report, Fraunhofer ISE

Raimondo D and Olesen B et al. (2013), Field test of a thermal active building system (TABS) in an office building in Denmark, *Proceedings of BS2013: 13th Conference of International Building Performance Simulation Association, Chambéry, France, August 26-28*

Kalz, D., Herkel, D., Sonntag, M., & Pfafferott, J. (2014), Energieeffiziente Kühlung und thermischer Raumkomfort: Erfahrungen aus dem Langzeitmonitoring von 42 Bürogebäuden, *Bauphysik* , Volume5/2014 (In German)

Fraunhofer(2015), Integral energy concept for a building complex in Munich, *Fraunhofer Institute of for Building Physics, Projects: Building Systems and Services*

Schmelas, M., Feldmann, T., & Bollin, E. (2015). Adaptive predictive control of thermo-active building systems (TABS) based on a multiple regression algorithm. *Energy and Buildings*, *103*, 14-28.

IMPACT OF NATURAL VENTILATION IN ENERGY DEMAND AND THERMAL COMFORT OF RESIDENTIAL BUILDINGS IN CATALONIA

Joana Ortiz*[1], Verdiana Russo[1] and Jaume Salom[1]

1 Catalonia Institute for Energy Research – IREC
Jardins de les Dones de Negre, 1, 2ª pl., 08930
Sant Adrià de Besòs, Spain

**Corresponding author: jortiz@irec.cat*

ABSTRACT

The most representative typology of residential buildings of Catalonia has been simulated in TRNSYS to evaluate the impact of both infiltration and natural ventilation. The typology is a block of apartments constructed during 1950-1980.
In this paper the methods used to characterize the infiltration and the natural ventilation are described. The infiltration model (UNE-EN 15242, 2007) considers the tightness of the construction and allows the use of measurement data obtained from experimental studies, as blower door tests. The natural ventilation of the building is single sided ventilation with courtyard effect. The model used (Gids and Phaff, 1982) depends on the indoor temperature, courtyard temperature, outdoor temperature and wind velocity. The control strategy defined for the natural ventilation is based on the results obtained in surveys of the building characterization study done in the framework of the MARIE project (www.marie-medstrategic.eu). The control strategy follows the assumption that the occupants use the natural ventilation to cool the households, and when it is not enough to have comfortable conditions, they use the cooling system.
The results are presented in terms of energy demand and thermal comfort. The evaluation of the comfort is based on the adaptive ASHRAE model (ASHRAE 55, 2004), which is applied in buildings without mechanical cooling systems. The comfort indices evaluated are the Long-term Percentage of Dissatisfied (Carlucci, 2013) and the hours of overheating. The typology has been evaluated in different situations: current building and refurbished building; with natural ventilation and without natural ventilation. The objective is to show the effect of the natural ventilation in different building configurations.

KEYWORDS

Natural ventilation, infiltration, control strategy, thermal comfort, overheating

1 INTRODUCTION

European objectives 20-20-20, defined by the European Council (Energy Performance of Building Directive and Energy Efficiency Directive), are committed with the aim of reducing the greenhouse gas emissions by 20%, saving 20% of energy consumption through increased energy efficiency and promote renewable energy to 20%. In this context, it is needed to develop political guidelines to achieve these goals, establishing techno-economic criteria to make decisions at regional and local level.

Therefore, it is essential to use an approach that enhances the use of passive strategies, both in the design of new housing and in the refurbishment projects. The different constructive solutions, dimension, distribution of spaces, orientation, exposition to wind and the urban environment have an important role in the infiltration and in the natural ventilation, and in consequence in the energy demand and the thermal comfort of the buildings. In addition, the behaviour of the users plays an especially role in the use of the buildings and consequently in its consumption, such as the works done in the framework of the IEA-EBC Annex 66 demonstrate (www.annex66.org).
Within this context, the present paper describes a study where methods for improving the definition of the air leakage and the natural ventilation and how the users interacts with the building, have been implemented in a detailed building model through TRNSYS. The effects of both phenomena are evaluated in terms of thermal comfort and energy demand. In addition, four different scenarios have been compared, in order to analyse the behaviour of passive strategies applied in the refurbishment of buildings.

2 DESCRIPTION OF THE BUILDING'S MODEL

The paper is focused in the most representative typology of residential buildings of Catalonia, which represents the 45% of the dwellings (Garrido et al., 2012). This typology was built during 1951-1980, before the first building regulation (NBE-CT-79, 1979) and is characterized for having a low thermal performance. The building typology is a block of apartments with a commercial ground floor and five residential floors. There are two dwellings per floor with a 78.8 m^2 of surface each one. The typology has been simulated by (TRNSYS 17.1, 2012) in four climates of Catalonia, but in this paper only the results of the climate of Barcelona are presented.

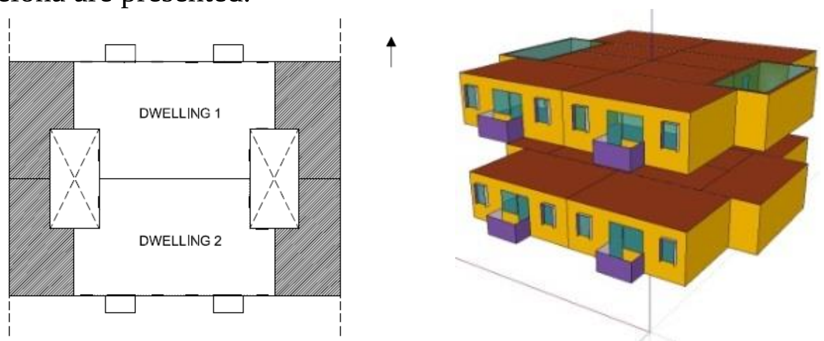


Figure 1: Building typology: block of apartments 1951-1980

The building's geometry (Figure 1) is introduced in the simulation by a multizone 3D model, using the plugin Trnsys3D for Google SketchUp. Only two floors are included, in order to simulate the building with more detail: the standard floor and the under roof floor. Each dwelling is divided following two zonification criteria: night and day use, and orientation. The building model includes the external environment and the corresponding shadings.
In the simulation, the occupancy has been defined as the main driver of the use of the building (heating, cooling, natural ventilation, solar protection and lighting use). For that reason, one of the main objectives is to use realistic profiles of the occupants. These profiles have to reproduce the variability of the actual occupant and, at the same time, their behaviour has to be representative of the average occupant. The stochastic profiles are created from the Time Use Data survey of Spain (INE, 2010). This survey allows knowing what people are doing at each moment of the day. Then, an annual profile was created applying a statistical analysis of the raw data, assigning a state of each occupant: outside of home, passive at home, and active at home. The family composition of the dwelling typology is made up of two adults and a kid.

The energy systems have been defined by a method based on the efficiency of the different parts of the system: generation, emission and control. The efficiency of generation is calculated using (IDAE, 2009) and the efficiency of the emitters and control following (EN 15316, 2008). The building model includes also the lighting and appliances consumptions, in order to take in consideration all the energy uses of the household. The details of the approach used in the simulations are explained in (Salom et al., 2014).

3 METHOD FOR INFILTRATION

3.1 Selection of the method

The infiltration or air leakage is the unintentional introduction of outside air into a building, typically through cracks in the building envelope and through the joint of doors and windows. In order to include a detailed model of infiltration in the building simulation, four different methods have been analysed:

1. K1, K2, K3 approach (ASHRAE, 1989): this method calculates the instantaneous air change rate, depending on outdoor temperature, indoor temperature, wind speed, and K1, K2, K3 coefficients. These coefficients have different values for tight, medium and loose construction. The method can be implemented in TRNSYS, using Type 571 (Thornton, 1998).
2. LBL infiltration model (ASHRAE, 2009): this method calculates the instantaneous air change rate, depending on the effective leakage area (ELA) and the superposition of wind and stack effects. The ELA depends on leakage coefficient and it can be calculated from experimental data of blower door test. The wind and stack effects depends on the outdoor temperature, indoor temperature, wind speed, the height of the building and its environment. The method can be implemented in TRNSYS, using Type 960 (McDowell, 2006).
3. Sherman Grimsrud approach (ASHRAE, 2005): this method is based on the LBL infiltration model too, with the difference that two coefficients are used to consider the superposition of the wind and stack effects. The method can be implemented in TRNSYS using Type 932 (Bradley, 2005).
4. EN15242 method (UNE-EN 15242, 2007): the direct method for exfiltration and infiltration calculates the instantaneous air change rate as a superposition of wind and stack effects. The result depends on outdoor temperature, indoor temperature, the height of the building, wind speed, and a coefficient that considers the pressure difference between windward and leeward sides. In addition, the method takes into consideration the building conditions, using the results of the blower door test (air changes per hour at 50 Pa, n_{50}) to calculate both the wind and stack air change rate. The blower door test is a common test to identify the air leakage of buildings. The method has been implemented in TRNSYS using equations.

In order to choose the infiltration method, the results of these four methods have been compared with the reference values of the PassivHaus design, which is based on (EN-ISO 13790, 2004). It permits to calculate a constant annual air renovation rate, as a superposition of wind and stack effect. It depends on the n_{50} parameter and two tabulated exposure coefficients: the number of façades exposed to wind and the environmental exposure.

Figure 2 compares the four methods with the reference of PassivHaus considering three values of n_{50} (5, 7 and 10), obtained as typical values from experimental data in existing buildings. Analysing the results of the different methods, it is possible to observe that the K1, K2, K3 approach is not able to distinguish the building features with a high detail, in comparison with the other methods, due to the qualitative definition of the construction (tight, medium and loose). It means that this approach is not able to distinguish different levels of

loose construction, and the same coefficients have been used in the three cases, with no changes in the result. If the analysis is focused on both ELA methods (LBL infiltration model and Sherman Grimsrud approach), the average air change rates have a direct relationship with the different n_{50} values; however, the air changes rates obtained are the lowest of all methods. If both methods are compared with the PassivHaus reference, the air change rate is 0.1-0.3 h^{-1} lower, being this difference higher as the value of n_{50} increases. Finally, the EN15242 is analysed: the air change rate is also lower than the reference value, with a difference between 0.05-0.1 h^{-1}. In this case, the difference is lower in comparison with the other methods, and the relationship with the n_{50} and the air change rate is similar to the PassivHaus reference.
After this analysis, the selected method is the EN15242, due to two main reasons: the method allows to relate the air leakage of the household with experimental data (blower door test), and the results of the method are consistent with the reference of PassivHaus Design.

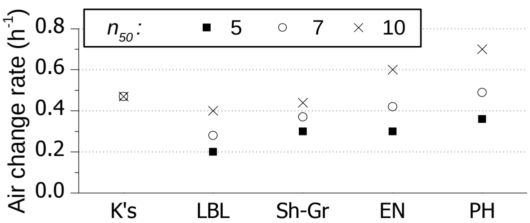


Figure 2: Comparison of the methods of air leakage's modelling (K's: K1, K2, K3; LBL: LBL infiltration; Sh-Gr: Sherman Grimsrud; EN: EN15242; PH: PassivHaus)

3.2 Implementation

The EN15242 method has been implemented in the building model to estimate the air change rate due to the air leakage of the building. In this study, the assumptions of n_{50} values are the following: 7.5 h^{-1} for existing building, and 5 h^{-1} for refurbished building. These values have been obtained from (Alfano et al., 2012; Jimenez et al., 2013), where they collected and analysed experimental data of air tightness from residential and Mediterranean buildings.

4. METHOD FOR VENTILATION

4.1 Selection of the method

The method for modelling the renovation rate due to natural ventilation depends on the building features and the type of ventilation. This can be: single sided ventilation, cross ventilation or stack effect due to courtyards. The references used to model each natural ventilation phenomenon and which are implemented in a dynamic simulation, are described in the paragraphs below.

1. Single sided ventilation, using (Gids and Phaff, 1982): this method calculates the air change rate in function of the opening dimensions, wind speed, indoor and outdoor temperature depending on wind and buoyancy effect.
2. Cross ventilation, using (British Standard, 1991): this method calculates the air change rate considering the thermal buoyancy effect and the wind effect, depending on the wind speed and the difference of indoor and outdoor temperature, in each moment. The method takes also into consideration the opening area, the height of the building and pressure coefficients.
3. Courtyard effect: in this case, the stack effect due to courtyard effect has been implemented in a simplified way, due to the complexity of the calculation. The courtyards are designed to extract the air from the households, due to the difference of temperatures between the outdoor and the courtyard. For that reason, the rule used to define the

courtyard effect is mainly related to the outdoor temperature (T_{out}) and the courtyard temperature (T_c), because depending on that difference, the direction of the air flow changes (from household to outside, or from outside to household). If the $T_c>T_{out}$, the air flow goes from household to outside and the effect is the desirable. On the contrary, if the $T_c<T_{out}$ the air flow is opposite and does not comply with the design. Usually, the courtyard ventilation is a complementary phenomenon from the main ventilation strategy: single sided or cross ventilation. For that reason, the air change rate is related to the main ventilation method of the household, and the temperature comparison defines if the courtyard ventilation is active or not. If the courtyard ventilation is active, then the rooms (zones) of the households that are influenced by the courtyard are ventilated.

4.2 Implementation

In the building typology that is analysed in this paper, the natural ventilation is single sided with the courtyard effect. For that reason, the Gids and Phaff method has been implemented.

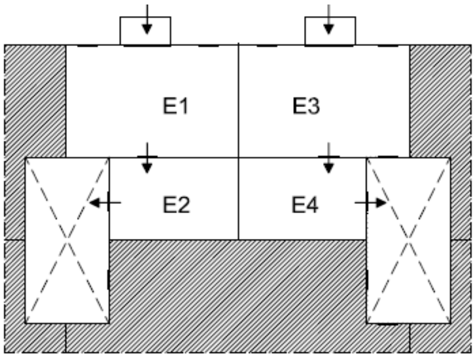


Figure 3: Natural ventilation strategy for a dwelling

Figure 3 represents how the natural ventilation has been implemented in the building model, depending on the zones' distribution. On the one hand, there are two external zones (E1 and E3), where the single sided ventilation method is implemented, assuming a windows' total area of 4 m^2 for each zone. On the other hand, the zones E2 and E4 are internal zones connected to the E1 and E3, respectively, and to the courtyard zones. In that case, the ventilation of both zones depends on the relation between the outdoor temperature and the courtyard temperature, as it has been introduced in the previous section. If the effect of the courtyard is positive ($T_c>T_{out}$, so that the air flow goes from household to outside), there is an air exchange between the external zone and the internal zone. In the other case, if the courtyard effect is negative ($T_c<T_{out}$), there is not air exchange between these zones. The air change rate between zones has been assumed to be the same than the single sided ventilation in each moment. Finally, the courtyard zone has been simulated as a ventilated zone and 4 air changes per hour have been assumed over all the year.

5. CONTROL STRATEGY

The natural ventilation is considered as the main strategy to reduce the temperature during the warm season, following vernacular behaviour in Mediterranean zones. The strategy is based on the following assumption: the users use the natural ventilation for cooling the household. In the case that the natural ventilation is not enough and overheating occurs, then, the windows are closed and the cooling system is switched on. This assumption is consistent with the results obtained in the surveys done in Catalonia during the MARIE project (www.marie-medstrategic.eu). The survey shows that 80% of the households use the natural ventilation for cooling and the 60% of households use the cooling system punctually, when the weather is very hot and consistent with adaptive comfort theory

In the building model, the infiltration is related mainly to the window perimeter. For that reason, although actually the infiltration is present all the time, in the building model the effect of the infiltration is active only when the natural ventilation (window opening) is not used.

The control of the natural ventilation depends on the following parameters: occupancy, operative temperature of the zone, courtyard temperature and outdoor temperature. Table 1 describes the control rules of the natural ventilation applied in the simulation model. In general terms and if there is occupancy in the household, the natural ventilation is *on* when the operative temperature is between 24ºC and 28ºC. This range of temperature is comfortable for ASHRAE adaptive comfort model (ASHRAE, 2004), especially when the outdoor temperature is higher than 20ºC (warm season). When the indoor temperature reaches the 28ºC, then the natural ventilation is off, and the cooling system turns on until the outdoor temperature is lower than the operative temperature, usually at night.

Table 1 Control strategy implemented in the building model

General rules of control	**Condition**	**Natural ventilation**
First condition:		
Occupancy	>0	YES
	0	NO
If the occupancy is >0		
Operative temperature (Top)	Top<24ºC	OFF
	24ºC>Top>28ºC	ON
	Top>28ºC	OFF
If the natural ventilation is OFF because Top>28ºC		
Outdoor temperature (Tout)	Tout>Top	OFF
	Tout<Top	ON
If there is a courtyard in the household and the natural ventilation is ON		
Courtyard temperature (Tc)	Tc>Tout	Courtyard effect ON
	Tc<Tout	Courtyard effect OFF

6. RESULTS

The results presented in this section are for the current building (base case) and of the building after a deep renovation with the implementation of passive measures (deep renovation). Table 2 shows the characteristics of the two building models.

Table 2 Characteristics of the buildings: base case and deep renovation

Characteristics	Base case (BC)	Deep renovation (DR)
Façade (U-value, W/m^2K)	With air chamber (1.22)	External insulation EPS 12cm (0.24)
Roof (U-value, W/m^2K)	With air chamber (1.17)	Internal insulation mineral wool 8cm (0.32)
Window (U-value, W/m^2K – g, -)	Aluminium without thermal break with clear double glazing 4/12/4 (5.68 – 0.85)	PVC with clear double glazing 4/16/4 (2.83 – 0.75)
Air leakage (h^{-1})	7.5	5
Solar protection	Internal blinds	Internal blinds
Heating system	Conventional NG boiler	Conventional NG boiler
Cooling system	Conventional AC Split (E1 & E3 zones)	Conventional AC Split (E1 & E3 zones)

The building model has been configured with the option to simulate the building with natural ventilation (NV) and without natural ventilation (nNV). The objective of this configuration is to be able to distinguish the buildings that have the possibility to do natural ventilation or not, due to its surrounding conditions and location (the possibility of ventilation is not the same in a spacious village than in a compact city).

6.1 Building behaviour and control strategy

The results presented in the Figure 4 show the results of the base case simulation with natural ventilation and without natural ventilation for a representative summer week (left and right side, respectively). The objective of these figures is to show the behaviour of the building model with the implementation of the infiltration and natural ventilation strategy described in the previous chapters.
The first row of figures shows the occupancy profile, which is the main driver of the use of the building. If there is occupancy, then the cooling systems or the natural ventilation are used. The second row of figures represents the outdoor temperature, the indoor temperature and the operative temperature. The behaviour of the indoor and operative temperatures follows the use of the building, reaching comfort conditions when the occupancy is in the household, thanks to the cooling strategies. The third row of figures shows the air change rate due to infiltration and natural ventilation, and the wind speed. In that case it is possible to see that in the simulation without natural ventilation, the infiltration is always active. For the simulation with natural ventilation, the natural ventilation is switched on, depending on the occupancy and the operative temperature, and following the control strategy presented before. Finally, the last row of graphs represents the cooling consumption, reflecting an important difference on the use of the cooling system between the simulation with natural ventilation and without natural ventilation.

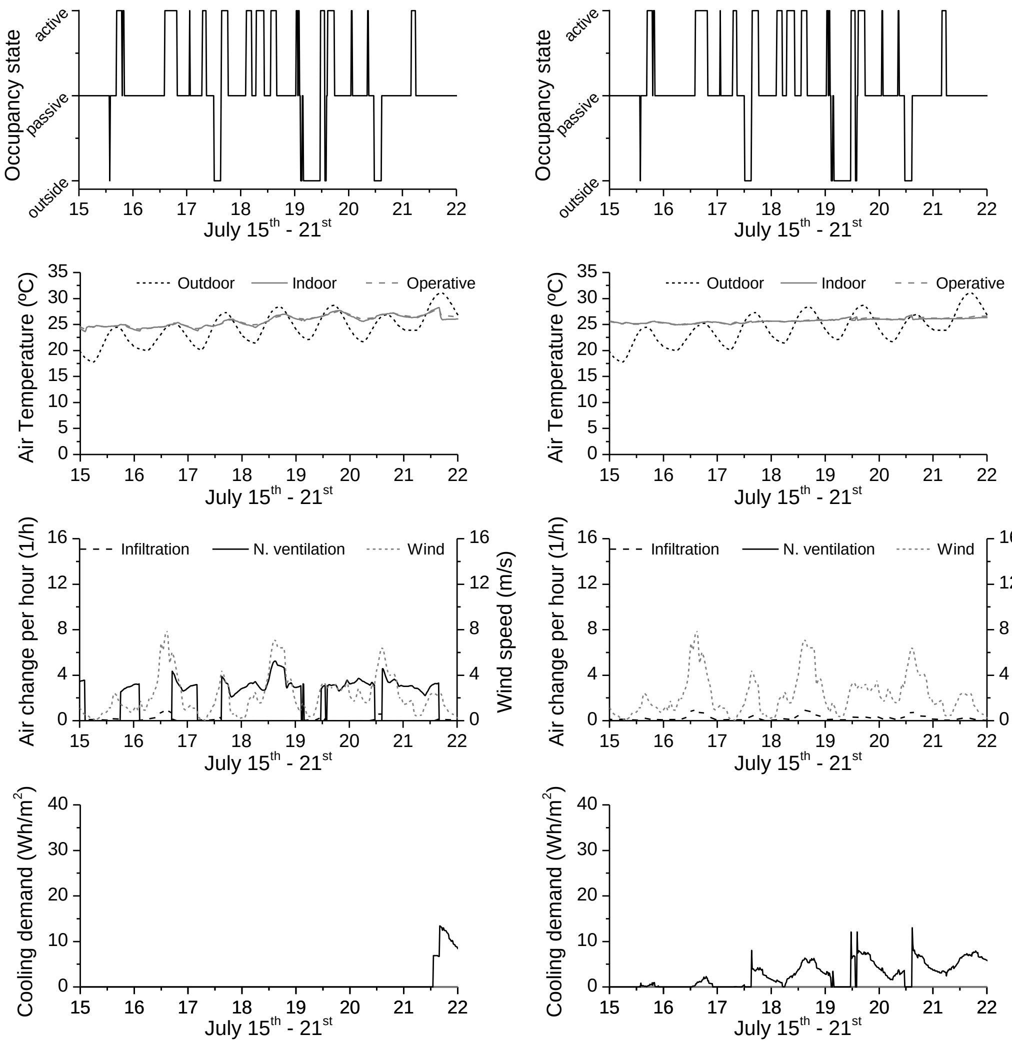


Figure 4: Results of the simulation for a summer with natural ventilation (left) and without natural ventilation (right).

6.2 Thermal comfort

For the comfort evaluation, the building has been simulated without the use of the heating and cooling system (free running mode) and the comfort model used is the ASHRAE adaptive model (ASHARE 55, 2004). The purpose is to explore to what extend the passive measures are able to improve the comfort conditions without the use of the mechanical systems. The comfort parameters used for the evaluation are the Long-term Percentage of Dissatisfied (LDP) developed by Carlucci (Carlucci, 2013) and the hours of overheating (OH).

The LDP is a long-term index that evaluates the comfort along a period. The index has been calculated for three periods (annual, cold season and warm season), in order to have information about the behaviour of the building under different weather conditions. The comfort requirement for a residential building is LDP < 20% (ASHARE 55, 2004). It means that the occupants have a comfortable condition at least during the 80% of the time. The calculation details of the LDP are explained in (Carlucci, 2013).

The hours of overheating (OH) are included in order to complement the LDP for the warm period. One of the main problems of the Mediterranean regions is the increase of the overheating hours due to a not appropriate design of the building. The analysis of this parameter can help to detect overheating problems and then, evaluate the possibility to avoid an active cooling system. The criterion used is that the percentage of OH has to be lower than the 1% of the warm season period in order to have a comfortable building. If the hours of OH are lower than 1%, it means that the building achieves comfortable conditions without the use of mechanical cooling system, and then it could be removed. The criterion was proposed by CIBSE (CIBSE, 2006), however, an adaptation in the calculation of the index has been done: the upper threshold is not a constant value and it depends on the ASHRAE adaptive comfort model.

$$P_{OH} = \frac{\sum_{t=1}^{T} p_t \cdot OH_t}{\sum_{t=1}^{T} p_t \cdot h_t} \quad \begin{cases} OH_t = 1 \Rightarrow T_{op,t} > T_{upperASH,t} \\ OH_t = 0 \Rightarrow T_{op,t} \le T_{upperASH,t} \end{cases} \tag{1}$$

Where P_{OH} is the Percentage of hour of overheating, $T_{op,t}$ is the operative temperature and $T_{upperASH,t}$ is the upper comfort temperature of the ASHRAE adaptive comfort model at time t. For the climate of Barcelona the 1% of the warm season hours corresponds to 41 hours.

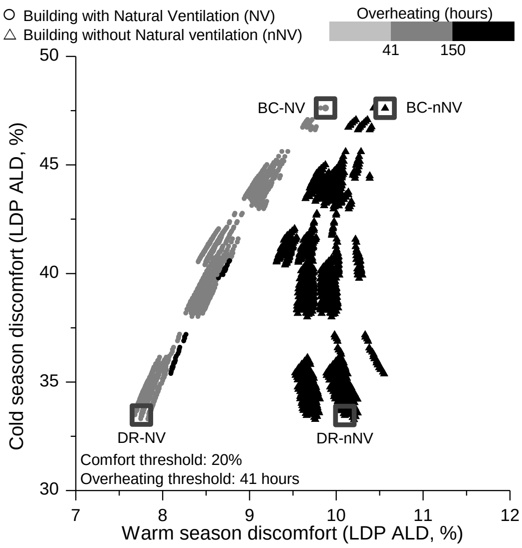


Figure 5: Comparison of the simulation with natural ventilation (circles) and without natural ventilation (triangles). Cold season comfort vs. warm season comfort. Colour scale: hours of overheating.

Figure 5 shows the results of the simulation in terms of thermal comfort. Each dot on the graph reflects a result of a simulation, where the building typology has been simulated with different combination of passive measures. The highlight simulations (square) are the base case with natural ventilation (BC-NV), the base case without natural ventilation (BC-nNV), the deep renovation with natural ventilation (DR-NV) and the deep renovation without natural ventilation (DR-nNV), which are analysed in the present paper. The x-axis represents the thermal discomfort during the warm period, and the y-axis the thermal discomfort during the cold period. The colour scale is the hours of overheating, revealing that for this building typology, all the simulations are suffering overheating (>41 hours).

An important difference between the set of simulation with natural ventilation (circle) and the set of simulation without natural ventilation (triangle) is observed. The first group of simulations follows a linear trend: when the thermal comfort of the cold period is improved, the comfort of the warm period is also improved. However, for the simulations without natural ventilation, this situation is opposite, especially when the thermal comfort of the cold period is better (deep renovation).

6.3 Energy demand

Table 3 shows the results of energy demand for heating and cooling, for the base case and for the deep renovation. As explained before, each case has been simulated with and without the possibility to use natural ventilation as a cooling strategy. A deep renovation of the building permits to reduce the heating demand around 65%. In the case of cooling demand, the deep renovation allows to reduce the cooling demand around 26%, in the case with natural ventilation. No significant changes are shown in the case without natural ventilation; however a slight tendency to increase the cooling demand can be sensed, as shown in the comfort results. Finally, comparing the same building with or without natural ventilation, the cooling demand can be reduce around 70-80%.

Table 3: Results of energy demand

Energy demand	**Base case (BC)**		**Deep renovation (DR)**	
(kWh/m^{2}yr)	**Heating**	**Cooling**	**Heating**	**Cooling**
With Natural Ventilation (NV)	60.9	1.1	21.5	0.8
Without Natural Ventilation (nNV)	60.9	3.9	21.5	4.0
Comparison (NV vs. nNV)	-	-72%	-	-80%

7. CONCLUSIONS

The paper presents a dwelling simulation developed with TRNSYS, where the estimation of the air thickness effect and the natural ventilation has been implemented through two detailed methods: EN15242 and Gids and Phaff, respectively. In addition, a control strategy of the natural ventilation has been developed in order to reproduce the actual behaviour of the occupants (based on surveys): the users use the natural ventilation as a main strategy for cooling the household. The occupancy profile used in the simulation has been obtained from the TUD analysis. Four different scenarios have been analysed and compared (with or without natural ventilation, existing building and renovated), in order to evaluate the effect of the natural ventilation, in terms of thermal comfort and energy demand.

The results show that the methods implemented in the dwelling simulations reproduce properly the effect of the natural ventilation and air infiltration. The analysis of the thermal comfort indices, LDP and OH, is consistent with the energy results, being useful and complementary information to evaluate the adaptation of different energy efficiency measures in each building typology.

Regarding to the building typology analysed, the results show that the natural ventilation allows to reduce the cooling demand around the 70-80% in comparison with the same building without natural ventilation. Despite of the positive effect of the natural ventilation, in that case, it is not possible to avoid the cooling system due to the dwelling reflects some overheating problems based on the current comfort models.

8. ACKNOWLEDGEMENTS

The research was supported by the MED Programme of the European Union under the MARIE strategic project (Agreement Nº 1S-MED10-002) and has been done with the collaboration of Universitat Politècnica de Catalunya (UPC).

9. REFERENCES

Alfano, F.R.d.A. Dell'Isola, M. Ficco, G. Tassini, F. (2012). *Experimental analysis of air tightness in Mediterranean buildings using the fan pressurization method*. Building and Environment, 53 (0) 16-25.

ASHRAE (1989). *Handbook of Fundamentals*

ASHRAE (2005). *Handbook of Fundamentals*

ASHRAE (2009). *Handbook of Fundamentals*

ASHRAE 55, ANSI/ASHRAE Standard 55-2004 (2004). *Thermal environmental conditions for human occupancy*. American Society of Heating Refrigerating and Air-Conditioning Engineers Inc., Atlanta.

British Standard, BS 5925:1991 (1991). *Code of practice for ventilation principles and designing for natural ventilation.*

Carlucci, S. (2013). *Thermal comfort assessment of buildings.* Ed.: Springer.

Chartered Institution of Buildings Services Engineers (CIBSE), (2006). *Guide A: Environmental Design*. London.

Bradley, D. (2005). *Type 932: Sherman Grimsrud Infiltration.* Utility Library Mathematical Reference. TESSLibs 17

EN 15316-2-1. (2008) *Heating systems in buildings. Method for calculation of system energy requirements and system efficiencies. Space heating emission systems.*

EN-ISO 13790 (2004). *Thermal performance of buildings - Calculation of energy use for space heating.*

Garrido-Soriano, N., Rosas-Casals, M., Ivancic, A., Álvarez-del Castillo, M.D. (2012). *Potential energy savings and economic impact of residential buildings under national and regional efficiency scenarios. A Catalan case study*. Energy and Buildings, 49 (0) 119-125.

Gids, W. Phaff, H. (1982). *Ventilation Rates and Energy Consumption due to Open Windows. A Brief Overview of Research in the Netherland.* 3rd IEA Air Infiltration Center Conference.

IDAE, Instituto para la Diversificación y Ahorro de la Energía. (2009). *CALENER-VYP. Viviendas y edificios terciarios pequeños y medianos. Factores de corrección de equipos.*

INE, National Statistics Institute. (2010). *Time Use Data Survey 2010.*

Jimenez Tiberio, A. Branchi, P. (2013). *A study of air leakage in residential buildings*. New Concepts in Smart Cities: Fostering Public and Private Alliances(SmartMILE), pp.1-4.

McDowell, T. (2006). *Type 960: Lawrence Berkeley National Labs Infiltration Model.* Utility Library Mathematical Reference. TESSLibs 17

NBE-CT-79 (1979). *Norma básica de la edificación sobre condiciones térmicas de los edificios.* Real Decreto 2429/1979, 6 julio 1979.

Thornton, J.W. (1998). *Type 571: Infiltration into a Conditioned Zone (K1, K2, K3 Method).* Utility Library Mathematical Reference. TESSLibs 17

TRNSYS 17.1 (2012), SEL. Solar Energy Laboratory. University of Wisconsin, Madison USA.

Salom, J. Ortiz, J. Russo, V. (2014). *Method to develop cost-effective studies of energy efficiency measures for Mediterranean residential existing buildings with multi-criteria optimization.* World Sustainable Building SB14, Barcelona, Spain.

UNE-EN 15242 (2007). *Ventilation for buildings. Calculation methods for the determination of air flow rates in buildings including infiltration.*

THERMAL COMFORT ASSESSMENT IN A SUSTAINABLE DESIGNED OFFICE BUILDING

Silvia Soutullo[*1], Mª Nuria Sanchez[1], Ricardo Enriquez[1], Mª Jose Jimenez[1] and Mª Rosario Heras[1]

1: Efficiency in Buildings Research Unit. Renewable Division of CIEMAT
Avenida Complutense 40. Madrid 28040, Spain
Corresponding Author: silvia.soutullo@ciemat.es

ABSTRACT

Thermal comfort improvement at the lowest energy consumption is a key issue when dealing with sustainability in buildings. An appropriate passive design is mandatory under those circumstances. Prior to construction, simulation tools help to make designs more sustainable. However, it is recognized a gap between real performance and the predicted one. This article presents the comfort methodology applied in an office building located in the north of Spain, characterized by a continental Mediterranean climate. The edifice is constructed under the principles of the sustainability optimizing its energy performance. The thermal balance between indoors and outdoors and user behaviour shows the proximity of each office to the comfort bands. Warmer or colder set point temperatures correspond to higher energy demands. The application of the Fanger methodology produces high percentages of neutral sensation and low values of people dissatisfied, especially during the wintertime. In this period, warmer sensations increase the percentages of people dissatisfied.

KEYWORDS

Sustainability, comfort sensation, experimental monitoring, Fanger method

1 INTRODUCTION

Building sector represents a high percentage of the final energy consumption in European southern countries (Pérez-Lombard et al., 2008), particularly in Spain this percentage supposes about 31% (Eurostat, 2013). In order to reduce this energy consumption as well as the greenhouse gases emissions to the atmosphere, more efficient buildings must be constructed. The first stage of this objective is to design almost zero energy buildings. These buildings use natural resources in a passive way, which leads to a decrease in their energy demands (ARFRISOL, 2008). The second stage is to optimize the conditioning system, including renewable energies systems, to minimize their energy consumptions (GhaffarianHoseini et al., 2013). These energy reductions must not compromise thermal comfort sensation and healthy conditions.
Thermal comfort is defined as the human sensation of heat and cold for each climate conditions (Givoni, 1992), so it is quite difficult to quantify due to the high subjectivity involved. Factors such as climate, activity, age, metabolic rates, expectations or adaptability have a strong influence on the total calculation.

This article presents the comfort methodology applied in an office building constructed under the principles of the sustainability in the north of Spain during the summer 2014 and the winter 2015. The methodology applied is described in different phases (Soutullo et al., 2014). The first one analyses the temperature oscillations between indoors and outdoors and its distribution along the time. These analyses indicate the temperature gap and the associated deviation, giving an idea of how many "hourly degrees" are needed to achieve a hypothetical heating or cooling demand based on specific set points. To quantify how comfortable are these records, different bands have been established depending on the season of the year.
The second phase evaluates quantitatively the comfort sensation and the level of satisfaction reached inside the offices. The Fanger methodology (Fanger, 1967), has been applied for this study. This Standard proposes an iterative methodology that considered thermal balance, clothing characteristics and type and level of activity to predict the thermal sensation and the degree of discomfort of the people inside the room.

2 CLIMATIC REPRESENTATIVENESS

The office building studied is located in the north of Spain in Valladolid, characterized by a cold semi-arid climate. The latitude of the building is 41.65 N, the longitude is 4.76 W and the altitude is 735m. Comparing with the representativeness meteorological year (TMY) of AEMET (AEMET, 2015), during the summer 2014 higher values of solar radiation, softer extreme temperatures and less percentage of humidity have been registered. In the winter 2015 higher values of solar radiation, lower extreme temperatures and similar percentage of humidity have been obtained in comparison with the TMY.

2.1 Givoni chart

Analyzing with a Givoni Chart (Givoni, 1992), the hourly combination of the ambient air temperature and the humidity ratio measured along the summer 2014 and the winter 2015, the most representative passive and active techniques have been highlighted. As it can be seen in Figure 1, during the summer period there are many recorders inside the comfort band (orange zone), and two success have been detected. On one side, low temperatures especially in June and the first hours of July and August, demanded minimal heating. This could be supplied by internal loads to reach the comfort zone. On the other side, solar shadings, evaporative process, ventilation and refrigeration produced by internal thermal mass are needed to achieve the comfort band. The wintertime is cold and needs the combination of passive and active heating with conventional heating to reach the comfort zone.

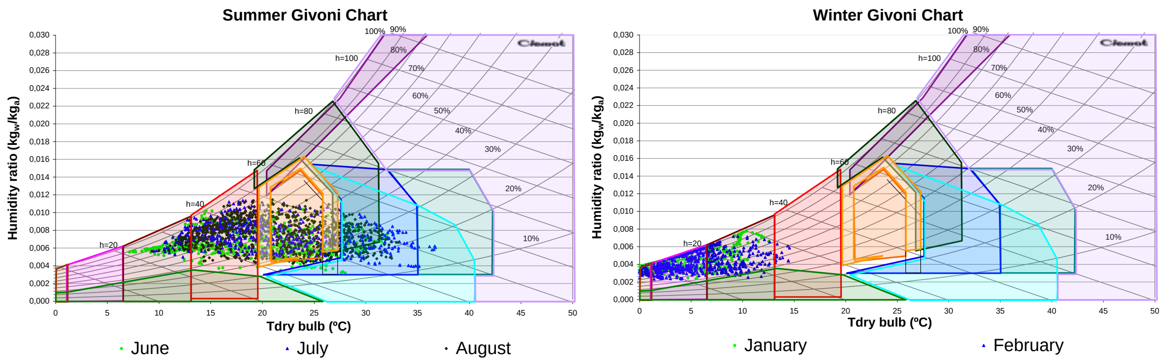


Figure 1: Summer and winter Givoni charts

2.2 Hourly cooling and heating degrees

The comfort set point temperatures for each band and the ambient measurements have been used to calculate the hourly heating and cooling degrees. When the ambient temperate is higher than the upper comfort limit, the difference between them is defined as cooling degrees. In the other side, if the ambient temperature is lower than the lowest comfort limit, the difference is defined as heating degrees. Applying this procedure, 9500ºC hourly heating degrees has been obtained during the winter 2015 with similar results in both months (January and February). This value points to cold winters. During summer 2014 and considering the ambient recorders, 934ºC hourly heating degrees and 2155ºC hourly cooling degrees have been obtained with warmer results in July and August. These values point to slightly hot summers.
As a relative humidity concerns, winter is a wet period with mean values that vary from 89% to 77% while summer is a temperate period with an average humidity of 51%.

3 BUILDING DESCRIPTION

The construction of this singular building has a usable floor area of approximately 1000 m^2 distributed across one floor. Main services developed in this building are administrative management, catering, gardening and retailing. The administrative area is divided at South orientation into closed offices.

3.1 Constructive information and systems description

Building materials used in this construction are green materials with low VOCs emissions. The edifice, constructed under bioclimatic criteria, is equipped with innovative active and passive systems. Different strategies were identified and integrated into the design of the building (Figure 2), optimizing its energy efficiency, improving the indoors thermal comfort and decreasing the CO_2 emissions. Most relevant strategies are: vegetation open areas under offices and a gardened roof, natural ventilation system composed by a grid system, and distributed lucernaires coupled to the air exchange system. Additionally, relevant spaces like a central atrium and a patio improve the thermal conditioning of surrounding and office areas, increasing natural light in winter and providing natural ventilation and solar protection to avoid overheating in summer.

Figure 2: Passive and active strategies. From left to right: gardened roof, lucernaire and central atrium

Moreover, façade design considers solar orientation. Southern façade is glazed to increase the direct solar gains within the offices, providing the use of natural light in winter. In summer, different shading devices and a ventilated façade reduce cooling loads (Figure 3). A solar absorption chiller provides pre-heated air to the air exchange system reducing thermal loads in summer.

Figure 3: From left to right: view of the South façade of the building and detail of solar absorption chiller

Energy loads are supplied by means of a combination of several renewable technologies, focusing on solar techniques. A 65 m^2 solar thermal system and an 87 KWp Photovoltaic system are installed on the roof. A vertical geothermal system with 77 KW cooling capacity and 101.5 KW direct heating system has been installed. Two biomass boilers using pellet fuels are installed: 244.5 KW for offices and 320 KW for industrial areas. A solar cooling system with a capacity of 4000 l and absorption refrigeration power of 264 KW has been installed.

3.2 Monitoring set-up

All relevant indoors and outdoors variables have been measured at an occupied building under real conditions. Monitoring is designed to evaluate the decrease in energy consumption and thermal comfort levels. The experimental campaign of this study has been done from June 2014 to February 2015. So the summer period comprises: June, July and August while the winter period includes: January and February. December must be excluded due to problems in the electrical fed of the acquisition system.

A 16-bit A/D resolution data acquisition system is used and modules distribution minimise wiring. Data are sampled every one second and one minute averages are recorded although other recording intervals are available. Time of all data provided will be in local time: this is Central European Time with daylight saving, i.e. in summer (UTC +2).

The location of all sensors will be done taken into account the special characteristics of the building in order to correctly evaluate the different representative areas. Across central areas from 3 to 6, closed office spaces are equally distributed. The two first offices are 12.5m^2 while the east one is 17.6m^2 (Figure 4).

Specific offices had been selected based on the occupancy level of the office, the characteristics of its construction and the implemented systems. Three representative offices have been selected for this study: Zn3_DC, Zn3_DO and Zn5_DE, belonging to central, west and east types respectively. The experimental measurements available in these monitored offices are: air temperature, relative humidity and CO_2 concentrations. Cooling and heating system performance is also monitoring.

Furthermore, outside boundary conditions are measured in a meteorological station installed on the roof of the building. South horizontal solar radiation, longwave radiation, air temperature, relative humidity, wind speed and direction and CO_2 concentration are the variables measured.

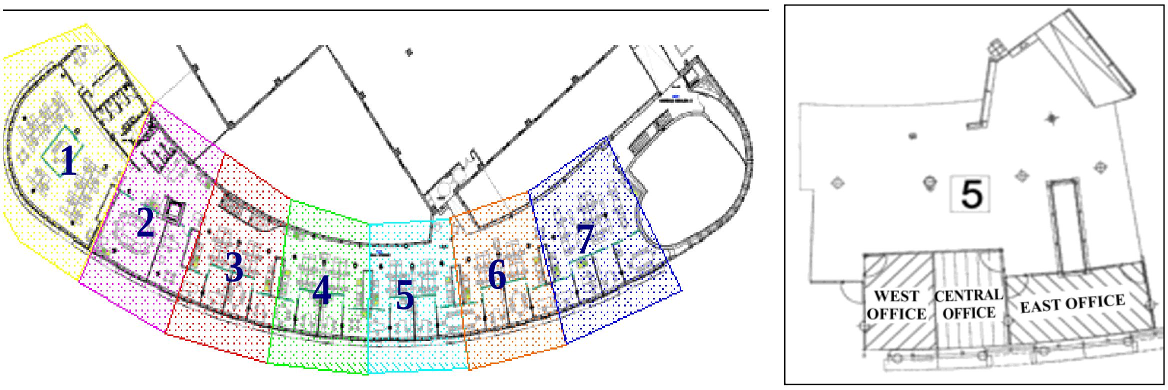


Figure 4: From left to right: Monitoring project design into seven areas, and detail of the office distribution on central areas.

4 COMFORT EVALUATION

The thermal balance reached inside the three offices studied has been evaluated by means of the temperature oscillations between indoor and outdoors as well as their comfort evaluation (Soutullo et al., 2014; Jimenez et al.,2013) To quantify how comfortable are these results during the whole year, two bands have been selected for each season. During summer period the comfort band is 24ºC ± 1ºC and during the winter period the comfort band is 22ºC ± 1ºC.

The energy demands of each office have been measured along the two periods, giving different values as a result of the user behaviour. In both periods heating and cooling demands are supplied. During the summer 2014, the cooling demands measured are: 19 kWh/m^2 in Zn3_DO, 12.7 kWh/m^2 in Zn3_DC and 0.3 kWh/m^2 in Zn5_DE, while the heating demands are always less than 1 kWh/m^2. During the winter months, the heating demands obtained are: 2.7 kWh/m^2 in Zn3_DO, 5 kWh/m^2 in Zn3_DC and 22.6 kWh/m^2 in Zn5_DE, while the cooling demands are always less than 1 kWh/m^2.

Taking into account that all the offices are faced to South and have similar use, the warmest behaviour is always registered in Zn5_DE while the coolest behaviour is registered in Zn3_DO.

4.1 Thermal characterization of building

Temperature oscillations produced by the climatic perturbations measured during the year 2014, allow the thermal characterization of the three studied rooms. The gap between these indoor temperatures and the comfort bands gives an idea of how efficient is this building. These offices have been performed from Monday to Friday, considering the occupational timetable from 7 am to 3 pm.

Figures 5 and 6 represent the frequency reached by the hourly mean temperature during the summer 2014 and the winter 2015 respectively in the three studied offices.

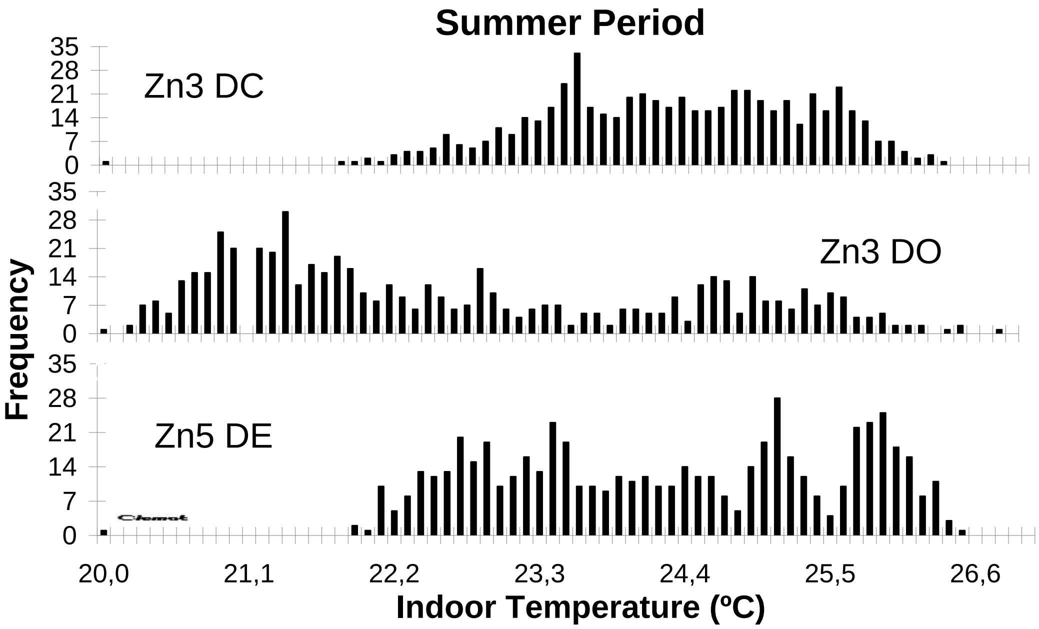


Figure 5: Indoor temperature histogram registered in the three studied offices during the summer 2014

During the summertime (Figure 5), the most frequent temperatures are: 23.6ºC in Zn3_DC, 21.4ºC in Zn3_DO, 25.1ºC in Zn5_DE. Considering the three levels established by the summer comfort band: <23ºC, 23º-25ºC and >25ºC, different percentages of occurrence have been obtained by each office and each level during these summer months (Table 1).

Table 1: Percentages of occurrence with the summer comfort temperatures

Office	Tª<23ºC (%)	23ºC<Tª<25ºC (%)	Tª>25ºC (%)
Zn3_DC	8	64	27
Zn3_DO	61	25	14
Zn5_DE	20	45	35

As it can be seen in Figure 6, the most frequent temperatures registered during winter are: 23.5ºC in Zn3_DC, 21.5ºC in Zn3_DO and 24.9ºC in Zn5_DE.

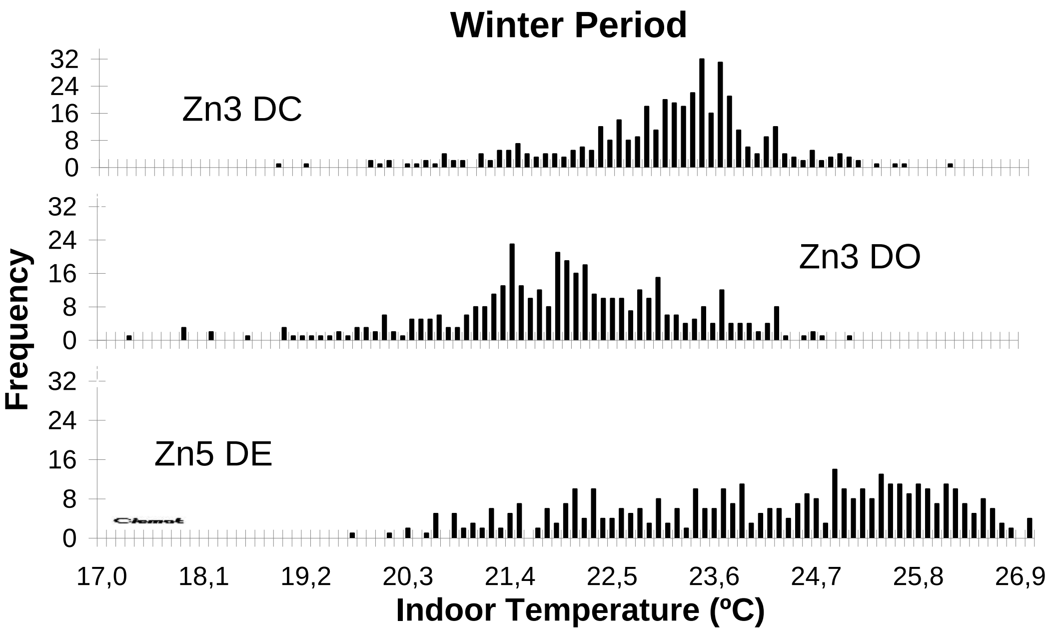


Figure 6: Indoor temperature histogram registered in the three studied offices during the winter 2015

Taking into account the three levels established by the winter comfort band: <21ºC, 21º-23ºC and >23ºC, different percentages of occurrence have been obtained by each office during these winter months (Table 2).

Table 2: Percentages of occurrence with the winter comfort temperatures

Office	Tª<21ºC (%)	21ºC<Tª<23ºC (%)	Tª>23ºC (%)
Zn3_DC	5	33	62
Zn3_DO	15	62	22
Zn5_DE	4	25	71

4.2 Heating or cooling degrees approach

Temperature differences between indoors and outdoors can represent the dynamic process that occurs inside each office. This balance shows the thermal gap distribution produced during a period of time. Both comfort bands (one for winter and one for summer) have been marked in these charts to quantify the results.
The limit comfort band temperatures have also been used to quantify the hourly effective heating and cooling degrees inside each office (method described in 2.2), giving an idea of how is the thermal deviation from the comfort scale established.
Figure 7 represents the hourly temperature drops measured inside Zn3_DC (red points), Zn3_DO (green points) and Zn5_DE (grey points) versus the ambient temperature during the summer 2014 and the winter 2015.

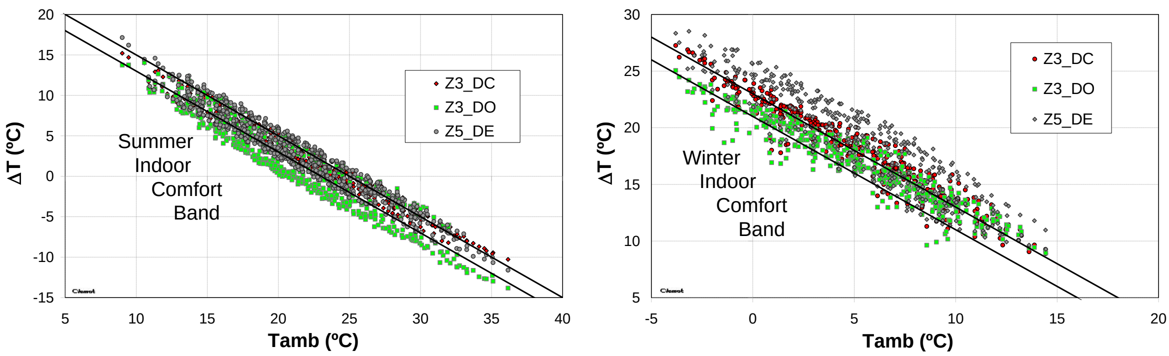


Figure 7: Thermal oscillation obtained inside the three studied offices during the summer 2014 and winter 2015

As it can be seen during the summer months the thermal dispersion from the comfort band has registered different behaviours, reaching the highest value in Zn3_DO (76%) and the lowest value in Zn3_DC (37%). In this period, the hourly heating degrees obtained in these offices are 51ºC in Zn3_DC, 571ºC in Zn3_DO and 87ºC in Zn5_DE while the hourly cooling degrees are 70ºC in Zn3_DC, 42ºC in Zn3_DO and 123ºC in Zn5_DE. The office Zn3_DO has been established lower set point temperature so higher energy demand is needed to reach this thermal condition.
During the winter months, the studied offices have reached high dispersion percentages from the comfort band, with the lowest value registered in Zn3_DO (40%) and the highest percentage in Zn5_DE (76%). The hourly heating degrees achieved in these offices are 13ºC in Zn3_DC, 67ºC in Zn3_DO and 8ºC in Zn5_DE while the hourly cooling degrees are 192ºC in Zn3_DC, 67ºC in Zn3_DO and 594C in Zn5_DE. These results show a warmest behaviour in Zn5_DE due to the highest set point temperature established by the user. This leads to an increase of the energy demand of this office as described in previous paragraphs.

4.3 Fanger thermal quantification

There are many processes that can quantify the indoor thermal comfort but the methodology recommended by the Spanish Building Code (Government of Spain, 2014) assumes the steady state conditions propose by the Fanger method, described by the International Standard ISO 7730 (ISO 7730, 1994).This Standard proposes a method to predict the thermal sensation and the degree of discomfort taking into account all the variables that influence the thermal exchanges between people and the environment. The comfort equation proposed by Fanger is a function of several parameters: meteorological variables, clothing characteristics and type and level of activity. As a result, two different indices can be obtained: the Predicted Mean Vote (PMV) and the Predicted Percentage of Dissatisfied people (PPD). The first index predicts the hypothetical mean value of the thermal sensation obtained in the energy balance equation. This level varies between -3 and 3 and represents a thermal comfort scale from hot to cold environments. The centre of this 7-point thermal sensation scale is the comfort band that corresponds to zero energy balance or thermal equilibrium. The second index predicts the percentage of people dissatisfied with the environment conditions and varies between 5 and 100%.

Applying this iterative methodology to the office rooms studied, the PMV and PPD indices have been calculated for summer and winter period. Ambient conditions, low values of wind speed, sedentary activity and seasonal dress code have been considered as input values.

Figures 8 and 9 show the PMV values reached inside the three rooms during the summer 2014 and the winter 2015 respectively. In both figures a comfort band has been marked in grey to highlight the neutral sensation.

During the summer working hours the thermal sensation profiles achieved inside the three offices vary as a function of the user behaviour. The neutral sensation has reached percentages of 81.2% in Zn3_DC, 38% in Zn3_DO and 70.4% in Zn5_DE. To complete the thermal balance, slightly cool environments have been registered with percentages of 18.8% in Zn3_DC, 62% in Zn3_DO, 29.6% in Zn5_DE.

In wintertime both offices inside Zn3 have been achieved very high percentages of neutral comfort sensation, with values of about 94%. Slightly cool and slightly warm environments have also been obtained, with percentages that vary from 0.3 to 5.7% in Zn3_DC and 3.9 to 1.5% in Zn3_DO, respectively. However the comfort profiles registered inside the office Zn5_DE are different, with warmer values obtained during the hours of maximum solar irradiance. This effect is traduced to lower neutral sensation percentages (75.8%) and higher slightly warm percentages (24.2%).

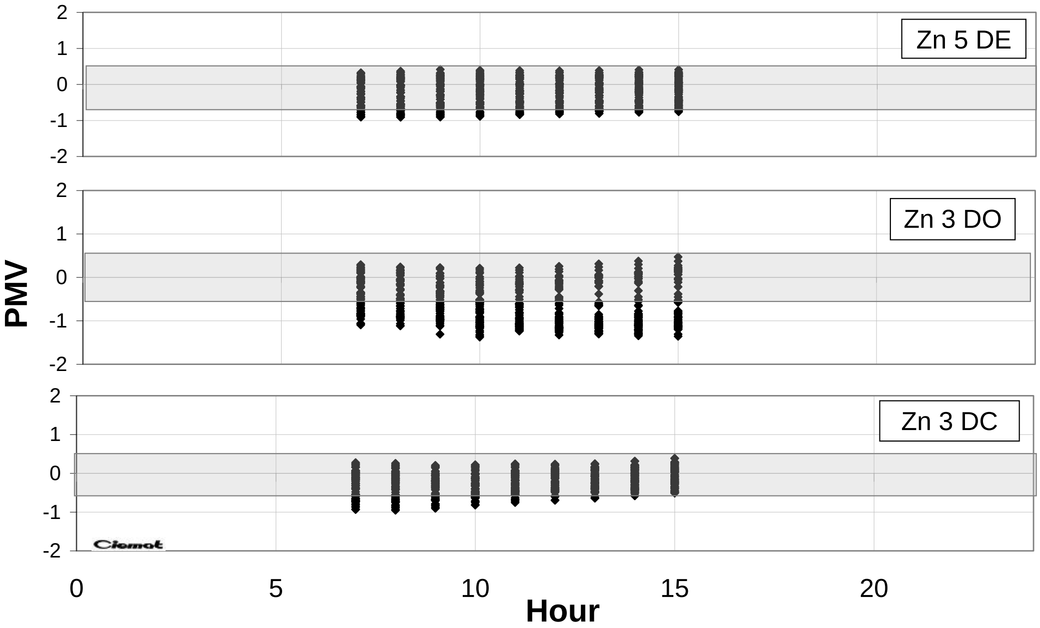


Figure 8: PMV index reached inside the three analyzed offices during the summer 2014

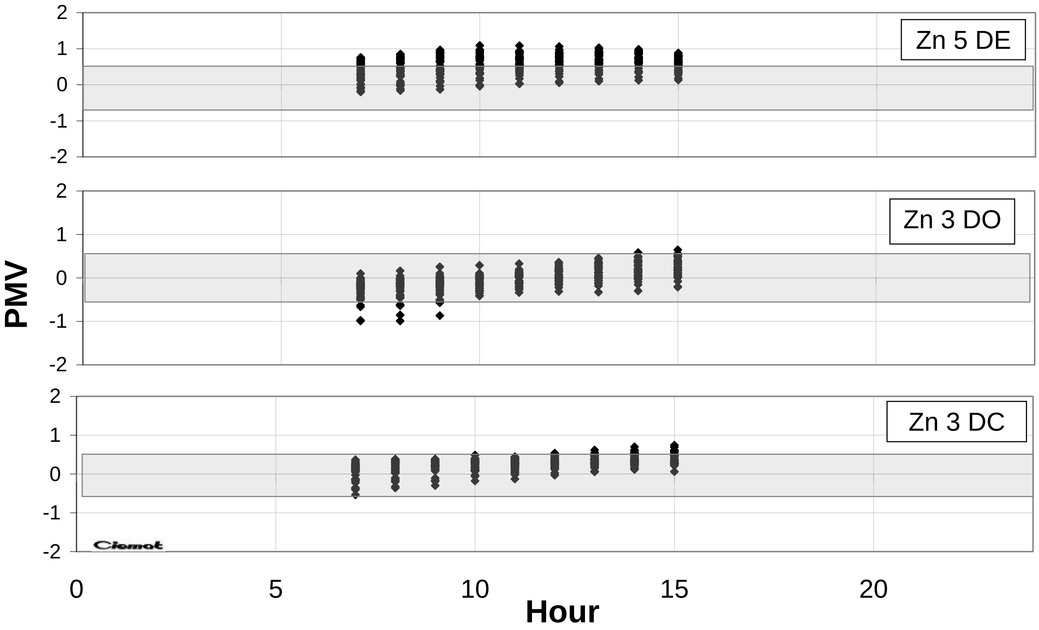


Figure 9: PMV index reached inside the three analyzed offices during the winter 2015

These thermal scales can be traduced in percentage of people dissatisfied with the environment, showing in the following figures as the PPD index (Figures 10 and 11).
As it can be seen, during the summertime the average profiles of people dissatisfied with the environment are similar inside Zn3_DC and Zn5_DE (mean value of 8.4%), growing to 18% inside the office Zn3_DO. The maximum percentages obtained oscillate between 24.2% in Zn3_DC, 44.2% in Zn3_DO and 22.5% in Zn5_DE.
In winter period the users are more dissatisfied in the office Zn5_DE and more comfortable inside both offices Zn3, with mean percentages that vary from 13.3 to 6.8% respectively. The maximum values registered oscillate between 30% in Zn5_DE, 25.7% in Zn3_DO and 16.5% in Zn3_DC.

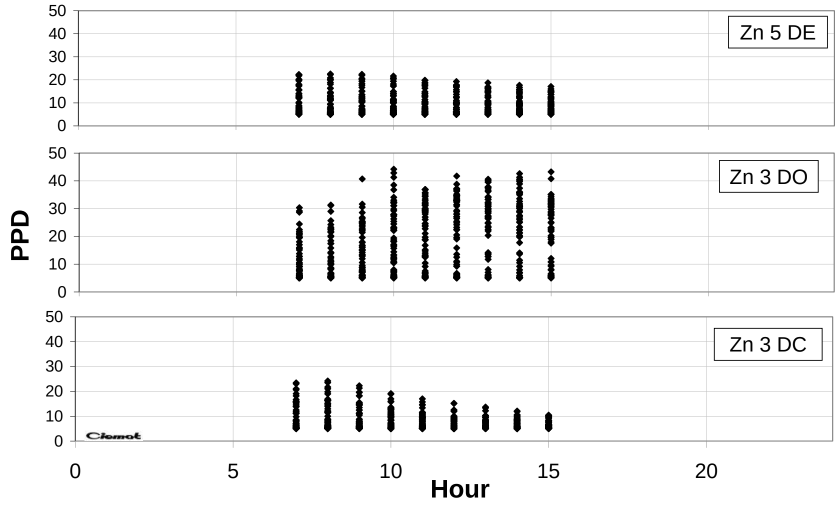


Figure 10: PPD index reached inside the three analyzed offices during the summer 2014

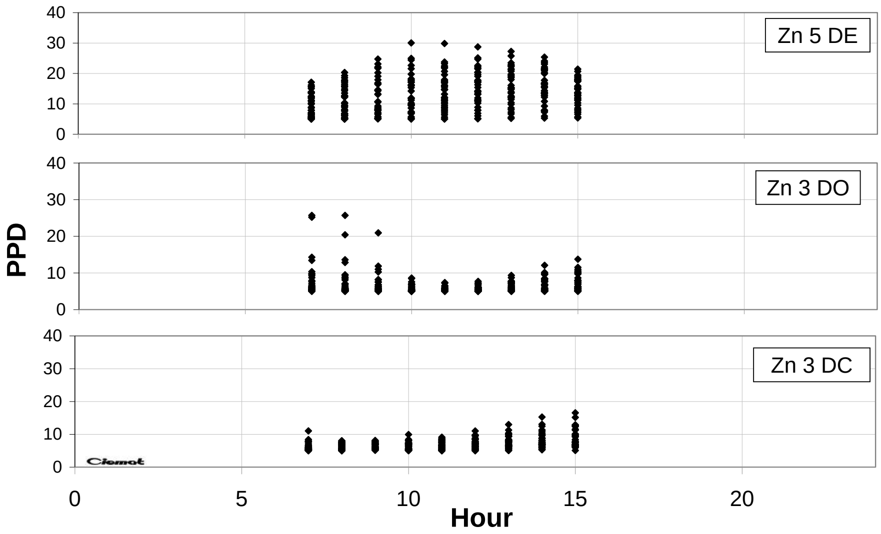


Figure 11: PPD index reached inside the three analyzed offices during the winter 2015

5 CONCLUSIONS

To evaluate the temperature profiles along the year, three office rooms have been monitored during the summer 2014 and winter 2015. The quantification of the thermal environment has been considering with two comfort bands: 23º-25ºC in summer and 21º-23ºC in winter. Taking into account these temperatures as set points, the percentages of occurrences within these bands have been calculated. Summer results show high frequency in Zn3_CD (64%), medium in Zn5_DE (45%) and low in Zn3_DO (25%). In winter the frequencies switch this trend to reach the highest value in Zn3_DO (62%) and low values in Zn3_DC and Zn5_DE.
Hourly heating and cooling degrees have been calculated considering the limits of these comfort references. These values point to low effective cooling degrees during the summer months in all the offices, but Zn3_DO needs medium effective heating degrees to achieve the comfort band. These results are corroborated by the energy measured which indicates that this office demanded more energy than the others. During the winter period low heating degrees are needed by all the offices but medium cooling degrees are demanded by Zn5_DE to reach the comfort band. This trend is corroborated by the energy measured in this office.

These results show different thermal profiles in each office depending on the user behaviour. Warmer or colder set point temperatures than the comfort bands correspond to higher energy demands.
Applying the Fanger methodology, high percentages of neutral sensation have been obtained in Zn3_DC and Zn5_DE (average 75%) decreasing to low percentages in Zn3_DO (38%) during the summer period. The PPD index shows that the colder behaviour of Zn3_DO increase the percentage of people dissatisfied (18%).
During the wintertime very high percentages of neutral sensation have been obtained in both Zn3 offices (higher than 90%) with low percentages of people dissatisfied (lower than 7.5%). In these months, the office Zn5_DE have been achieved warmer thermal sensations increasing and higher percentages of people dissatisfied with this environment (13.3%).

6 ACKNOWLEDGEMENTS

This work has been developed in the framework of the OMEGA-CM programme. The OMEGA-CM programme, reference P2013/MAE2835, is a multidisciplinary R&D programme supported by the Madrid Regional Government approved within the call “Tecnologías 2013”, co-financed by EU Structural Funds. The authors wish to express their gratitude for this support in the OMEGA-CM programme.

7 REFERENCES

Pérez-Lombard, L.; Ortiz, J.; Pout, C. (2008). A review on buildings energy consumption information. *Energy and Buildings,40, 394-398.*
Eurostat. (2013). Final energy consumption by sector. http://ec.europa.eu/eurostat (Access June 2015).
Givoni, B. (1992). Comfort, climate analysis and building design guidelines. *Energy and Buildings, 18, 11-23.*
PSE-ARFRISOL. (2008). Proyecto Científico-Tecnológico Singular de carácter Estratégico Arquitectura Bioclimática y Frío Solar. www.arfrisol.es (Access June 2015).
GhaffarianHoseini, A.; Dahlan, N.D.; Berardi, U.; GhaffarianHoseini, A.; Makaremi, N.; GhaffarianHoseini, M. (2013). Sustainable energy performances of green buildings: a review of current theories, implementations and challenges. *Renew Sustain Energy Rev, 25, 1–17.*
Soutullo, S.; Enríquez, R.; Jímenez, M.J.; Heras, M.R. (2014). Thermal comfort evaluation in a mechanically ventilated office building located in a continental climate. *Energy and Buildings,*81*, 424-429.*
Fanger, P.O. (1967). Calculation of thermal comfort: Introduction of a basic comfort equation. *ASHRAE Transactions, 73(2), 1-20.*
Agencia Estatal de Meteorología,AEMET (2015).www.aemet.es (Access June 2015).
Jiménez, M.J.; Enríquez, R.; San Isidro, M.J.; Soutullo, S.; Heras, M.R. (2013). Desarrollo de metodologías para la evaluación energética experimental de edificios a escala real: contribución del PSE-ARFRISOL. *Revista Española de Física,27(2), 1-16.*
Government of Spain. (2014). *RITE: Reglamento de Instalaciones Térmicas de Edificios, Spanish Regulation of thermal installations in buildings.* Spanish Official Bulletin (BOE) 207.
International Organization for Standardization, Standard 7730 (1994). *Moderate Thermal Environments. Determination of the PMV and PPD indices and specification of the conditions for thermal comfort*. ISO Documents Genève, Switzerland.

IMPROVEMENT OF COMFORT AND ENERGY EFFICIENCY IN EXISTING BUILDINGS USING ADAPTIVE THERMAL COMFORT ALGORITHM

José Manuel Salmerón Lissén[*1], José Sánchez Ramos[2], Pablo Aparicio Ruiz[3], Javier García Ramos[1], Servando Álvarez Domínguez[1], José Luis Molina Félix[1], Elena Barbadilla Martín[3]

1 Grupo de Termotecnia. Seville University
Camino de los descubrimientos S/N
Escuela Técnica Superior de Ingenieros
Seville 41092, Spain
**Corresponding author: jms@us.es*

2 Cádiz University
C/Chile 11002
Cádiz, Spain

3 Grupo de Investigación Operativa. Seville University
Camino de los descubrimientos S/N
Escuela Técnica Superior de Ingenieros
Seville 41092, Spain

ABSTRACT

Comfort and energy saving are two important concepts treated in current buildings in order to maintain a good air quality reducing the energy consumption. According to International Energy Agency (IEA) buildings represent 32% of total final energy consumption, and the need for reduction of CO_2 emission leads to pay attention to the energy demand in buildings. On the other hand maintaining a good-quality environment helps to improve the productivity and effectiveness of workers. Thus thermal comfort models that optimize the consumption of energy guaranteeing the comfort of occupants are gaining importance in the building sector.
This is what adaptive thermal comfort theory pursues, being the aim of this paper to develop an adaptive thermal comfort algorithm for air conditioned buildings located in Seville. Two different buildings have been chosen to perform the experiments, two operator's room with about 15 people which are surveyed every day to carry out the field studies. Sensors have been placed around the room to measure the environmental parameters.
The main aspects treated in this paper are:

1. Instrumentation required to measure the environmental parameters and its location in the room are introduced,
2. The transversal and longitudinal questionnaires that are necessary to perform the field studies.
3. Description of the characterization method that is going to be used in order to evaluate the energy savings.

The experimental phase lasts until the end of August 2017.

KEYWORDS

Comfort, energy efficiency, Ventilation, Adaptative thermal comfort

1 INTRODUCTION

This paper is based on the works carried out in the ME4CA project. The objective of the project is to develop an adaptive thermal comfort algorithm (ACA) that guarantee comfort conditions leads to energy savings benefits and keeps an adequate indoor air quality.

In most of the existing buildings, the indoor air temperature is maintained between a relative narrow temperature limits –in practice, 22 ± 2^{o}C-. International standards as EN ISO 7730 [1] and ASHRAE 55-92 [2] suggest the calculation of the comfort temperature as a function of the clothing levels and the methabolic rate of the occupants. Additionally, due to the need of reduce the CO2 emmissions the energy consumption of buildings have been tackled and both criteria have been considered together.
At the nearly 70s, Nicol and Humphreys developed an introduction to the adaptive comfort theory, where they show as the occupants could tolerate indoor conditions out of those recommended by the standards. Humphreys suggested that the comfort temperature could be related with the outdoor temperature in a given location, this is shown in the next figure and leads to what is known nowadays as ACA.

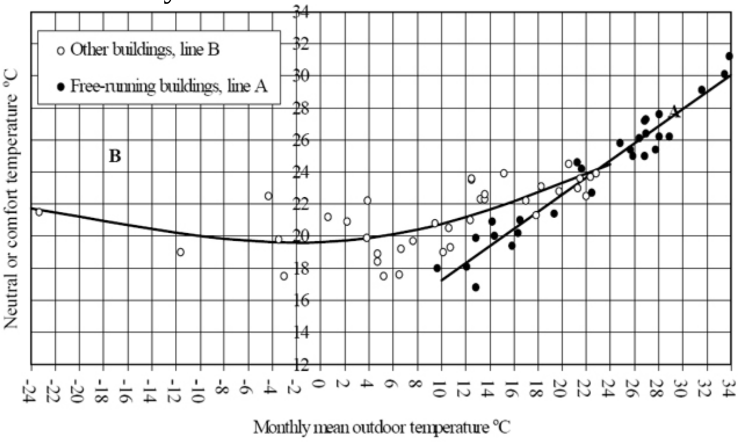


Figure 1: Comfort temperature as a function of the monthly mean outdoor temperature. Source Nicol, Humphreys and Nicol [8].

In order to evaluate the energy savings due to the use of the new comfort techniques it is necessary to use a thermal characterization method that could forecast the energy consumption of buildings.
In the first section of the present paper we describe the variables that are required to develop the ACA and the characterization method, in the second section we describe the survey used to recover the information from the occupants; and finally in the third section we describe the thermal characterization methodology.

2 MEASUREMENTS OF ENVIRONMENTAL PARAMETERS

The studied building is a call center used for emergency purposes. There are air inlets on the floor symmetrically distributed, and air outlets on the floor and ceiling. The reason why we selected this building is that there are enough people to perform the field study, working during the night and changing the seats frequently. Figure 2 shows this room:

Figure 2: Call center selected to carry out the field study

The parameters that are going to be registered are the next:

- Air temperature (ºC)
- Globe temperature (ºC)
- Relative humidity (%)
- Air velocity (m/s)
- CO_2 concentration (ppm)
- Illuminance on the work plane (lux)

These are the general parameters that have also been measured in the SCATs project "Project controls and thermal comfort" [8]. CFD simulations have been developed in order to determine the required number of sensor and their location in the room.

2.1 Location of the air velocity sensors

The whole room is split in 5 sections. The sections 1 to 5 are represented in the next figure for the velocity field. According to the figure 3, there should be at least one sensor in each row, but keeping in mind that the workers are concentrated in the first 4 rows (section 1 and 2) and the price of the multidirectional velocity sensor is high, only one sensor has been placed at head level in the centre of the occupied area (section 2 in figure 3).

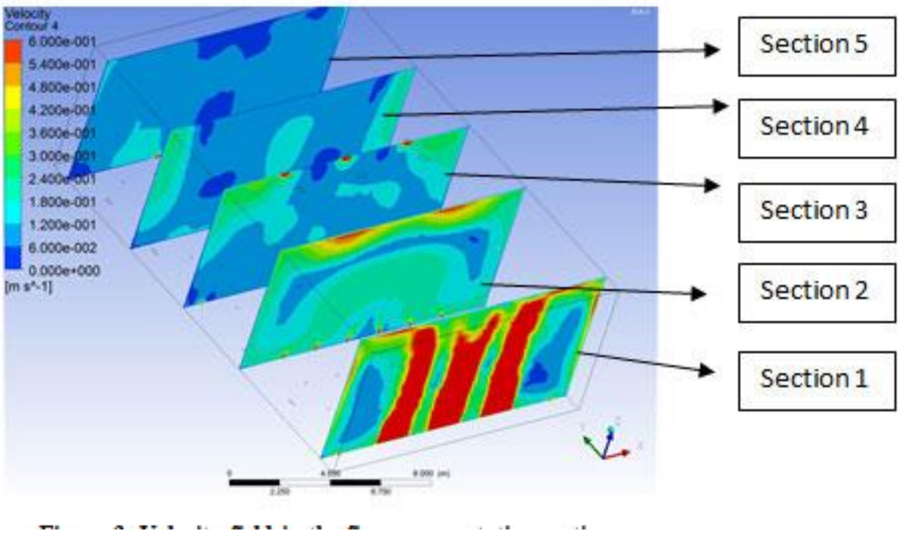


Figure 3: Velocity field in the five representative sections

2.2 Location of the air temperature sensors

As mentioned above, the whole room is split in 5 sections. The sections 1 to 5 are represented in the next figure for the temperature field. According to the figure 4, there should be at least three sensor in each row, but keeping in mind that the workers are concentrated in the first 4 rows (section 1 and 2), only three sensors has been placed at head level in the first 3 rows.

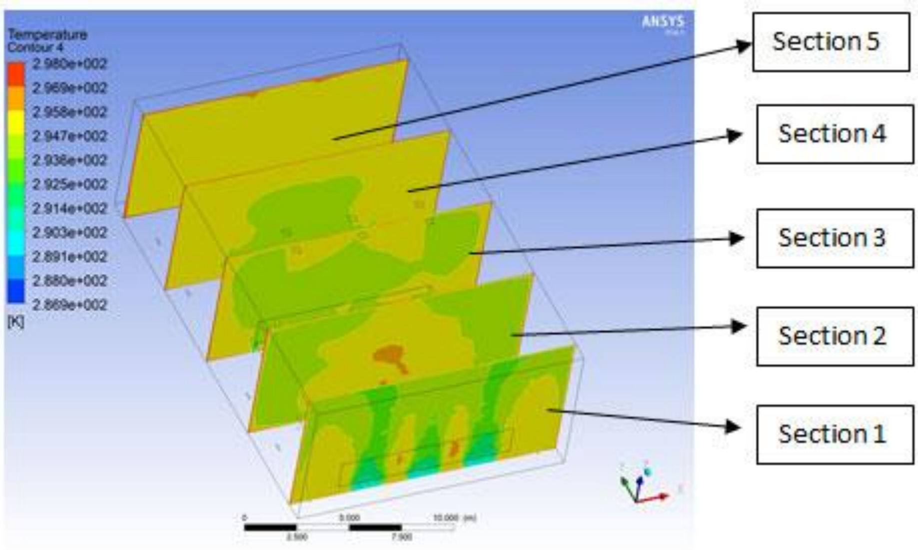


Figure 4: Temperature fields in the five representative sections

Note how the temperature profiles are symmetric in each room, but the occupants are not located symmetrically, so it cannot be used this property when deciding where to place the sensor.

2.3 Location of the CO_2 sensors

Qualitatively, the CO_2 profiles are similar to the air temperature profiles, thus the figure 4 can be taken into account to select the place where to locate the sensors. Since it is required the maximum value of CO_2 concentration, 1 sensor should be installed in each section of the occupied area.

2.4 Monitoring system

Among the sensors mentioned above, it has been located luxometers, relative humidity, globe temperature and surface temperature sensors. The number of sensors and their position are listed below:

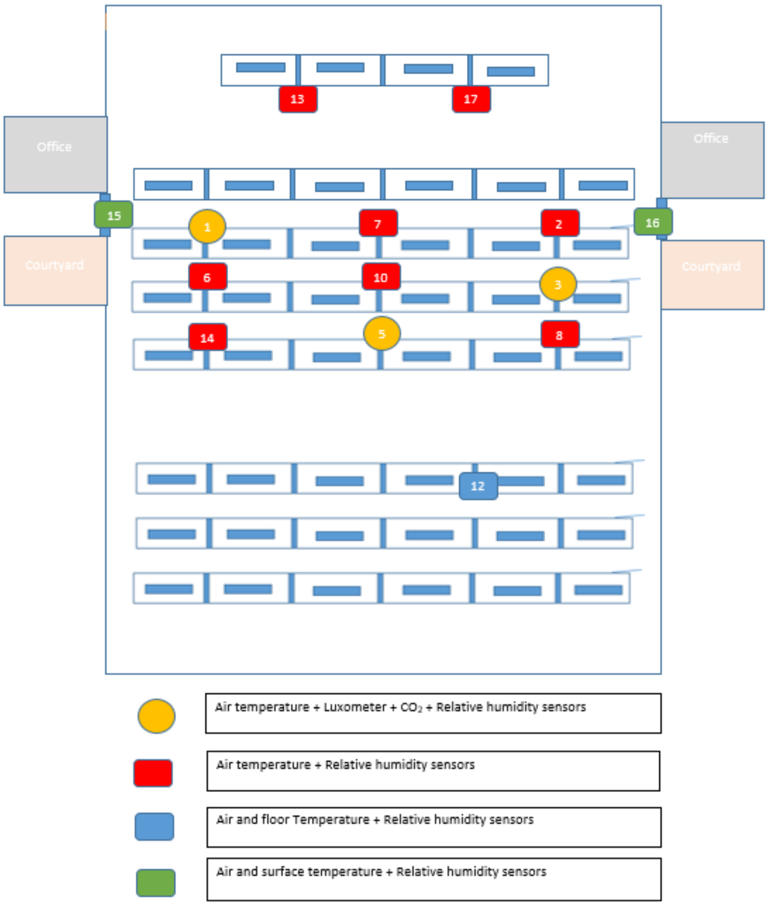


Figure 4: Location of each sensor of the monitoring system

3 THERMAL COMFORT EVALUATION THROUGH QUESTIONNAIRES

It has been shown that there is often an acute discrepancy between the comfort objective and subjective comfort [3]. The survey as a fundamental element of subjective data collection is presented by ASHRAE thermal comfort as a means to study the comfort [4]. The sampling of the users comfort, allows observing and predicting the level of comfort on the HVAC system. The development of online sampling systems assists in the recognition of the behavior patterns that occur in the offices [5].

The survey allows knowing the different perceptions of the users about indoors comfort levels. The system stores information relating to the comfort related to the temperature and humidity at the moment when the survey data are collected. It is selected the international standard ISO 10551:1995 [6] which looks at the ergonomics of the thermal environment as a basis employing subjective judgment scales. Even so, in spite of the requests of those surveyed, all systems must be limited to certain norms whether determined by the ergonomics expert or by the laws or regulations of a country.

The experiment look at such things as the difference between individuals and the effect of time series on comfort. The problem with this method is period of study, because if the survey period is completed in a short time, say a day then the variety of environmental conditions you investigate with any group of subjects is generally small, since the variation of temperature in a single day may be limited. The method need to study the reactions of a different sample to each value of temperature.

In general, this studies take weeks to complete the enough result to study the adaptive model. The comfort characteristics alter its character over time and space. So whilst the conditions

may vary, you are in effect measuring the user with each temperature. For this reason, this project works through a long-term experiment. This is an experimental procedure that runs through a long period of time, although the number of subjects in this group is not strictly limited, only a relatively small sample will be possible with a view to maintaining continuity in the survey answers.
An interesting example was presented by Humphreys and Nicol (1970) [7] who carried out a wide-ranging survey based on monthly records collected from a small group of subjects over a period of 15 months. In 2010, the SCATs project were collected 4655 sets of environmental and subjective data, and 1449 of these were collected in buildings which were free-running at the time of the measurements [8]. Many surveys have been conducted using subjects in a tropical context and relate observed sensation on the ASHRAE scale (or the similar Bedford scale) to the physical environment. Examples of authors are: Webb, Sharma and Ali, Busch, Matthews, Nicol, Taki, Bouden and many others are quoted in the literature e.g. (Nicol, 2004) [9]. They found that subjects differed from themselves from month to month by as much as they did from each other. In this way a relatively small group can be representative of a much larger population.
This sample of users chosen are familiar with their surroundings and the climate they are living in and the experiment allow them to go about their usual routine and use their own clothes.
In our case the particular group of individuals for the study are workers in a big office. The individuals in the sample are willing to devote the effort needed for taking part in the survey and their Trades Union was contacted for the approval.
To make a sample representative of the population, the project select user with different sex, age and body dimensions and they are registered in the system.
The survey is divided in three questionnaires:

1. Clothing questionnaire
2. Transversal questionnaire
3. Longitudinal questionnaire

The clothing survey, this questionnaire is filled every week and include different clothes aspects like:

- Q: Shoes (A: Sandals, Shoes, Boots)
- Q: Sock (A: Thin socks, Nylon stockings, Thick socks or none)
- Q: Pants (A: Shorts, Trousers, Corduroys or none)
- Q: Dresses – Skirts (A: Summer skirt, Skirt winter, Light Dress, Winter dress or none)
- Q: Shirts - Blouses - T-shirt (A: Short sleeves, Long sleeved, Long sleeve winter or none)
- Q: Vest – Sweater (A: Sleeveless, Lightweight, Thick, Thick gooseneck or none)
- Q: Jacket (A: Summer Jacket, Winter Jacket, Coat, anorak or none).

The transversal questionnaire is filled every week, but three days after the clothing survey. This questionnaire only has five questions.

- Q: How would you describe the quality of the natural lighting in your work area? (A: Clearly acceptable, Acceptable, Unacceptable, Clearly unacceptable)
- Q: How would you describe the quality of the artificial lighting in your work area? (A: Clearly acceptable, Acceptable, Unacceptable, Clearly unacceptable)
- Q: How do you find the air quality right now? (A: Clearly acceptable, Acceptable, Unacceptable, Clearly unacceptable)
- Q: How strong do you find the smell to be right now? (A: No smell, Weak smell, Moderate smell, Strong smell, Very strong smell, Overwhelming smell)

- Q: At the moment you are... (A: Happy, Optimistic, Excited, Normal, Uninterested, Depressed, Sad)

The longitudinal questionnaire is filled every day and contain 8 questions:

- Q: How do you feel just now? (A: Cool , Slightly cool, Neutral, Slightly warm, Warm)
- Q: How do you find the temperature right now? (A: Acceptable, Unacceptable)
- Q: How do you want the temperature to be? (A: Higher, Slightly higher, Unchanged, Slightly lower, Lower)
- Q: The activity that you are carrying out... (A: Is stressful, Is normal, Is relaxing)
- Q: Have you eaten or drunk anything in the last 30 minutes? (A: Nothing, I have drunk something or I have eaten and drunk something)
- Q: You are wearing at this moment a shirt, blouse or t-shirt ... (A: Short sleeves, Long sleeved, Long sleeve winter or none)
- Q: You are wearing at this moment a vest or sweater... (A: Sleeveless, Lightweight, Thick, Thick gooseneck or none)
- Q: You are wearing at this moment... (A: Summer Jacket, Winter Jacket, Coat, anorak or none)

The personal thermal condition (perceptual and emotional evaluation and temperature preferences) is incorporated as in the UNE-EN ISO 10551:2002, the thermal environment (personal acceptance and tolerance) is included in the longitudinal questionnaire and about the emotional state, the level of stress appear in the longitudinal questionnaire and the worker's mood in the transversal survey.

A Web application has been made under an Apache server and developed in PHP language, the data is stored in a MySQL database. The answers to the questionnaires are stored on the web-server. PMV (Predicted Mean Vote) is calculated in real time after the user answers the survey, the value CLO (Clothing insulation) is based on the responses to the questionnaire, the PMV is calculated in general (with the clothing survey) or a second CLO changes the clothing survey with the last three questions in the longitudinal questionnaire. All the process is carried out with the survey and sensors data through other program, this program is used to calculate the body mass index, or other important and necessary information.

4 CHARACTERIZATION METHOD FOR THE EVALUATION OF THE ENERGY SAVINGS

To develop a method to characterize the energy savings is necessary in order to evaluate the benefit of the implementation of the adaptive comfort algorithm (ACA) due to the fact that we cannot compare simultaneously the building with and without this kind of control. It is necessary to carry out a base line period without the ACA, and then characterize the consumption in order to compare the data with those obtained after the implementation of the ACA.

The characterization model described below is based on transfer functions [10], [11], so it requires an instantaneous relation between input data and output, but not at other way. Transfer functions are used in many applications to evaluate dynamics variables; because, this models can be too easy but very efficiency.

The main objective of the model is defining tool for characterizing energy consumption of existing building. In addition, the main requirements for the model are:

- Identify their coefficients through experimental data
- Adaptable to different time steps: small (hour) or big (days).
- Enough accuracy to be a tool to evaluated energy savings.

So, mathematical and basic formulation results:

$$f(t) = \sum_{i=1}^{m} \sum_{j=0}^{n} a_{ij} \; Y_i(t-j) - \sum_{k=1}^{d} d_k \; f(t-j) \quad (1)$$

Where:

- f(t) objetive variable
- Y_i excitations (i=1,..,m)
- a_{ij} coefficients for each independent variable Y_i
- d_k coefficients of inertia effect.

Equation (1) is a general formulation of the model that it is possible to specify for the application studied. For this one, it is necessary to know:

- Temperature of outdoor air [ºC]
- Global radiation [W/m^2]
- Temperature of indoor air [ºC]
- Schedule of operation
- Energy consumption of HVAC.

Hence the basic model for characterize sum energy consumption of HVAC for term z is:

$$CI(Z) = \sum_{i=0}^{m} a_i \cdot T_{SA}(z-i) + \sum_{i=1}^{m} b_i \cdot T_{INT}(z-i) + \sum_{i=1}^{n} d_i \cdot CI(z-i) \quad (2)$$

It should be noted that, CI(Z) has a double correlation, because the model needs temperature of indoor air but this temperatures is depends on energy consumption of HVAC.
Therefore, it can do another model for correlation average temperature of indoor air. This new model has two formulation: (3) when HVAC is turned off and indoor air is in free-floating, and (4) when HVAC is on, average indoor air temperature is the objective variable and CI(Z) is a input of the model.

$$T_{INT}(Z) = \sum_{i=0}^{m} e_i \cdot T_{SA}(z-i) + \sum_{i=1}^{n} d_i \cdot T_{INT}(z-i) \quad (3)$$

$$T_{INT}(Z) = \sum_{i=0}^{m} g_i \cdot T_{SA}(z-i) + \sum_{i=0}^{m} h_i \cdot CI(z-i) + \sum_{i=1}^{n} d_i \cdot T_{INT}(z-i) \quad (4)$$

Where:

- $T_{INT}(Z)$: Average temperature of indoor air [ºC].
- $GI(z)$: average of internal gains for term z [W/m^2]
- $T_{SA}(z)$: Equivalent temperature sun-air.
- a_i, b_i, e_i, g_i and h_i : coefficients of the model
- d_i: Coefficients depends on the time constants of the system. ASHRAE [12]–[14] and UNE 13790 [15].

The double model, energy consumption vs temperature of indoor air can be know

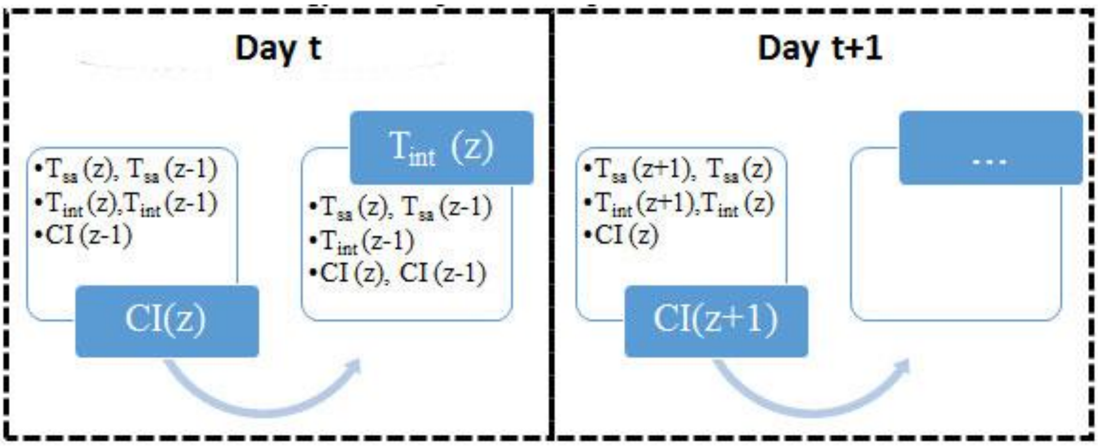


Figure. Simplified diagram of double model CI(z)-T_{INT}(z).

Equivalent temperature sun-air is combination of global radiation [Wh/m^2] and temperature of outdoor air [ºC]. For calculating, it is necessary to make an experiment in free-floating of the buildings (without internal gains: people, lights… and with HVAC turns off). The model of indoor temperature in this conditions results:

$$T_{INT}(Z) = \sum_{i=0}^{m} aa_i \cdot T_{EXT}(z-i) + \sum_{i=o}^{m} bb_i \cdot RAD(z-i) + \sum_{i=1}^{n} d_i \cdot T_{INT}(z-i) \quad (5)$$

Where: $\mathrm{T_{EXT}}(z)$ is an average temperature of outdoor air for term z, and $RAD(z)$ is the sum of global radiation incident. So, temperature sun-air is defined as:

$$T_{SA}(z) = \mathrm{T_{EXT}}(z) + k \cdot RAD(z) \quad (6)$$

Where k [ºC/Wh/m^2] is obtained in steady state of expression (5).

$$k = \frac{\sum_{i=o}^{m} bb_i}{1-\sum_{i=1}^{n} d_i} \quad (7)$$

This parameter k refers increment of indoor temperature due to effect of radiation.
Similar study, it is feasible to do with internal gains. For it, it has to do an experiment in normal operation of the building but HVAC has turn off. In this case, the model of average temperature of indoor air results:

$$T_{INT}(Z) = \sum_{i=0}^{m} cc_i \cdot T_{SA}(z-i) + \sum_{i=o}^{m} gg_i \cdot GI(z-i) + \sum_{i=1}^{n} d_i \cdot T_{INT}(z-i) \quad (8)$$

Therefore, steady-state parameter associated to internal gains is:

$$P_{GI} = \frac{\sum_{i=o}^{m} gg_i}{1-\sum_{i=1}^{n} d_i} / (V \cdot \rho \cdot C_P) \quad (9)$$

This parameter refers to increment of indoor temperature due to effect of internal gains.
Finally, it has developed a double model to characterize energy building consumption and average temperature of indoor air. The combination of these models allows solve the main objective and the final application in the project.

5 FUTURE DEVELOPMENTS

We are currently measuring the parameters inside the room to be able to characterize the consumption before implementing the algorithm, and the questionnaires is being filling in by occupants to evaluate the comfort. Thus this summer and winter we will be collecting measures to be able to create the thermal comfort algorithm. Next summer and winter, this algorithm will be implemented and evaluated to determine the improvements in comfort and energy consumption.

REFERENCES

[1] EN ISO 7730 (2006). *Ergonomics of the thermal environment - Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria.*

[2] ASHRAE 55-2013. *Thermal Environmental Conditions for Human Occupancy*

[3] Meir, I.A., Garb, Y., Jiao, D., Cicelsky, A. (2009). Post Occupancy Evaluation: An Inevitable Step Toward Sustainability. *Advances in Building Energy Research*, 3, 189-220.

[4] ASHRAE (1997). *Ashrae handbook-fundamentals*. American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta.

[5] Aparicio, P., Guadix, J., Onieva, L., & Escudero, A. (2014). Methodology for analysis and decision making by sampling in buildings. *Dirección y Organización*, (53), 65-74.

[6] ISO 10551 (1995). Ergonomics of the thermal environment - Assessment of the influence of the thermal environment using subjective judgement scales. International Organization for Standardization, Geneva.

[7] Humphreys, M. A., Nicol, J. F. (1970). An investigation into thermal comfort of office workers. *Journal of the Institution of Heating and Ventilating Engineers*, 38, 181-189.

[8] Nicol, F., Humphreys, M. (2010). Derivation of the adaptive equations for thermal comfort in free-running buildings in European standard EN15251. *Building and Environment*, 45 (1), 11-17.

[9] Nicol, F. (2004). Adaptive thermal comfort standards in the hot–humid tropics, *Energy and Buildings*, 36 (7), 628-637.

[10] D. G. Stephenson and G. P. Mitalas, "CALCULATION OF HEAT CONDUCTION TRANSFER FUNCTIONS FOR MULTI-LAYER SLABS.," ASHRAE Trans., vol. 77, no. Pt 2, pp. 117–126, 1971.

[11] E. Ponsoda, S. Blanes, and P. Bader, "New efficient numerical methods to describe the heat transfer in a solid medium," Math. Comput. Model., vol. 54, no. 7–8, pp. 1858–1862, 2011.

[12] T. J. R. Cooper, "1986 ASHRAE handbook: Refrigeration systems and applications: ASHRAE, Inc., USA 1986, ISSN 82 64 3223, US$100," Int. J. Refrig., vol. 10, no. 4, p. 243, 1987.

[13] "96/00806 1995, ASHRAE Handbook, HVAC applications, Si edition," Fuel Energy Abstr., vol. 37, no. 1, p. 49, 1996.

[14] ASHRAE, "HANDBOOK FUNDAMENTALS 2009." 2009.

[15] AENOR, "Eficiencia energética de los edificios. Consumo global de energía y definición de las evaluaciones energéticas," UNE-EN 15603. 2008.

ELECTROTHERMAL ACTUATORS WITH PWM CONTROL

Kamil Szkarłat, PhD. Eng.
Zakład Informatyki Stosowanej, Wydział Fizyki, Uniwersytet Adama Mickiewicza w Poznaniu [Applied Information Technology Institute, Department of Physics, Adam Mickiewicz University in Poznań]

Andrzej Górka, PhD. Eng., Radosław Górzeński, PhD. Eng., Michał Szymański, PhD. Eng.
Instytut Inżynierii Środowiska, Wydział Budownictwa i Inżynierii Środowiska, Politechnika Poznańska [Environmental Engineering Institute, Department of Building and Environmental Engineering, Technical University in Poznań]

Keywords*:*

electrothermal actuators, PWM-control

Abstract

In recent years, as an alternative to continuous control with the use of standard analog automation signals (voltage or current), the Pulse Width Modulation (PWM) control was introduced. Although, it is often considered as the equivalent of a continuous control, in practice this continuous control strategy is not feasible with the use of simple electrothermal actuators. The paper presents the investigation results of selected electrothermal actuators operation under ON-OFF and PWM control. It shows clearly that the pulse width modulation for electrothermal actuators results in discontinuous control, which is only a slight modification of the two-point control method.

1. Introduction

The automation structure plays a significant role in improving the performance of a heating, ventilation and air conditioning systems. Very often, even if properly designed, they don't meet the requirements of the design intent. Among others, the reasons can include lack of integrated design between HVAC and control systems, aiming for reduction of investment costs in building automation, application of substitute solutions, however not equivalent. Sometimes problems arise not only at a control level but also at the physical one. The result of separation of system-derived aspects and automation is that industries mutually assume idealized work standards which are not kept in reality. Designers expect ideal control, without regulation control overshoot and time delays. Programmers working also as automation experts rarely use feedback - assuming instead, that the behaviour of actuators is proportional to the control signal.

Figure no. 1 shows a basic configuration of the automatic control system. This is a closed control system loop with a negative feedback automatically ensuring the required change of a controlled variable. For the HVAC industry, a valve and its drive often function as an actuator and a room with HVAC terminal elements (heater, chilled beam, fan coil) usually function as an object - inertial element of various orders.

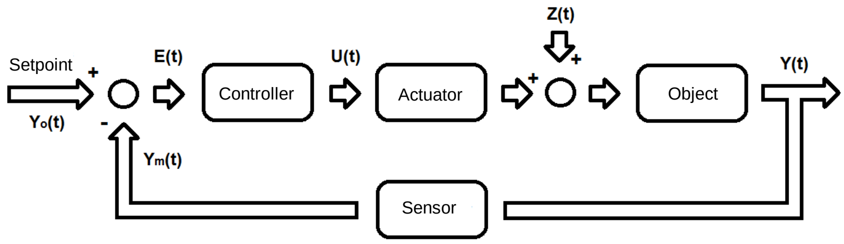


Fig. 1. The basic configuration of the automatic control system

In recent years PWM control is implemented as an alternative for continuous control, with the use of standard analog automation systems: voltage 0÷10V DC or current 4÷20 mA. Then, electrothermal actuators are very often used as driver elements. This method allows for reduction of investment costs of automation systems by reducing the number of wires, replacement of analog modules with digital ones in control cabinets as well as reduction of actuator costs. Although this method is very often treated as an equivalent of continuous control, in practice, due to very strong non-linearity, it is most often a functional equivalent of the two-point control for electrothermal actuators (usually used as part of the driver element in the HVAC industry).

2. PWM control

PWM is a control method of a voltage or current signal with a constant amplitude and frequency consisting in a change of pulse width modulation signal. The most important parameter of PWM signal is a duty cycle described by the formula no. 1:

$$k_w = \frac{T_{ON}}{T} \tag{1}$$

where:
k_W – duty cycle,
T_{ON} – high state period
T – total period of the signal described by the formula no. 2:

$$T = T_{ON} + T_{OFF} \tag{2}$$

where additionally:
T_{OFF} – low state period [2].

Signal interpretation with impulse modulation is presented in Fig. No. 2.

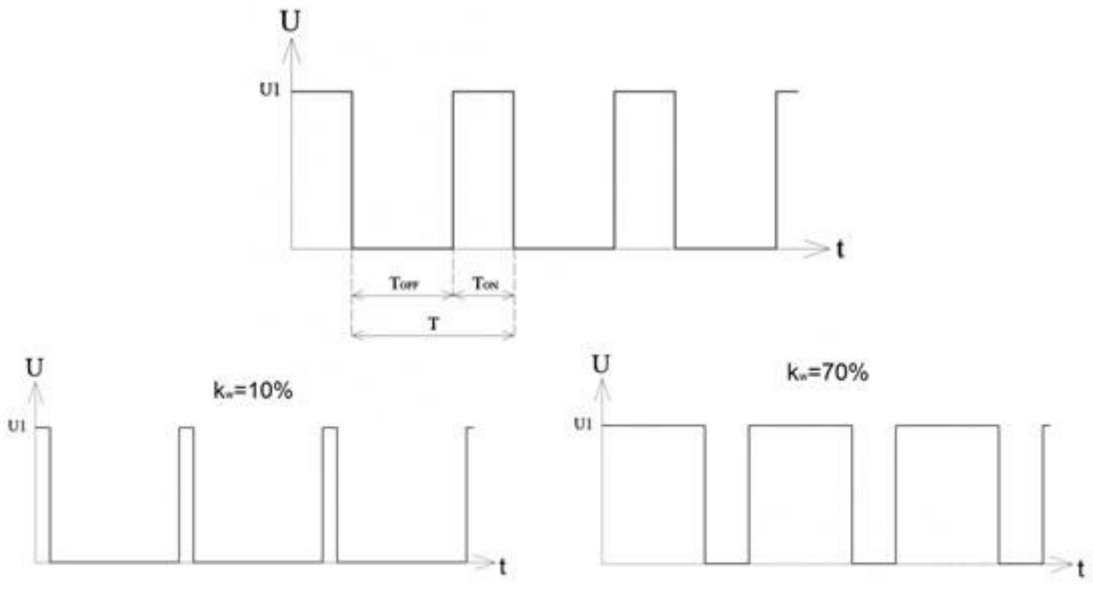


Fig. 2. Graphical interpretation of the pulse modulated signal

In non-continuous PWM control time T_{ON} is proportional to control signal value. Theoretically, it is then assumed that with the use of electrothermal actuators the valve can also reach intermediate opening stages. Opening/closing time of electrothermal actuators (usually several minutes) is long relatively to *T* period (the period of the signal is much shorter than the time constant of the actuator). One can theoretically assume that the average voltage of the signal causes intermediate valve opening states [1]. Unfortunately, practical solutions of electrothermal actuators and using them together with PWM control does not make it possible to reach the assumed goals.

3. Tests accomplished

3.1. Electrothermal actuators

In order to carry out tests, several representative electrothermal actuators from several manufacturers have been chosen (Danfoss, Oventrop, Salus, Siemens, TA Hydronics), which differ in supply voltage (24V AC, 230V AC) and in the location when inactive (NC – normally closed, NO – normally open).

3.2. Test and laboratory set-up

For the purpose of planned testing, a special dedicated test and laboratory set-up has been prepared. Figure No. 3 shows part of the set-up.

Fig. 3. Experimental set-up for testing electrothermal actuators

The laboratory set-up has been made to consist of the following elements: the element of the electrothermal actuator under test connected to a precise potentiometer (LCP–50FC–1k), with a software installed including a dedicated PWM algorithm of a PLC actuator (Siemens S7-1200), a data acquisition device (Ahlborn 2590) and regulated power supply.

3.3. Tests results

Measurements have been carried out in several series for two types of control modes: two-point and PWM. All measurements have been conducted at ambient temperature of 22°C.

Two-point control – by providing power supply to the actuator or by disconnecting it from the power supply in a predefined time, descriptions of opening/closing degree have been read as a function of time with concurrent setting of opening time and closing time counted until reaching 95 percent of the set value. Figure No. 4 shows the test results for ON/OFF control (opening degree of the actuator – SO as a function of time – t).Table no. 1 includes main parameters for tested actuators. Labels A-E are assigned randomly to the tested actuators.

PWM control – using properly programmed PLC emitting a pulse modulated signal at regular time intervals, opening/closing degree of the actuator have been read as a function of a duty cycle k_w (different level of a duty cycle). Figure No. 5 shows the test results for PWM control (opening degree of the actuator – SO as a function of a duty cycle k_w). Table No. 2 includes the opening degrees of a given actuator depending on a duty cycle k_w during opening stroke (O) and closing stroke (Z) of actuators.

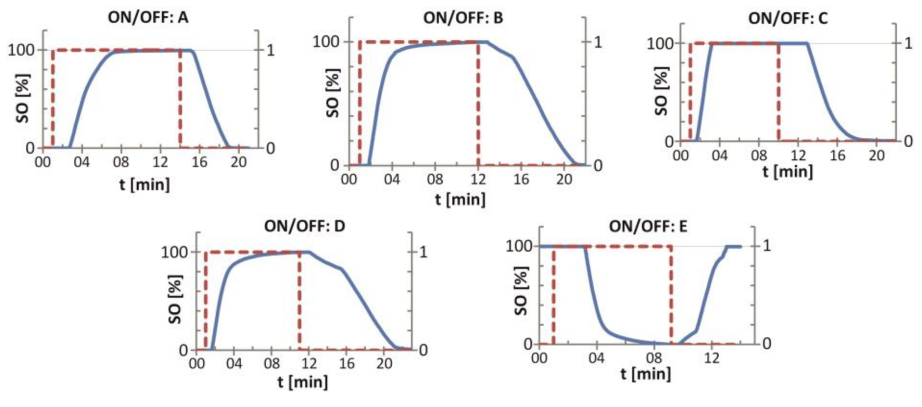


Fig. 4. The opening degree of the actuators as a function of time in the ON-OFF control

Table 1. The opening degree of the actuators as a function of time in the ON/OFF control

No.	**Actuators**	**Opening time**	**Closing time**	**Piston stroke**
		[min]	*[min]*	*[mm]*
1	A	5,41	3,39	3,65
2	B	4,05	8,36	5,37
3	C	2,05	7,19	4,79
4	D	4,41	9,59	5,27
5	E	3,45	5,09	4,06

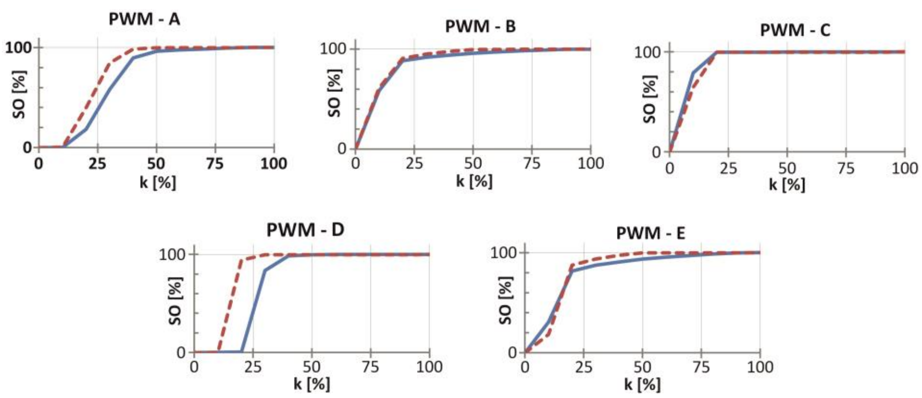


Fig. 5. The opening degree of the actuators as a function of duty cycle in the PWM control

Table 2. Table 1. The opening degree of the actuators as a function of duty cycle in the PWM control

k_w Ratio	**Opening degree of the actuators**									
	A		**B**		**C**		**D**		**E**	
[%]	*[%]*		*[%]*		*[%]*		*[%]*		*[%]*	
	O	Z	O	Z	O	Z	O	Z	O	Z
0	0	0	0	0	0	0	0	0	0	0
10	0	0	59	61	79	65	0	0	31	18
20	18	40	88	91	100	100	0	94	82	88
30	58	85	92	95	100	100	84	100	87	94
40	90	99	94	98	100	100	99	100	91	98
50	96	100	96	100	100	100	100	100	94	100
60	98	100	97	100	100	100	100	100	96	100
70	99	100	98	100	100	100	100	100	97	100
80	99	100	99	100	100	100	100	100	99	100
90	100	100	100	100	100	100	100	100	100	100
100	100	100	100	100	100	100	100	100	100	100

4. Conclusions

Based on the analyses that have been carried out, the conclusion can be drawn that the application of PWM type of control system in case of electrothermal actuators is a functional equivalent of the two-point control method. While PWM control enables determination of a average voltage ($U_{śr}$), when controlling LED brightness and DC motor speed, and at the same time gives the possibility to reach intermediate stages which imitate continuous control, then for elements functioning as electrothermal actuators, it must be treated as two-point or possibly as its slight modification.

Reference:

[1] Kargul K., Świątecki S., *Zawory regulacyjne w instalacjach klimatyzacji i grzewczych*, Chłodnictwo i Klimatyzacja 8/2012

[2] Nelik A., *PWM – modulacja szerokości impulsu*, Mikrokontrolery 5/2011

[3] Muffato L.A., Kunz D-S., *A high-fidelity simulation model for an asymmetric nonlinear electrothermal actuator*; Proceedings of 2013 European Control Conference (ECC), Zurich, 2013

ASSESSMENT OF SPATIAL AND TEMPORAL DISTRIBUTION OF THERMAL COMFORT AND IAQ IN LOW ENERGY HOUSES

Maria del Carmen Bocanegra-Yanez [*1], Paul Strachan[1], Paul Tuohy[1], Jon Hand[1], Tim Sharpe[2]

[1] *ESRU, University of Strathclyde*
75 Montrose Street
Glasgow, Scotland

[2] *MEARU, Glasgow School of Art*
167 Renfrew Street
Glasgow, Scotland

**Corresponding author: maria.bocanegra-yanez@strath.ac.uk*

ABSTRACT

According to the International Energy Agency, buildings represent over one-third of total final energy consumption. Thus, a more sustainable future begins with low energy buildings which must combine comfort and function using passive systems and new evolving technologies. Policies to reduce building energy consumption and carbon emissions have been developed worldwide during the last decades. As a consequence, Building Regulations and Standards require more insulated and air tight buildings which may lead to indoor environment issues when not designed appropriately or systems are not used as designed. Detailed building modelling and simulation can provide an indication of building performance and furthermore, it can be used to assess indoor environment issues. Although there have been several previous studies of the indoor environmental quality in low energy buildings, this research is focused on the variability of overheating risk and poor indoor air quality (IAQ) at different locations in the building at different times. The impact that different occupancy profiles and ventilation strategies has on the distribution of thermal comfort levels and IAQ in low-energy houses has been assessed using the detailed thermal simulation program, ESP-r.

The first part of the research involved developing a model of a low energy house in accordance with the Passivhaus (PH) standard. Comparison of modelling results and the results from the Passive House Planning Package (PHPP) confirmed that the modelling results were in good agreement with the PH Standard in terms of monthly heat gains and losses of the building. Secondly, more realistic assumptions were formulated so modelling results could be compared with measured data in terms of temperature, humidity and CO_2 distribution within various rooms of the building. Then a number of scenarios were formed, varying occupancy numbers and profiles with different ventilation regimes, which included natural ventilation, mechanical ventilation and mechanical ventilation with heat recovery options. These parametric variations were compared in terms of energy demand, plus temporal variation of indoor environment metrics (thermal comfort and IAQ).

The general conclusion arising from the analysis is that, contrary to the usual assumption of even distribution of the indoor environmental conditions, there can be significant variations in the internal distribution. Important factors are number and location of occupants and the movement of air within the building. Although this study was focused on climate representative of conditions in Scotland, similar variations would be expected in other climates.

KEYWORDS

Thermal Comfort, Indoor Air Quality, Passivhaus, Modelling

1 INTRODUCTION

Policies to reduce building energy consumption and carbon emissions have been developed worldwide during the last decades. In Europe the policy that started the path to energy efficiency in buildings was the European Directive 2002/91/EC (The European Parliament and the Council of the European Union, 2003) completed by the Directive 2010/31/EU (The European Parliament and the Council of the European Union, 2010) which established the European targets for reducing energy consumption by 20 % by 2020 through improved energy efficiency and integration of renewable energy in buildings.

Many energy performance standards have arisen to promote energy consumption reduction in buildings through different measures. Passivhaus is probably the most worldwide known low energy building standard since its appearance in Germany in the early 1990s (Building Research Establishment, 2011). The PH standard is based on reducing the energy demand needed for heating and cooling by the use of passive measures such as high levels of insulation, air tightness and control of solar gains.

Apart from its obvious benefits reducing energy consumption and environment impact of buildings, the PH standard can lead to health risks due to noise from installations, poor IAQ or overheating according to (Hasselaar, 2008). An example can be found in the Dormont Park Passivhaus Development situated close to Lockerbie, Scotland. This development consists of eight dwellings built to the PH standard. Indoor environmental conditions (temperature, relative humidity and CO_2 concentration) were monitored for two years in the living room, kitchen and main bedroom of the dwellings. In addition, occupants were interviewed regarding the thermal comfort showing that the main concern, common to all the dwellings, was the overheating during warm periods (MEARU, 2015).

Figure 1 shows that the temperatures in the South-facing bedrooms ranged from 20 to 35 °C, oscillating mainly around 25 °C during summer. However, the overheating frequency (when indoor temperature rises above 25 °C) calculated by PHPP (Passive House Institute, 2012) was only 0.2 % (MEARU, 2015), which highlights the limitations of PHPP overheating calculation under certain circumstances. This is clearly stated in the Certified European Passive House Designer Course (CEPH-Developing Group, 2013), indicating that the "accuracy is not very high for values above and below 10 %". Therefore, in those cases, dynamic simulation is needed if risk of overheating was to be avoided when designing a new PH.

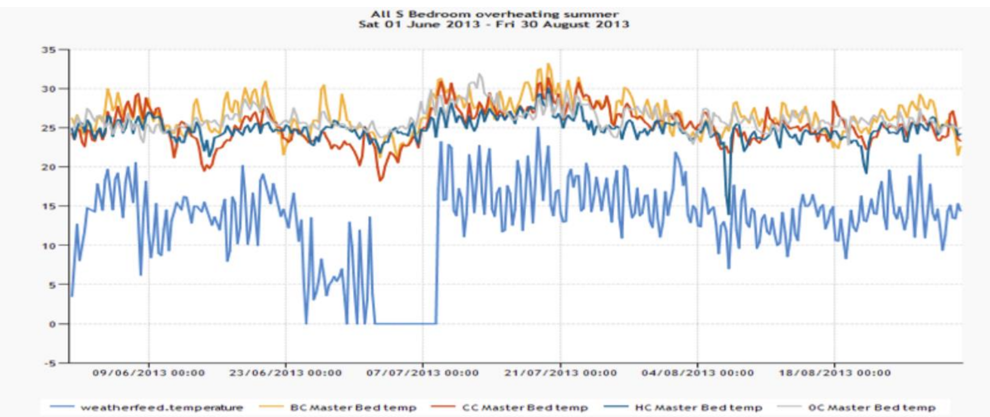


Figure 1. Temperature (in °C) in the south-facing bedrooms and ambient temperature for the summer period (June 1st to 26th August 2013) (MEARU, 2015)

Regarding the IAQ, monitored data shows that the 3-bedroom houses (with 4-5 occupants) have poor air quality issues in the main bedroom. Figure 2 shows that CO_2 concentration was above the acceptable range 50 % of the time in winter since windows remain generally closed. This fact also leads to Relative Humidity (RH) issues with levels below 40 % which can cause health problems.

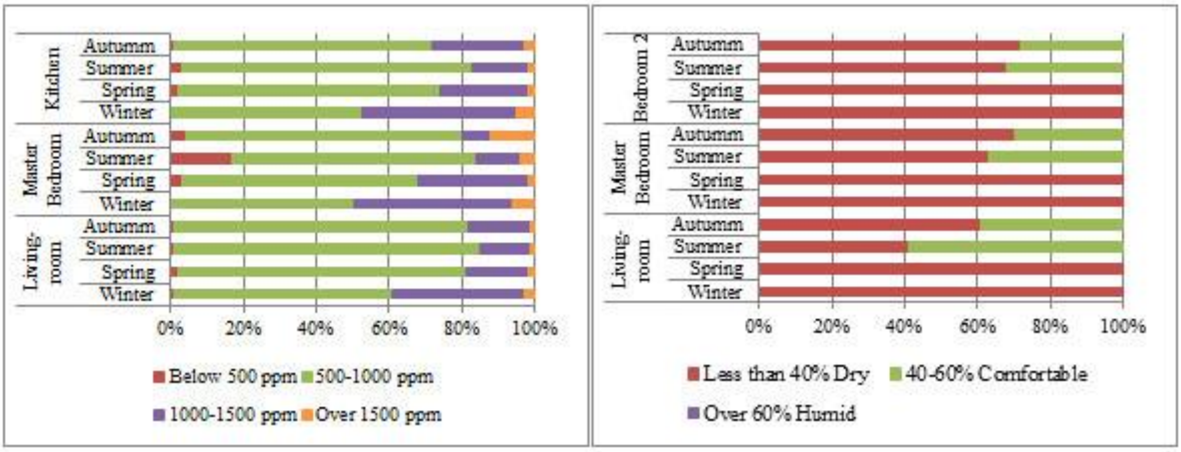


Figure 2. CO_2 concentrations by season for one of the 3-bedroom PH in Dormont (left) and Relative Humidity for one of the 2-bedroom PH (MEARU, 2015)

2 AIM

The aim of this study is to assess the impact that different occupancy profiles and ventilation strategies have on the distribution of thermal comfort levels and IAQ in a low energy house using the detailed thermal simulation program, ESP-r (ESRU, 2015). This program has been selected for its capability to simulate dynamic variations of indoor environmental conditions, in particular, temperature, relative humidity and CO_2 concentrations, in different zones within a building. Moreover, it has been extensively validated (Strachan et al, 2008).

3 MODELLING

This section describes the main characteristics of the model built using ESP-r, which will determine the thermal behaviour of the building. They are divided according to the following categories: Location and climate, Geometry and shading, Materials and constructions and Heating system. The house was chosen to represent a typical detached house in the UK, modified to conform to the PH standard. Once the model was built, it was crucial to investigate the results obtained to verify the model and make sure it was reasonable. This was done by a comparison between the results obtained using PHPP and ESP-r in terms of monthly heat gains and losses of the building.

PHPP is the official tool that is used for certifying that a house complies with the PH standard, and it has been validated and proved against measured data (Passivhaus Institut, 2014). However, PHPP has limitations. It follows the quasi-steady monthly method included in the European Standard ISO 13790:2008 for the calculation of energy use for space heating and cooling in buildings (Passive House Institute, 2012). Therefore, PHPP is focused on estimating annual energy performance, but it does not account for variation in the indoor environmental conditions in different parts of the house at different times. For that reason, the use of dynamic simulation tools is recommended when more accurate results are required.

Once the ESP-r model is shown to be in good agreement with the PH Standard, further modification of the ventilation options and the occupancy profile will allow extending the conclusions to other low-energy houses.

3.1 Model Description

The house modelled is situated in Dundee, East of Scotland. The climate is typically mild, with temperatures ranging from -6.1 to 24 °C, and prevailing southwest winds with maximum speed of 19.7 m/s. The dwelling comprises two floors and ten rooms. The first floor includes a hall, living-room, dining-room, kitchen and equipment room, while the second floor consists of a hall, three bedrooms and a bathroom. Each room has been modelled as an independent zone to quantify its indoor conditions separately. A sketch of the model can be seen in Figure 3 and the constructions used are detailed in
Table 1.

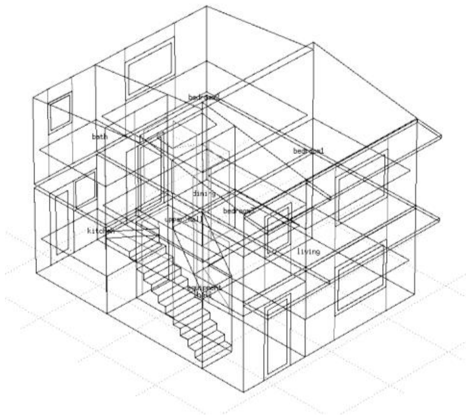


Figure 3. ESP-r Low Energy House Model

Table 1. Main construction details

Construction	U-value (W/m^{2}K)	Insulation		
		Thickness (mm)	Thermal Conductivity (W/mK)	Type
External Wall	0.075	500	0.04	Mineral fibre
Ground Floor	0.085	400	0.04	XPS
Roof	0.097	400	0.04	Glasswool
Window/Door:				
Glazing	0.490	-	-	-
Frame	0.800	-	-	-

Regarding control of solar gains, two types of shading devices are used, overhangs and blinds. Overhangs are designed in all south-facing windows to avoid overheating during summer when solar altitudes are greater, but allowing passive heating during winter when the direct solar incidence is more perpendicular to the windows. The blinds are assumed to be used in all south-facing windows when the indoor temperature rises over 23 °C in winter and 25 °C in summer.

Finally, the system that provides the heating demand to the dwelling needs to be defined. There are several ways to heat a PH to keep the indoor temperature at 20 °C. In this case, heating is supplied by a post-air heating unit as part of the Mechanical Ventilation Heat Recovery (MVHR) system. This unit senses the temperature in the hall downstairs and supplies warm air when the temperature drops below 20 °C. The maximum temperature of the supply air is set to 50 °C, slightly below the maximum set point imposed by the PH standard, which is 52 °C, to avoid dust burning inside the ventilation ducts.

3.2 Model Verification

Once the model is built, it is crucial to investigate the results obtained to verify the model and make sure it is reasonable. This has been done by a comparison between the results obtained using PHPP, ESP-r and monitored data. For the PHPP – ESP-r comparison, the model built using ESP-r has been simplified using PHPP assumptions, such as constant internal heat gains (IHG), constant monthly ambient temperatures, etc. Table 2 shows good agreement in terms of heat gains and losses for the month of January. The only variable that differs from one program to the other is the available solar heat gains. This can be explained by the impossibility of setting constant monthly irradiation values in the ESP-r climate file to match those in PHPP. For this reason, a further comparison has been made looking at annual heat gains and losses. In that case, available solar heat gains obtained from ESP-r only differ by 6 % from that obtained from PHPP, which proves the acceptability of the model.

Table 2. Comparison between the results obtained for January using PHPP and ESP-r

Unit: kWh	PHPP	ESP-r	Difference (%)
Infiltration losses	32.7	33.3	-2.0%
Ventilation losses	24.7	25.2	-2.0%
Available Solar Heat Gains	96.0	126.7	-32.0%
Internal Heat Gains	133.0	131.7	1.0%
Heating Demand	228.0	190.7	16.4%

3.3 Scenarios

The scenarios considered are divided in two categories: ventilation strategy and occupancy profile.

Ventilation strategy

Design options regarding the ventilation system are MVHR, mechanical ventilation without heat recovery and natural ventilation.

I. MVHR:
   The PH Standard requires the use of a heat recovery unit with a minimum efficiency of 75 %. More efficient units are available in the market and efficiencies around 90 % are common practice. In this case, the efficiency was assumed to be 92 %.

II. Mechanical Ventilation:
   This represents the case of a fault occurring in the heat recovery unit during the cold months or the use of summer bypass to avoid overheating during summer.

III. Natural Ventilation:
   This situation applies when fans are turned off and ventilation naturally occurs due to temperature and pressure differences between the inside and the outside of the house. Since the heating was supplied by a post-air heating unit in the MVHR system, this scenario requires another heating system to keep the house at 20 °C, for example, radiators in the different rooms. To control the ventilation in this case, it is assumed that occupants will open the windows of a room when it is occupied and the indoor temperature rises over 23 °C in winter and 25 °C in summer. In the case of bedrooms, windows remain closed during the sleeping hours.

Occupancy profile

The IHG of a building account for the heat generated by the occupants, the lighting and the appliances within each zone. For the present analysis, two different profiles have been considered:

A. 3-member family with PHPP appliance use assumptions
B. 5-member family with typical UK use of appliances

To calculate the heat generated by the occupants, an occupancy profile needs to be defined. This has been done assuming a typical three member-family and five member-family behaviour, differentiating between weekdays, Saturdays and Sundays, based on data from the UK Time Use Survey (TUS) 2000 (Flett, 2014), (ONS, 2003). The total heat load due to the use of the appliances has been calculated using the IHG sheet from PHPP for Scenario A, and a more realistic model based on typical UK use of appliances (Richardson et al, 2010) for Scenario B.

4. RESULTS AND DISCUSSION

This section analyses the modelling results obtained for the different scenarios described in the previous section. Each scenario has been assigned with a code (see Table 3) to facilitate the later result discussion.

Table 3. Scenario codes

Code	Scenario
A1	3-member family profile with PHPP appliance use assumptions and MVHR
A2	3-member family profile with PHPP appliance use assumptions and mechanical ventilation without heat recovery
A3	3-member family with PHPP appliance use assumptions and natural ventilation
B1	5-member family with typical UK use of appliances and MVHR
B2	5-member family with typical UK use of appliances and mechanical ventilation without heat recovery
B3	5-member family with typical UK use of appliances and natural ventilation

The results gathered for each scenario are the operative temperature, CO_2 concentration and relative humidity in every room, and the total heating demand for the whole building. Simulations were run for a winter period (January 1st to March 31st) and a summer period (June 1st to August 31st) using a time step of 15 minutes.

In order to compare these results, different categories were defined for each parameter using the recommendations in CIBSE Guide A (Chartered Institution of Building Services Engineers, 2006). These categories are shown in Table 4 and have been chosen to match the ranges used in the Building Performance Evaluation (BPE) of the Dormont Park Passivhaus Development.

Table 4: Categories for the indoor environmental parameters in this study (Mackintosh Environmental Architecture Research Unit (MEARU), 2015)

Variable	Factor	Winter		Summer	
		Other rooms	Bedrooms	Other rooms	Bedrooms
Temperature (°C)	Cold	<16°C	<16°C	<16°C	<16°C
	Cool	16-18°C	16-17°C	16-18°C	16-19°C
	Comfortable	18-22°C	17-19°C	18-23°C	19-23°C
	Warm	22-23°C	19-24°C	23-25°C	23-25°C
	Hot	23-28°C	24-26°C	25-28°C	25-26°C
	Overheating	>28°C	>26°C	>28°C	>26°C
CO_2 Concentration (ppm)	Ambient	<500 ppm			
	Ideal	500-1000ppm			
	Poor	1000-1500ppm			
	Very poor	>1500ppm			
Relative Humidity (%)	Dry	<40%			
	Comfortable	40-60%			
	Humid	>60%			

4.1 Heating demand

Figure 4 shows the heating demand during the winter period simulated for the different scenarios. As expected, results show greater heating requirements for the models with mechanical ventilation without heat recovery. However, the scenarios with natural ventilation present a lower heating demand than those with the MVHR system, which was not anticipated. These results are due to a very low ventilation rate in the case of the natural ventilation scheme as a result of the assumption of occupants opening the windows of a room only when this is occupied and the indoor temperature rises over 23 °C in winter or 25 °C in summer. This low ventilation rate led to overheating and IAQ issues as discussed in Section 4.2.

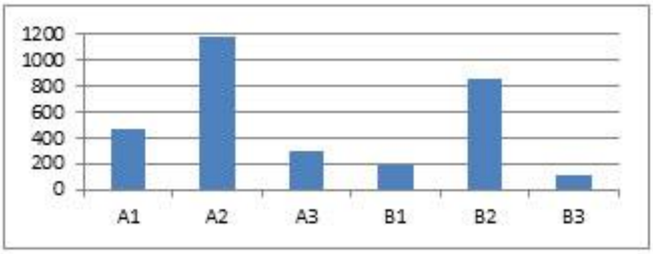


Figure 4. Heating demand (kWh) for the different scenarios

4.2 Overheating

Temperature results during the winter period show similar trends for all the scenarios considered. As an example, Figure 5 shows the result summary for Scenario A1. The temperature throughout the dwelling mainly stays in the comfortable and warm ranges, except for the kitchen, where it is hot 20 % of the time due to the use of cooking facilities. This can be seen more clearly by looking at the daily variation of indoor temperatures for a typical winter week (Figure 6).

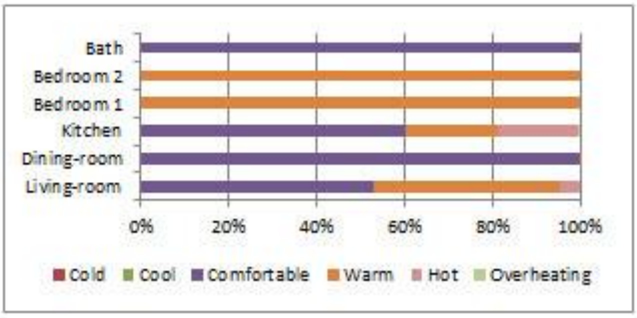


Figure 5. Temperature results for Scenario A1 for the winter period

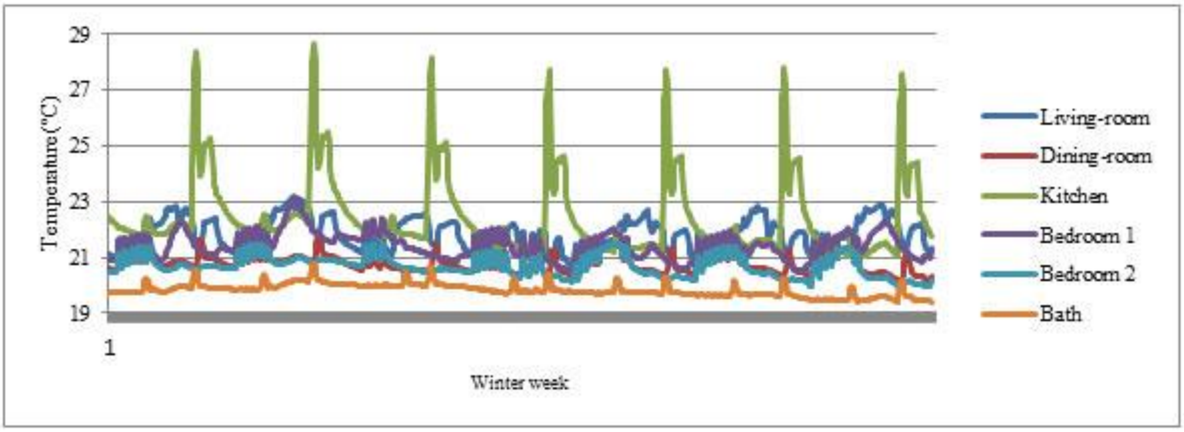


Figure 6. Indoor temperatures during a typical winter week for Scenario A1

For the summer period, differences between the scenarios considered are noticeable. There is more overheating in the 5-member family scenarios due to the higher level of IHG, as may be expected. Figure 7 shows the result summary for Scenario B1, B2 and B3, where overheating is an issue in all rooms for the MVHR and natural ventilation strategies. The use of summer bypass reduces the frequency of overheating but it is not enough to secure comfortable temperatures during summer as shown in Figure 8. Therefore, other strategies should be considered, like the use of hybrid ventilation or boost ventilation when needed.

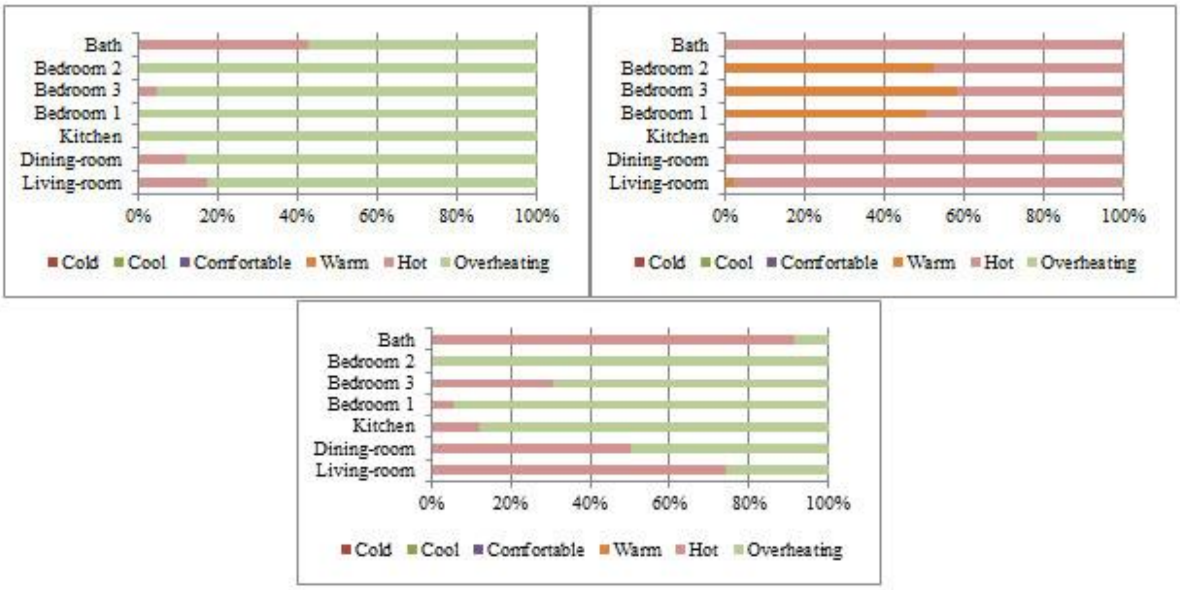


Figure 7. Temperature results for the summer period for Scenario B1 (top-left) , B2 (top-right) and B3 (down)

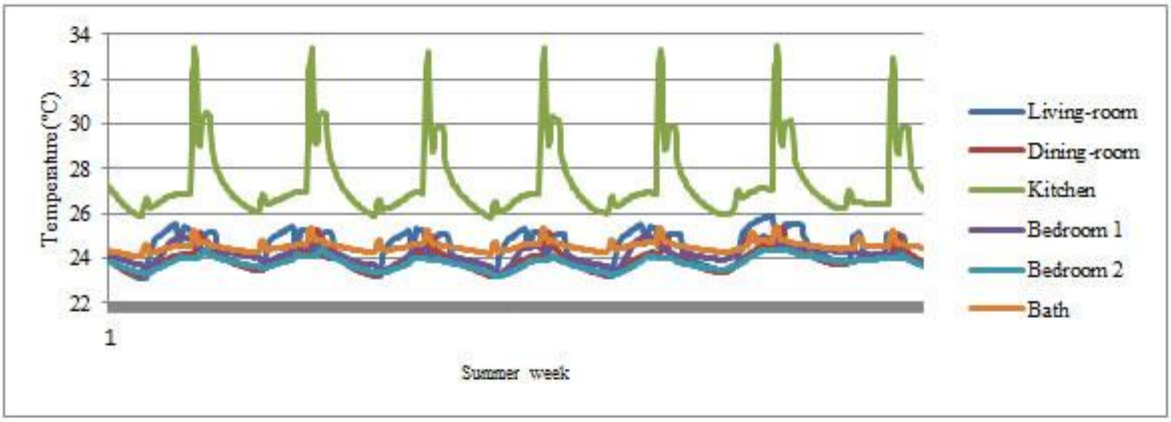


Figure 8. Indoor temperatures during a typical summer week for Scenario A2

4.3 CO_2 concentration

Predicted CO_2 levels show higher concentrations for the 5-member family scenarios as was expected. However, differences are not excessive. Hence, comparisons will be made between the mechanical and natural ventilation strategies. Figure 9 shows the CO_2 concentration frequencies for Scenarios B1/B2 and B3. As previously mentioned, it can be noticed that the assumption of occupants opening the windows of a room when this is occupied and the indoor temperature rises over 23 °C in winter and 25 °C in summer, does not lead to an adequate ventilation rate to keep good IAQ conditions inside the house, and this could lead to serious health problems in the long term. Further investigation should consider the impact of using trickle vents continuously so that a basic ventilation rate is guaranteed.

Beside the clear benefit of using mechanical ventilation in terms of IAQ, results in Figure 9 show that CO_2 levels remain too high for 50 % of the time in the living-room and 30 % of the time in the bedrooms, which represents essentially all the occupied time. A possible solution could be the use of boost ventilation when the CO_2 concentration rises above 1000 ppm.

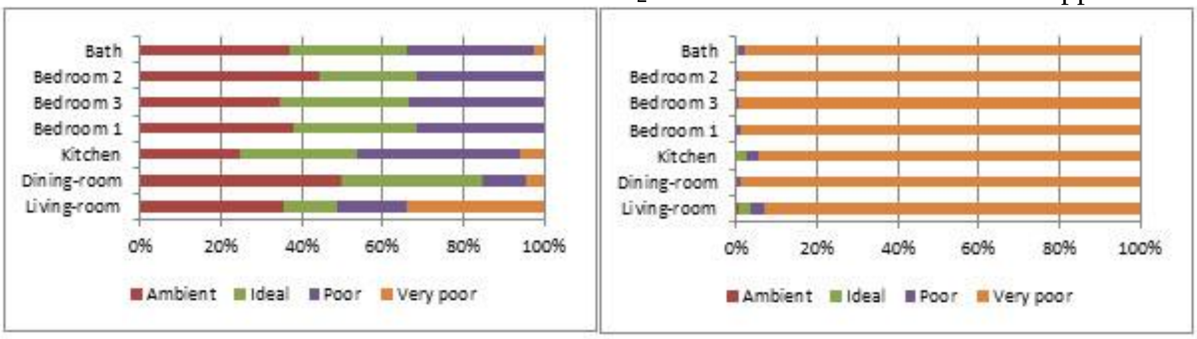
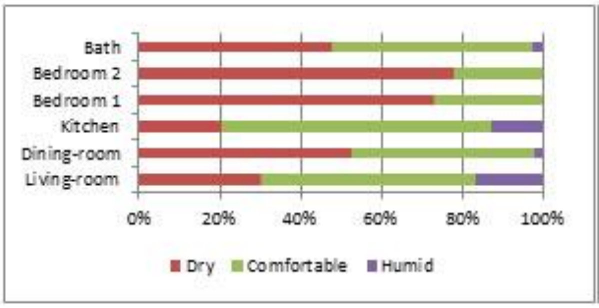


Figure 9. Temperature results for the winter period for Scenario B1/B2 (left) and B3 (right)

4.4 Relative Humidity

Relative Humidity (RH) should be in the range from 40 to 60 % according to CIBSE recommendations (CIBSE, 2006), in order to avoid mould growth and health issues. Results gathered from the simulation of the different scenarios show large discrepancies between the mechanical and natural ventilation strategies. Again, due to the low ventilation rate in the natural ventilation scheme, humidity is not released to the ambient and therefore, high levels of RH are found in all rooms, being slightly worse during winter (shown in Figure 10).



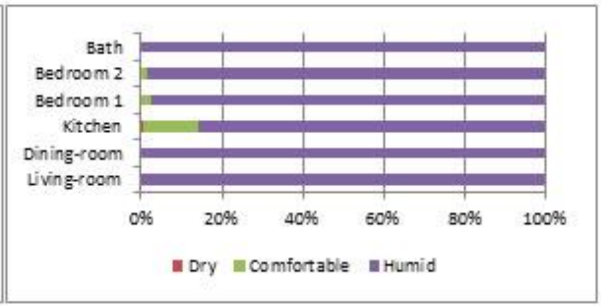


Figure 10. Temperature results for the winter period for Scenario A1/A2 (left) and A3 (right)

On the other hand, the use of mechanical ventilation may lead to very low levels of RH, mainly during the winter period. Results in Figure 10 show RH ranged from dry levels to comfortable levels in all rooms, with the frequency of dry air greater than 70 % in the bedrooms. Thus, moisture content of the incoming air should be controlled to keep the indoor conditions within the comfort range under different circumstances.

5. CONCLUSIONS AND FURTHER WORK

The main conclusion arising from the analysis is that, contrary to the usual assumption of even distribution of the indoor environmental conditions, there can be significant variations in the internal distribution. Important factors are the use of appliances in different rooms within the building at different times, and the ventilation strategy used.

From the scenarios analysed, it can be concluded that natural ventilation, which requires the action of the inhabitants opening windows, may lead to serious problems of overheating and poor IAQ, with high levels of CO_2 and humidity. However, mechanical ventilation strategies do not achieve ideal indoor environment conditions in all cases. The use of the heat recovery unit during summer led to unacceptable levels of overheating throughout the house with temperatures ranging from hot to overheating in all the rooms. This issue can be improved by the use of summer bypass. However, the risk of hot temperatures still remains very high: 50 % of the time for the bedrooms and almost all the time for the other rooms. Regarding IAQ and RH, although the use of mechanical ventilation showed better results than natural ventilation, CO_2 levels remain too high during 50 % of the time in the living-room and 30 % of the time in the bedrooms, which represents almost all the occupied hours. RH ranged from dry levels to comfortable levels in all rooms during winter, with the frequency of dry air greater than 70 % in the bedrooms.

As further work, additional scenarios should be simulated to investigate the impact of hybrid ventilation, with trickle vents continuously opened, or boost ventilation controlled by temperature and CO_2 concentration simultaneously. Also, new occupancy profiles could be defined in order to check the impacts of high occupancy levels, like inhabitants holding a party, children playing in a bedroom or intensive use of appliances like electronic devices throughout the house. The research should lead to recommendations for the design of appropriate ventilation systems and a better understanding of variations in the indoor environment.

6. ACKNOWLEDGEMENTS

We gratefully acknowledge the financial support received for this work from the BRE Trust.

7. REFERENCES

Building Research Establishment. (2011). *The Passivhaus Standard*. Retrieved May de 2015, from PASSIVHAUS: http://www.passivhaus.org.uk/

CEPH-Developing Group. (2013). *Certified European Passive House Designer Course* (Vol. A: The Passive House Standard).

Chartered Institution of Building Services Engineers. (2006). *CIBSE Guide A: Environmental Design.* London.

Energy Systems Research Unit. (2015). *Software: ESP-r*. Retrieved 2015, from University of Strathclyde: http://www.esru.strath.ac.uk/Programs/ESP-r.htm

Flett, G. (2014). Personal communication.

Hasselaar, E. (2008). Health risk associated with passive houses: An exploration. *Indoor Air 2008: Proceedings of the 11th International Conference on Indoor Air Quality and Climate.* Copenhagen: Technical University of Denmark.

Mackintosh Environmental Architecture Research Unit (MEARU). (2015). *Building Performance Evaluation. Final report. Domestic Buildings. Phase 2: In-use performance and post occupancy evaluation.* Dormont Estate.

ONS. (2003). *The United Kingdom 2000 Time Use Survey*. HMSO.

Passive House Institute. (2012). *Passive House Planning Package. PHPP. Version 7.* Darmstadt: Passive House Institute.

Passivhaus Institut. (2014). *PHPP - validated and proven in practice*. Retrieved 2015, from Passipedia - The Passive House Resource: http://passipedia.passiv.de/ppediaen/

Richardson, I., Thomson, M., Infield, D., & Clifford, C. (2010). Domestic electricity use: A high-resolution energy demand model. *Energy and Buildings, 42*(10), 1878-1887.

Strachan, P., Kokogiannakis, G., & Macdonald, I. (2008). History and Development of Validation with ESP-r Simulation Program. *Building and Environment, 43*(4), 601-609.

The European Parliament and the Council of the European Union. (4 de January de 2003). Directive 2010/31/EU. *Official Journal of the European Communities*, 65-71.

The European Parliament and the Council of the European Union. (18 de June de 2010). Directive 2010/31/EU. *Official Journal of the European Union*, 13-35.

MODEL PREDICTIVE CONTROL (MPC) OF HYBRID VENTILATION SYSTEMS IN OFFICE BUILDINGS WITH DYNAMIC GLASS FACADES

Tom Soendergaard Pedersen[1], Palle Andersen[*2], Christian Drivsholm[3], Niels Boel[4]

1 Aalborg University
Fredrik Bajers Vej 7
9220 Aalborg E, Denmark

2 Aalborg University
Fredrik Bajers Vej 7
9220 Aalborg E, Denmark
**Corresponding author: pa@es.aau.dk*

3 Danish Technological Institute
Kongsvang Allé 29
8000 Aarhus C, Denmark
Presenting author: cd@teknologisk.dk

4 Trend Control System
Oestergade 23-29
8000 Aarhus C, Denmark

Note: the contact addresses may be re-arranged

ABSTRACT

An advanced heat and electricity saving strategy for the regulation of hybrid ventilation systems with automatic night cooling (ventilative cooling), mechanical compressor cooling, natural ventilation and exterior solar shading by the inclusion of MPC (Model Predictive Control) has been developed in this project. The focus is on the optimization of the total energy cost (cost function) as compared to indoor climate requirements and variations in the outdoor climate. During the test period, the test persons could override the automatic control of the natural ventilation and solar shading. Their experience with the control strategy was studied by anthropologists.

KEYWORDS

Ventilative cooling, model predictive control MPC, natural ventilation

1 INTRODUCTION

The objective of predictive control (MPC) is to coordinate the use of mechanical ventilation, solar shading, window opening, heating and cooling coils to achieve room temperature and CO_2 containment within recommended limits on the cost in the cheapest way. It is used fixed energy prices any time of the day and focus hereby to achieve the lowest possible energy consumption. It is simple to use varying price forecasts and thus involve the availability of power across a predictive horizon. The predictive control coordinates the use of the above actuators using a dynamic model for the office, and is every half hour updated with the latest measurements of temperature and CO2 as well as the latest forecasts of outdoor temperature,

solar intensity and person load. From this calculates an optimized of the predictive control how the actuators set every half hour in a given time frame, for example 24 hours, the cheapest way to meet comfort requirements. In the example, calculates the predictive control every half hour a cycle of 48 signals for each actuator, that is, 5x48 values. The idea of predictive control is that only the first set of 5 values used in the controller, after half an hour corrected in subsequent values in a new optimization. For predicting outside temperature and solar intensity envisaged point forecasts from DMI which can deliver forecasts for every hour 54 hours until 4 times a day in a 3x3 km grid.
To illustrate the operation is made two types of simulations: one where constructed course outside temperature, solar intensity and person load in a winter scenario, a summer scenario and a transition scenario; in the other type is used DRY measured data for one year. In most simulations it is assumed that the model describes the real office sufficient and that the real temperatures of the process, etc. consistent with forecasts. This gives a picture of what the predictive control at best can achieve.

2 MODEL

If the calculations have to be very accurate, needs to be developed a large number of equations, in principle, one for each temperature of surfaces and thin layers of the heat-accumulating parts are desired; As mentioned, however, in many cases doubtful whether this trouble worthwhile. Here is, therefore, sought the greatest possible simplification with only 3 equations for each time interval.

Room air heat balance

$$\Sigma\alpha\cdot A\cdot(t_o - t_{air}) = \mathbf{B_o}\cdot(t_o - t_{air}) \qquad (1)$$

Where:
α is convective heat transfer for floors 2,5 $W/(m^2\cdot{}^oC)$, 2,0 $W/(m^2\cdot{}^oC)$ for ceilings and 3,0 $W/(m^2\cdot{}^oC)$ for walls
A is surface area, m^2
t_o is surface temperature, ^oC
t_{air} is room air temperature, ^oC

$$G\cdot c_p\cdot(t_{air} - t_i) = \mathbf{B_l}\cdot(t_{air} - t_i) \qquad (2)$$

where:
G is the airflow (natural or mechanical), kg/s
c_p is heat capacity of air, $W/(kg\cdot{}^oC)$
t_i is the inlet air temperature, ^oC

$$Q_{conv} + \mathbf{B_o}\cdot(t_o - t_{air}) = \mathbf{B_l}\cdot(t_{air} - t_i) \qquad (3)$$

Where:
Q_{conv} is convective heat to the room, W

Room surface heat balance

$$\Sigma k\cdot A = \mathbf{B_u} \qquad (4)$$
$$\Sigma k\cdot A = \mathbf{B_r} \qquad (5)$$

Where:
(4) is heat transfer from surface to out side
(5) is heat transfer from surface to neighbor room

$$\Sigma k'_a \cdot A = \mathbf{B_a} \tag{6}$$

Where:
(6) is heat transfer from surface to fictive accumulative layer in the wall

$$Q_{rad} = \mathbf{B_u} \cdot (t_o - t_u) + \mathbf{B_r} \cdot (t_o - t_r) + \mathbf{B_o} \cdot (t_o - t_{air}) + \mathbf{B_a} \cdot (t_o - t_a) \tag{7}$$

Where:
Q_{rad} is heat by radiation, W
t_a is temperature in fictive accumulated layer, ^oC

<u>Fictive accumulative wall layer heat balance</u>

$$\Sigma c_p \cdot \varrho \cdot A \cdot e \cdot \frac{\partial t_a}{\partial \tau} = \boldsymbol{S} \cdot \frac{\partial t_a}{\partial \tau} \tag{8}$$

Where:
S is the rooms heat capacitive, $J/^oC$
c_p is heat capacitive, $W/(kg \cdot {}^oC)$
ϱ is density, kg/m^3
A is surface, m^2
e is thickness of fictive accumulative layer, m

$$\mathbf{B_a} \cdot (t_o - t_a) = \mathbf{S} \cdot \frac{\partial t_a}{\partial \tau} \tag{9}$$

The differential equation (9) must be changed to a difference equation to be processed manually. Divided the analysis time in intervals of length $\Delta\tau$ applies to the n'th interval:

$$\mathbf{B_a} \cdot (t_{o,n-1} - t_{a,n-1}) = \mathbf{S} \cdot \frac{t_{a,n} - t_{a,n-1}}{\Delta\tau} \tag{10}$$
$$Q_{conv,n} + \mathbf{B_o} \cdot (t_{o,n} - t_{air,n}) = \mathbf{B_l} \cdot (t_{air,n} - t_{i,n}) \tag{11}$$
$$Q_{rad,n} = \mathbf{B_u} \cdot (t_{o,n} - t_{u,n}) + \mathbf{B_r} \cdot (t_{o,n} - t_{r,n}) + \mathbf{B_o} \cdot (t_{o,n} - t_{air,n}) + \mathbf{B_a} \cdot (t_{o,n} - t_{a,n}) \tag{12}$$

Summary – room simulation model

Input: $\mathbf{B_a}$, $\mathbf{B_l}$, $\mathbf{B_o}$, $\mathbf{B_r}$, $\mathbf{B_u}$, **S**, Q_{conv}, Q_{rad}, t_u, t_i, t_r
Output: t_{air}, t_o, t_a

2.1 Cost function

Electric power to the fans is given partly by an empirical model for the power consumption

$$P_{ventilatorer} = \left(\frac{q_l}{q_{max}}\right)^{2.5} \cdot q_{max} \cdot SEL_{\max} \ [Watt] \tag{13}$$

Where:

q_{max} is the maximum air flow, m^3/s
q_l is the actually air flow, m^3/s
SEL_{max} is the specific fan power at q_{max}, J/m^3. It is 2100 J/m^3 in Denmark

$$P_{cooling\ coil} = COP \cdot P_{el\text{-}compressor} \tag{14}$$

Where
COP is the compressor cooling factor. It has the value 3,0

$$P_{heating} = q_l \cdot \varrho \cdot c_p \cdot (t_i - t_{air}) \tag{15}$$

$$P_{heating\ coil} = q_l \cdot \varrho \cdot c_p \cdot (t_i - t_u') \tag{16}$$

Where:
t_u' is equal to $(t_u + \eta \cdot (t_{air} - t_u))$ (17)
η is the heat recovery value. In Denmark it is 0,70

$P_{heating\text{-}radiator}$ = desirable heat

Electricity price = 2,25 Dkr/kWh
Heating price = 0,75 Dkr/kWh

It is assumed that the cost of solar protection is negligible. In MPC routine, we have chosen to put a very small price of solar protection as a simple way to avoid foreclosure remains active for periods where there is no need for it.

In view of the mentioned costs we can now establish a cost function V, which is the price of comply comfort within a given time frame as a function of the use of the various actuators, ventilation (q_l), heating ($P_{heating}$), cooling ($P_{cooling}$), solar shading and window opening.

$$V\left(q_l, P_{heating}, P_{cooling}\right) = \sum_{k=1}^{N} hP_{ventilatorer}(k)El\_price + hP_{heating}(k)Heat\_price + \frac{hP_{cooling}(k)Heat\_price}{COP} + \left(b_s(k) - 1\right)^2 Solar\_shading \tag{18}$$

N is here the time horizon, h is the time resolution.

The cost function to be minimized, taking into account the following limitations given by comfort and actuators extremes:

Indoor temperature, T_{air}, determined from room temperature dynamics with the conditions in terms of climate and space use.
CO_2 concentration determined from room.
$T_{min} \leq T_{air} \leq T_{max}$ where $T_{min\text{-}summer}$= 23 ^oC, $T_{min\text{-}winter}$ = 20 ^oC, $T_{max\text{-}summer}$ = 26 ^oC and $T_{max\text{-}winter}$ = 24 ^oC
$CO_{2\ max}$ = 1000 ppm
$0 \leq q_l \leq q_{max}$, where q_{max} = 250 m^3/h
$P_{heating}$ and $P_{cooling}$ must be positive.
Ventilation through window opening between 0 and maximum window flow, $0 \leq q_{vent\text{-}nat} \leq q_{vent\text{-}nat\text{-}max}$, where $q_{vent\text{-}nat\text{-}max}$ = 1,8 liter/$(s \cdot m^2)$ floor area.

Indoor Climate Control allows sequences of actuator signals q_l (1) ... q_l (N), $P_{heating}$ (1) ... $P_{heating}$ (N), $P_{cooling}$ (1) ... $P_{cooling}$ (N), bs (1) ... bs (N), $q_{vent\text{-}nat}$ (1) ... $q_{vent\text{-}nat}$ (N) in order to comply with the restrictions. Many combinations of actuator values can comply with the limits. Model Predictive Control (MPC) can calculate the sequence of the actuator with the lowest cost, if it has the correct information in the form of weather predictions, number of people in the room and it has a model that can correctly calculate the inside temperature and CO_2 concentration.

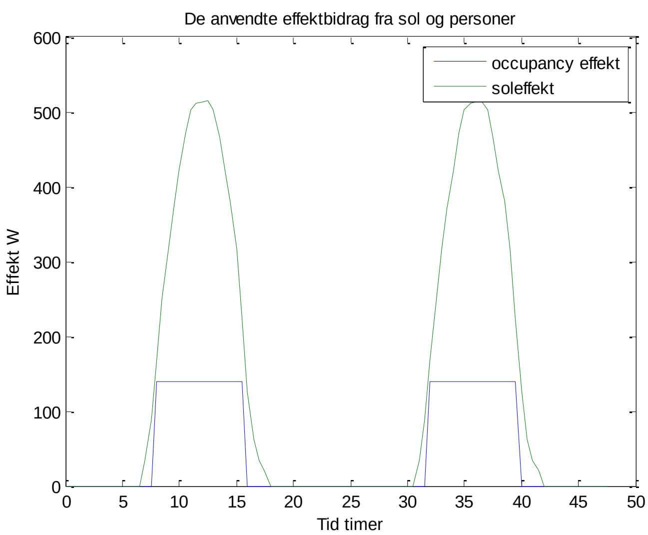
MPC operates so that within a predictive horizon, N, calculated a course of actuators which minimize the cost, then used the first value of the actuator signals.

The room simulation model and the minimum cost function is running in the Matlab environment with CVX included. CVX is a Matlab-based modeling system for convex optimization. CVX turns Matlab into a modeling language, allowing constraints and objectives to be specified using standard Matlab expression syntax.

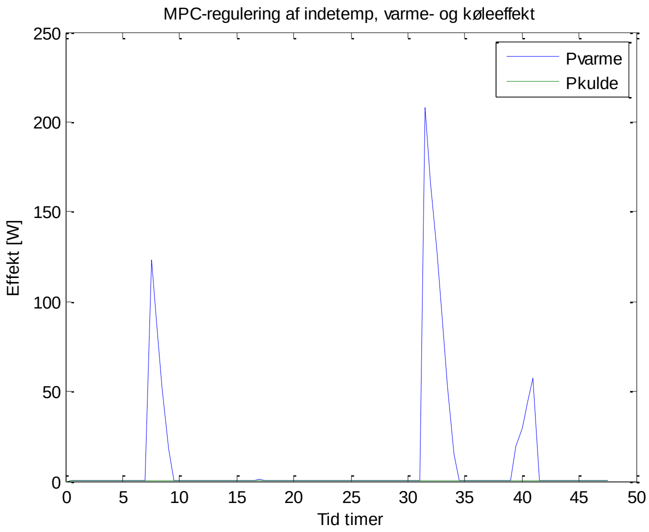
2.2 Results

There is in this paper presented some selected simulation.

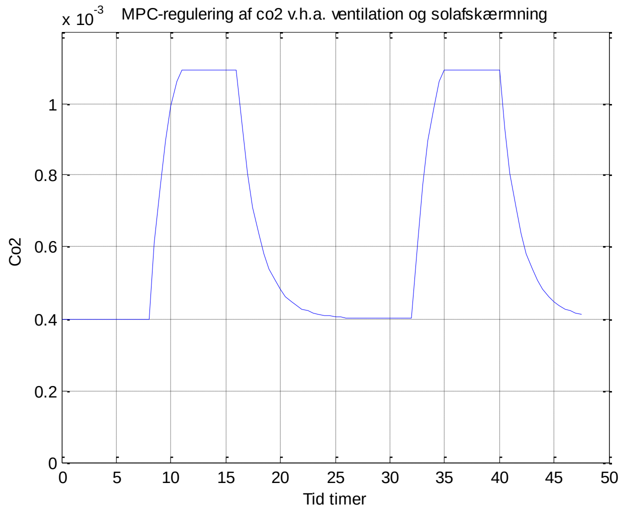
Winter with heating demand:

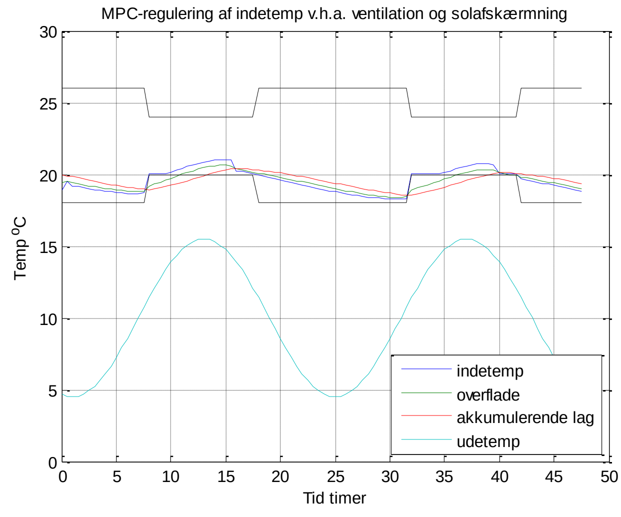


Green line = solar power.

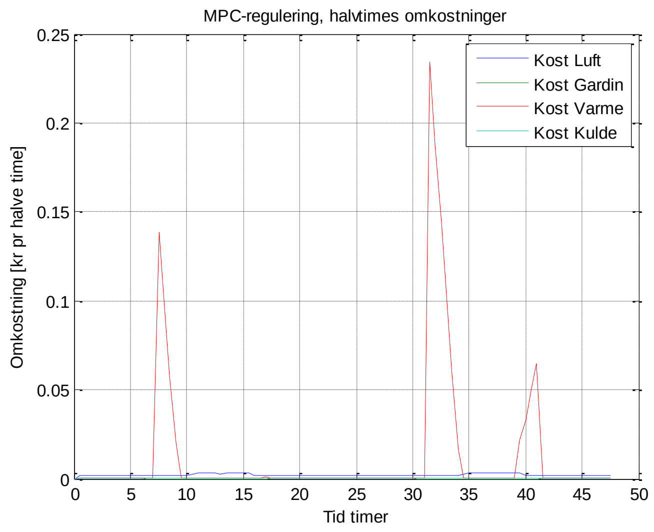


Blue line = $P_{heating}$ and green line = $P_{cooling}$



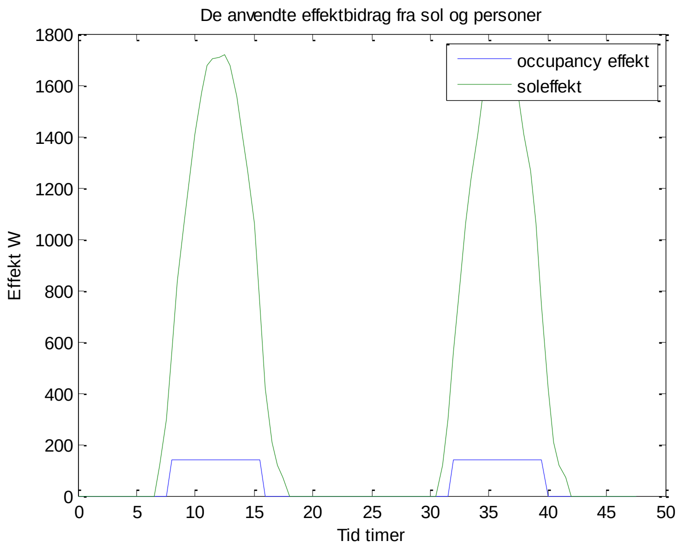


Blue line = t_{air}, green line = t_o, red line = t_a and light blue = t_u

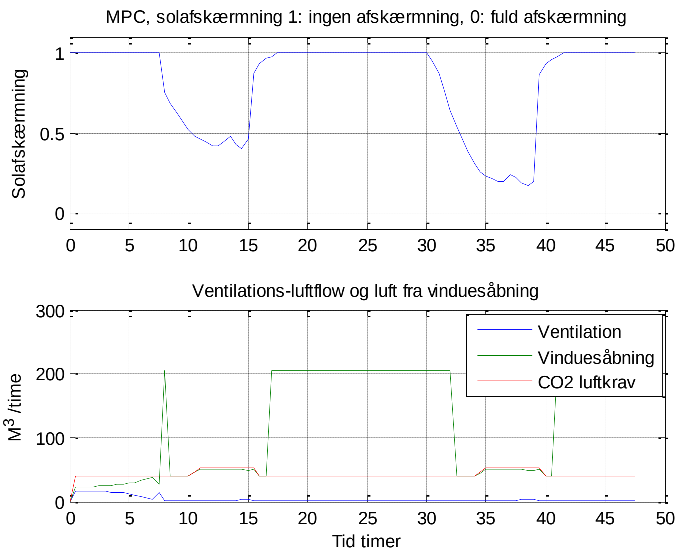


Red line = cost (heating), blue line = cost (ventilation)

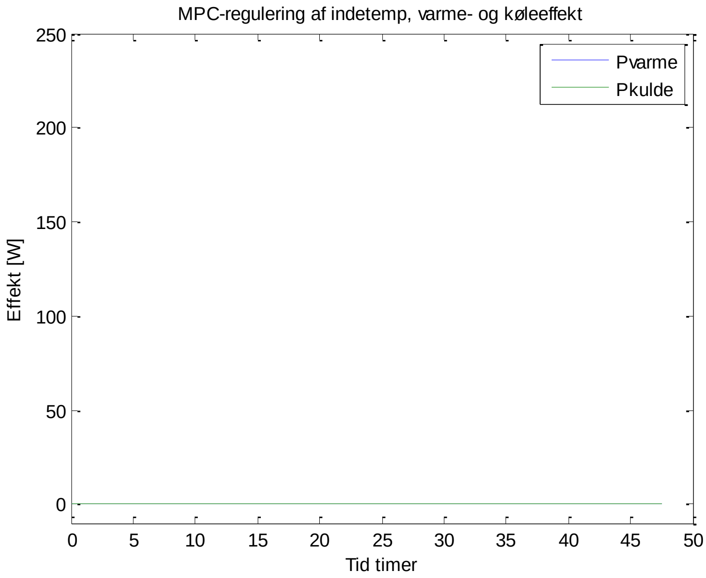
Summer:



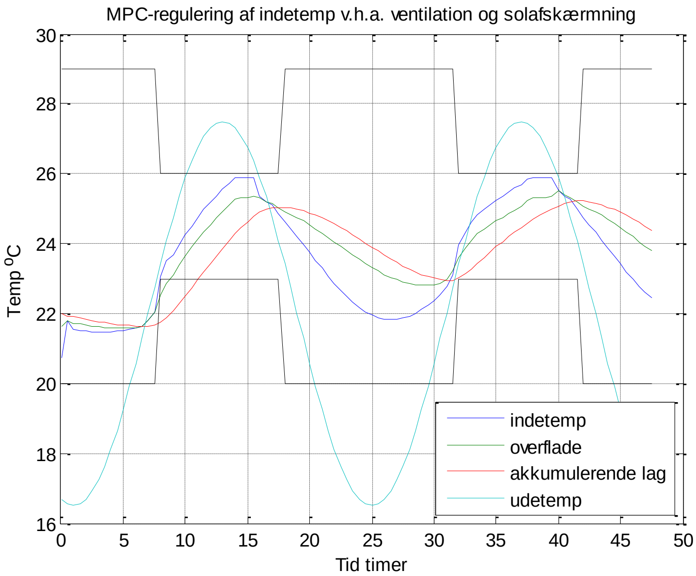
Green line = solar power.



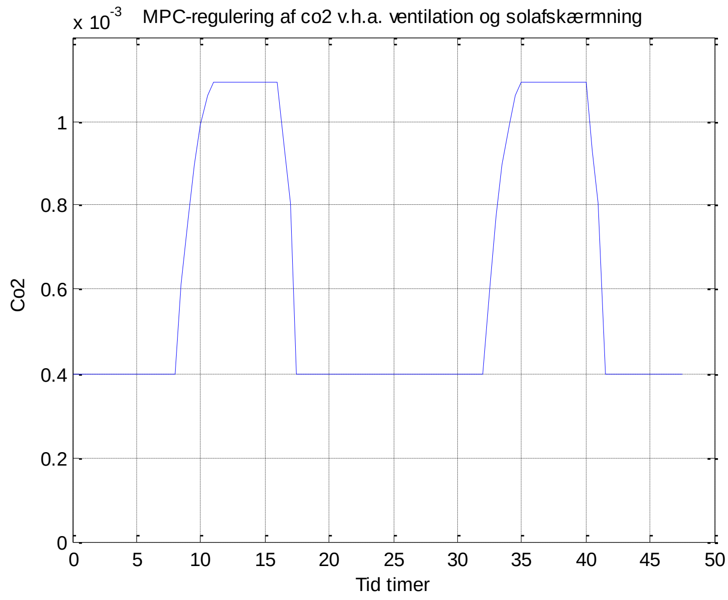
Solar shading. "1" = no shading
Blue line = mechanical ventilation, green line = natural ventilation

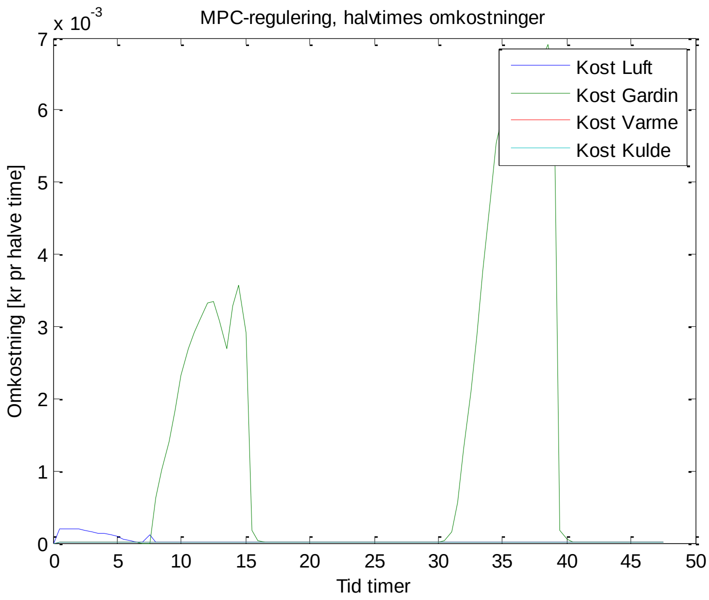


Blue line = $P_{heating}$ and green line = $P_{cooling}$



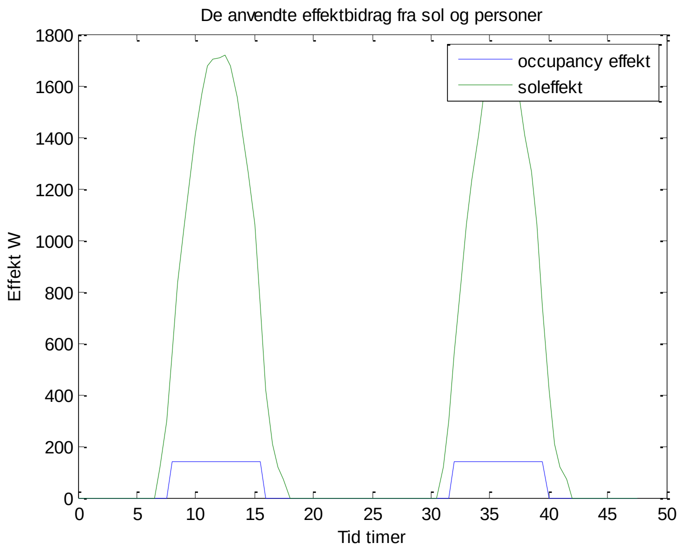
Blue line = t_{air}, green line = t_o, red line = t_a, light blue = t_u


x 10^-3
MPC-regulering af co2 v.h.a. ventilation og solafskærmning
Co2
0
0.2
0.4
0.6
0.8
1
0
5
10
15
20
25
30
35
40
45
50
Tid timer

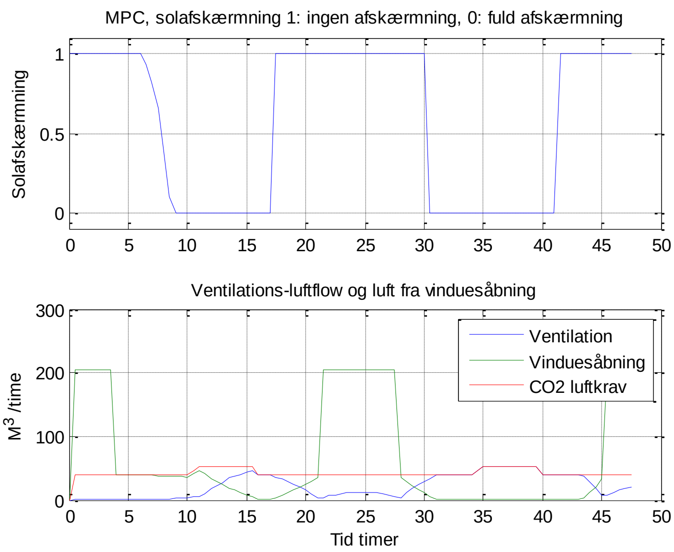

x 10^-3
MPC-regulering, halvtimes omkostninger
Omkostning [kr pr halve time]
0
1
2
3
4
5
6
7
0
5
10
15
20
25
30
35
40
45
50
Tid timer
Kost Luft
Kost Gardin
Kost Varme
Kost Kulde

Green line = ”cost” solar shading, blue line = cost ventilation.

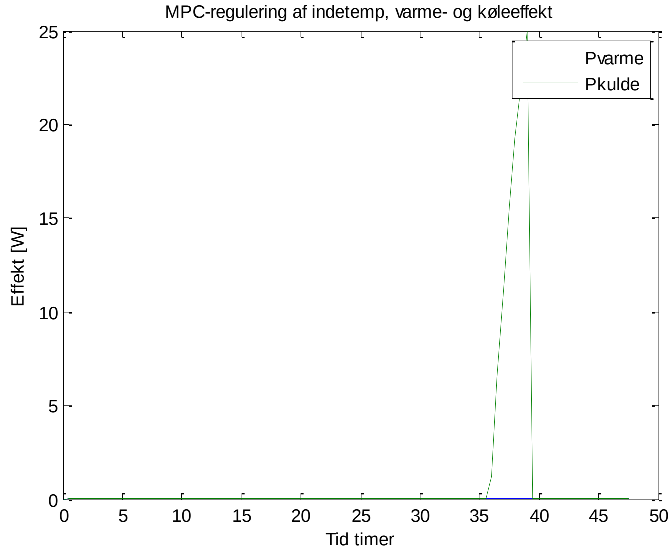
<u>Hot summer:</u>



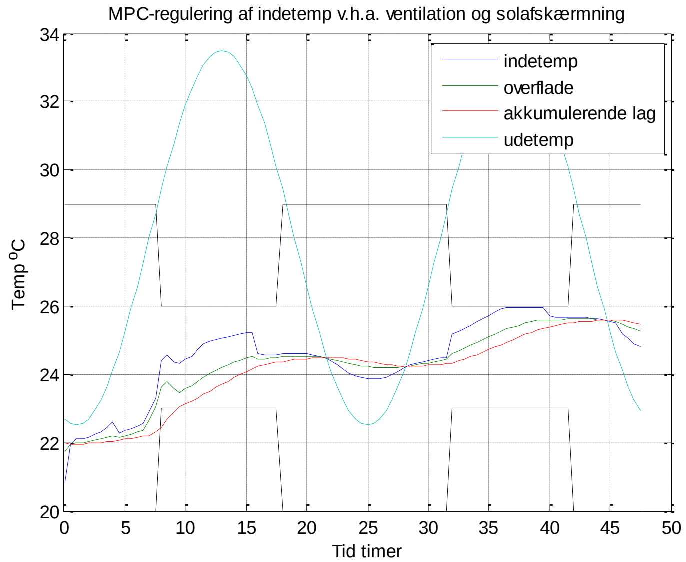
Green line = solar power.



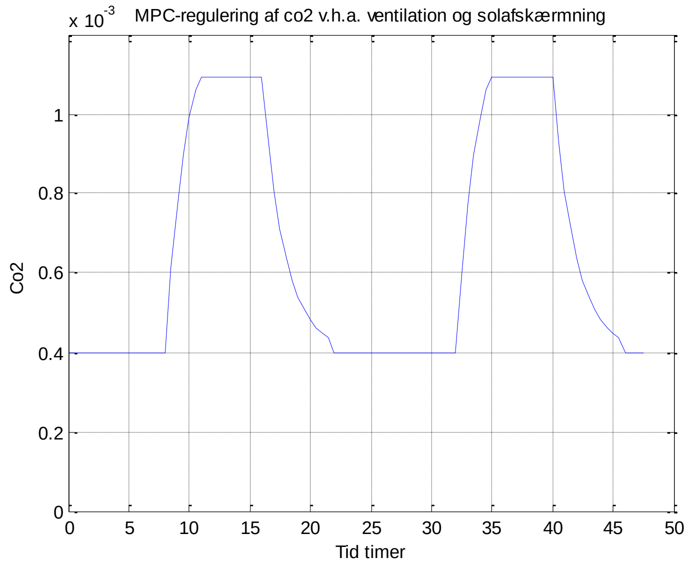
Blue line = solar shading. “1” = no shading.
Blue line = mechanical ventilation, green line = natural ventilation

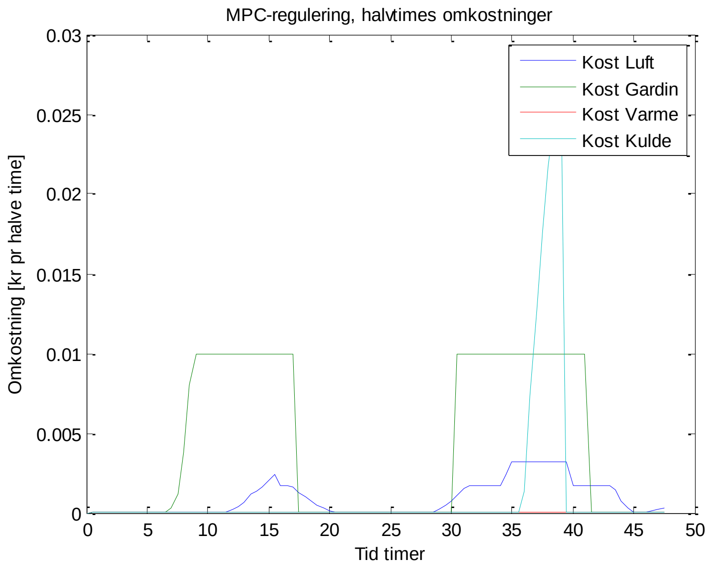


Green line = $P_{cooling}$ *and blue line* = $P_{heating}$



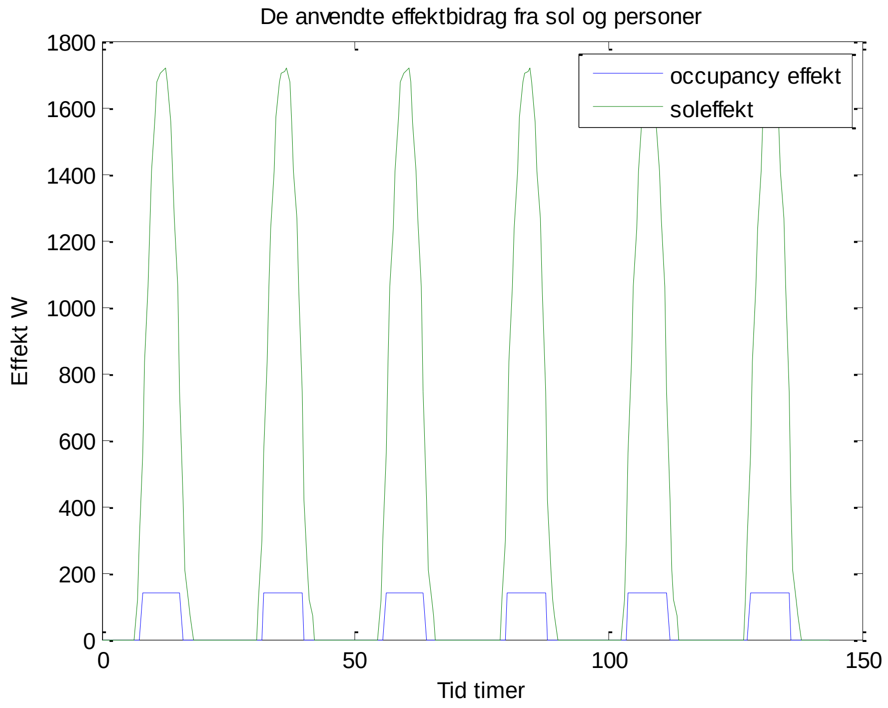
Blue line = t_{air}, *green line* = t_o, *red line* = t_a *and light blue* = t_u



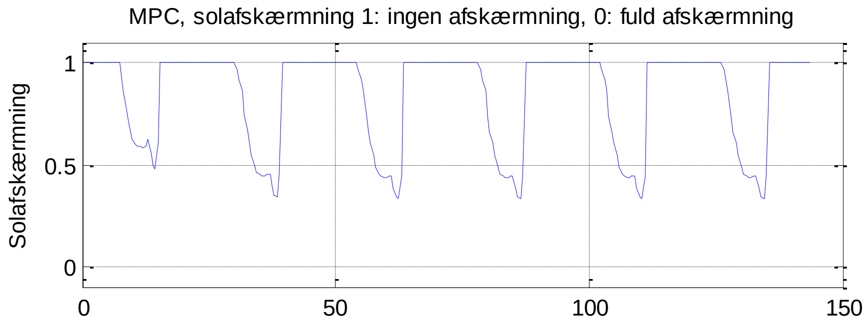


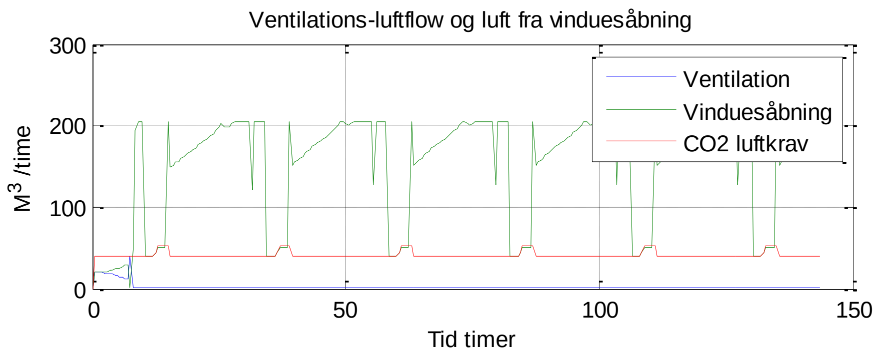
Blue line = cost (ventilation), green line = ”cost” (solar shading), light blue line = cost (cooling)

Summer (long time simulation)

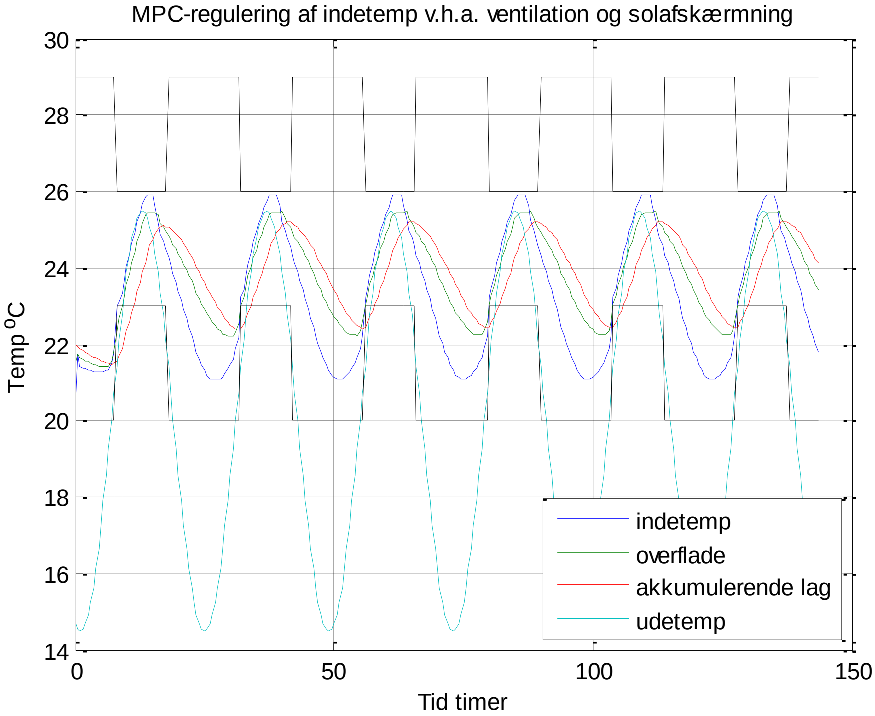


Green line = solar power.

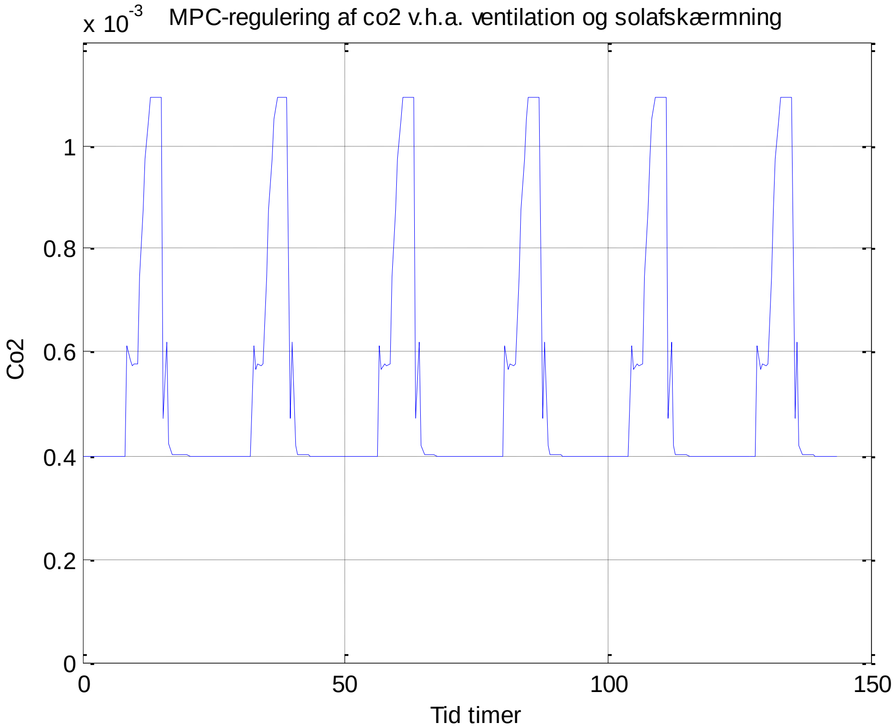


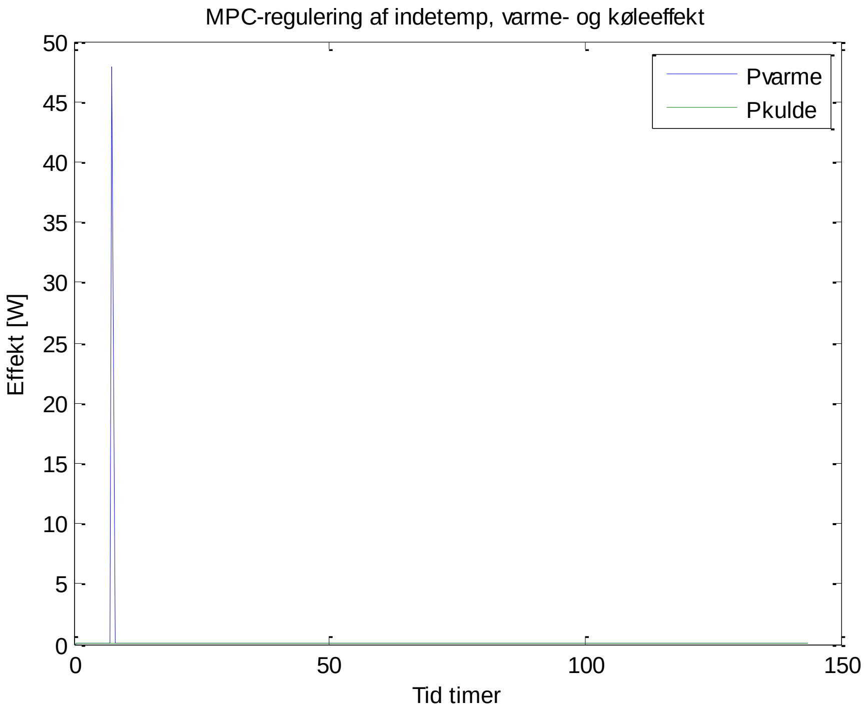


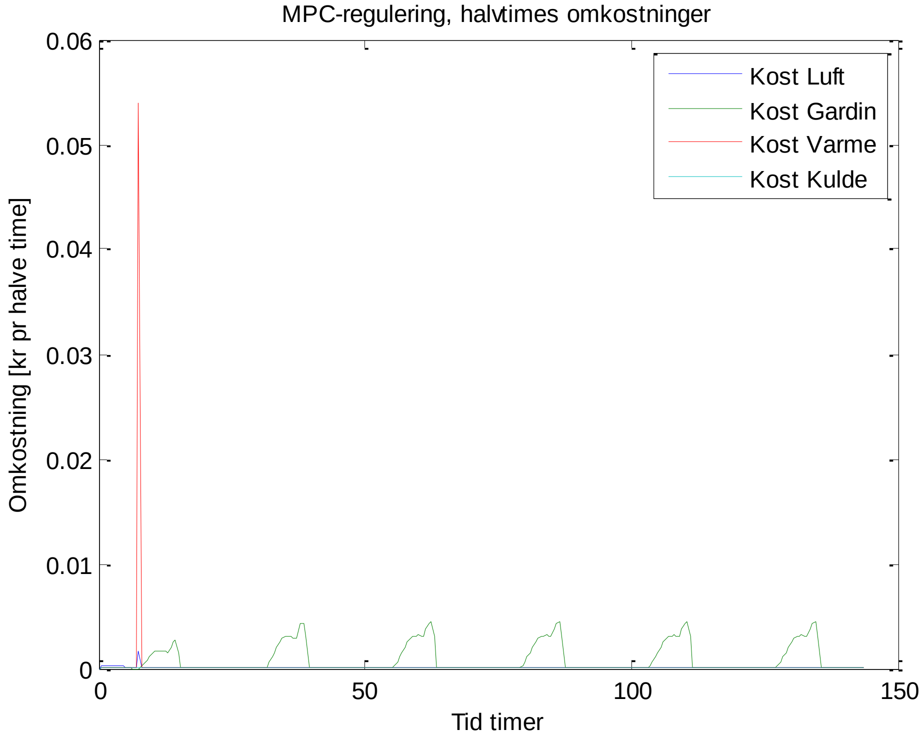
Blue line = solar shading. "1" = no shading.
Blue line = mechanical ventilation, green line = natural ventilation



Blue line = t_{air}, green line = t_o, red line = t_a and light blue = t_u







Green line = "cost" solar shading

3 CONCLUSIONS

Using an simple building model, MPC and weather forecast in combination with indoor climate requirements and cost function to control: Mechanical ventilation, natural ventilation, heating, cooling, solar shading in an optimal way. It is not necessary with more than 24 hours weather forecast. On a normal summer day in Denmark, it was possible to hit a room

temperature of 23 degrees C at 8 am and a room temperature of 26 degrees C at 16 without the use of mechanical cooling but only natural ventilation and solar shading.
The advanced control strategy can be optimized more through user feedback.

4 ACKNOWLEDGEMENTS

5 REFERENCES

Adamson, Bo (1968): "Värmebalans vid rum och byggnader", Tekniske Högskolan, Lund, Sverige.

Statens Byggeforskningsinstitut (1985): "tsbi, version 2.1 - termisk simulering af bygninger og installationer".

Danvak Grundbog (2006): "Varme- og Klimateknik", page 137-140.

DS/EN 15251 (2007): "Indoor environmental input parameters for design and assessment og energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics".

THERMAL PERFORMANCE ANALYSIS OF A SOLAR CHIMNEY, BASED ON THE EXPERIMENTAL STUDY OF THE MAIN DRIVING VARIABLES IN A PHYSICAL PROTOTYPE

Jesús Arce[1], María José Jiménez[*2], Ricardo Enríquez[2], Jesús Xamán[1], Sergio Castaño[2], Gabriela Álvarez[1] and María del Rosario Heras[2]

1 Mechanical Engineering Department.
Centro Nacional de Investigación y Desarrollo Tecnológico (CENIDET-TNM-SEP).
Cuernavaca, Morelos, México. 62490

2 Energy Efficiency in Buildings Unit.
Centro de Investigaciones Energéticas, Medioambientales y Tecnológicas (CIEMAT).
Madrid, Spain. E-28040.
**Corresponding author: mjose.jimenez@psa.es*

ABSTRACT

This work presents the thermal behavior of a stand-alone experimental solar chimney during one year. The dimensions of the solar chimney are 5.60 m high, 1.0 m width, and 0.52 m depth. The absorber plate is made of a common reinforced concrete wall of 4.5 m high, 1.0 m width and 0.15 m depth. This system was designed and constructed in 2003, and it is located in the "**L**aboratorio de **E**nsayos Energéticos para **C**omponentes de la **E**dificación ***(LECE)"*** at the "**P**lataforma **S**olar de **A**lmería ***(PSA)"*** in Spain. The inlet of this solar chimney was redesigned, and also the instrumentation of the system was increased and improved recently, as well as its air outlet fitting. During one year (2014), the solar chimney was monitored and several experimental variables were measured. The results present the temperature profiles of the different measured elements of the solar chimney as well as the air flow rate through the solar chimney channel. It was observed that the effect of the outdoor wind added to the thermal effects plays an important role affecting the performance of the solar chimney studied.

KEYWORDS

Natural Ventilation; Solar Chimney

1 INTRODUCTION

The use of conventional systems of heating and air conditioning to achieve thermal comfort represents nowadays a high percentage of the energy consumption in buildings. Therefore it contributes to the problem of the global warming.

With the purpose of contributing to the decrease of emission of pollutants, the use of non-renewable energy resources must be limited and the use of renewable resources of energy must be promoted. The following renewable resources can be mentioned: *wind, solar, thermal, photovoltaic, and biomass energies*.

The new bioclimatic designs of housings and buildings should consider the utilization of these renewable sources of energy and the use of passive systems of air conditioning, like wind towers, Bansal et al., (Bansal et al., 1994), trombe walls Ben et al., (Ben et al., 1991), and solar chimneys, Bouchair (Bouchair, 1994). The treatment of these sources of energy and its application represent a new challenge.

Part of the thermal comfort and the improvement of the air quality to the interior of housings and buildings are obtained through the use of natural ventilation. Among some experimental studies on these passive systems we can mention those undertaken by Khedari et al., (Khedari et al., 2000). They carried out an experimental research with four types of solar chimneys (The Solar Collector of Roof, the modified Trombe wall, the Trombe wall and the Metallic Solar Wall) connected to a room of 25 m^3. Common materials were used in the construction, and the surface area was 2 m^2 each one. The results show that the four devices allow inducing natural ventilation which improves thermal comfort and reduce overheating of the room up to 50 %.

An experimental study in laboratory conditions of Solar Chimneys was published by Chen et al., (Chen et al., 2003). The internal dimensions of the Solar Chimney were; 1.5 m high, 0.62 m width and a changeable space of the width of the channel (0.1 to 0.6 m). Uniform heat flows were applied of 200, 300, 400, 500 and 600 W/m^2 across one of the walls. The angle of inclination of the Chimney was changed from 15º to 60º every 15º with regard to the vertical position. The results show that there was a maximum air flow rate reached when the angle of inclination of the chimney was 45º with a width of 0.2 m and a height of 1.5 m of the chimney applying 400 W/m^2, which is equivalent to 45 % more than for a vertical chimney under similar conditions. It was observed that the distributions of temperatures and speeds of the air were uniform through the width of the studied chimney.

Khedari et al., (Khedari et al., 2003), conducted an experimental study of the operation of a solar chimney integrated into a room of a building equipped with air conditioning (AC) and a volume of 25 m^3. The results showed that the use of a solar chimney together with AC's system can diminish up to 30 % the electric power consumption, compared with the consumption of energy that would demand a building that uses only AC's systems.

Among the experimental studies to small scale, it can be mentioned the one held by Chakraborty and Fonseca (Chakraborty and Fonseca, 2005). Emad (Emad, 2006), conducted an experimental study using diverse technologies on the passive cooling in scaled models of rooms with metallic structures. The following technologies of cooling were used; a white Roof to diminish the heat gains to the interior of the room, thermal insulation over and/or below the roof, roof with water tank with and without insulation, evaporative cooling and by using a solar chimney.

The conclusion obtained was that the best modifications of the roof, to diminish the temperature inside enclosures constructed with metallic structures in hot regions, are the evaporative cooling and the solar chimney. The use of the solar chimney is an effective technology to reduce the temperature inside enclosures with metallic structures, beside producing ventilation and getting thermal comfort.

Another experimental study of passive cooling using a solar chimney and a wetted roof in a humid and warm climate in Thailand has been done by Chungloo and Limmeechokchai (Chungloo and Limmeechokchai, 2006). The results showed that the system works better when the use of the chimney and the water scattering in the roof there are combined. Burek and Habeb (Burek and Habeb, 2007), carried out experimental studies in a vertical channel, simulating the thermal performance of a solar chimney. Heat was supplied to the absorber plate through an electrical heater. The tests consisted in obtaining the profiles of temperature and air velocity of the in the channel, changing the amount of heat given to the system (200-1000 W) in steps of 200 W each one, for different depths of the channel (2.0 cm, 4.0 cm, 6.0 cm, 8.0 cm, 10.0 cm, and 11.0 cm), allowing that the system should reach the permanent conditions in each of the cases. The results showed the profiles of temperature of the different elements. It was concluded that the mass flow rate depends on the heat flow supplied to the system, as well as on the depth of the channel. Also it was observed that the efficiency of this system only depends on the amount of heat supplied.

Arce et al., (Arce et al., 2009) presented an experimental study of a full scale solar chimney outdoors. The solar chimney was not connected to any room to ease its independent study. The results showed that for a maximum irradiance of 604 W/m^2, about 13:00 hours, on September 15th, 2007, a maximum increase of the air temperature of 7 ºC was obtained. The system showed this day an average air flow rate of 177 m^3/h. It was observed that the air flow rate across the chimney is influenced by a difference of pressures between the entry and the exit caused by thermal gradients, and mainly by the outdoor wind speed.

Jiménez et al., (Jiménez et al., 2010) presented an experimental study of a full scale solar chimney outdoors. Data were recorded and analysed for one year (2007-2008). However, data of three days selected from this period were discussed, (one day for winter, one for autumn and one day near the beginning of summer). The authors concluded that, the air flow rate in a solar chimney is associated not only with the thermal effect and wind speed, but also with the combination of both effects on the system.

The purpose of the present study consists in continuing the previous work, with some modification on the same experimental passive system *(solar chimney)*. This study is supported by the experimental information acquired during one year.

2 PHYSICAL MODEL

Figure 1, shows a front view of the solar chimney in study. The above mentioned system has the following dimensions: 5.60 m, 1.20 m width and 0.52 m deep. The system is in vertical position and facing south. The main components are: an absorbing plate, thermal insulation, a glass cover, and wood around the lateral and back side.

The absorber plate has 0.15 m of thickness, 1.0 m of width and 5.0 m high, it was made of concrete, in which solar incident radiation is absorbed and stored. Thermal insulation was used to reduce heat losses from the absorber plate. The glass cover was 5 mm of thickness and was used to reduce convective and radiative losses to the environment, and at the same time it takes part of the air channel. Other accessories like wood around the lateral and back side, an air inlet in the low back side and an air exit at the top, are also included. In order to measure the main variables, such as air temperature and air velocity inside the channel, sensors and instruments were set up.

In order to measure the environmental conditions, a meteorological station and a system to record and process the information, are located close to the solar chimney. The experimental data of different variables are measured every second, but they are averaged and recorded every minute.

Figure 1. Photograph of the solar chimney.

A lateral view of the solar chimney is shown in Figure 2, as well as the instrumentation set up. Thirty thermocouples type “T”, previously calibrated were used to measure temperature, and nine hot wire anemometers (Mod. TSI-8475) were used to measure the air velocity.

With the purpose of avoiding a counter flow through the air channel, a driving air protection was set up at the top. The hood moves itself with the wind direction, which generates a fall of pressure near the exit and, at the same time, it helps to the air extraction. Thus, wind forces add to the thermal ones. In order to avoid air whirlpools at the entry of the solar chimney, a wooden box with some perforations was installed.

3 OPERATION GENERAL PRINCIPLE

The functioning of a solar chimney is originated when a difference of air pressure takes place between the entry and the exit, inducing an air flow through the system in positive or negative direction. Certainly, it is always intended to obtain positive differences of pressure, with the purpose of ventilating a house or a building.

4 RESULTS

Data recorded for one year (2014) were analysed. The information corresponds to the solar chimney of LECE at the Plataforma Solar of Almería in Spain. Data of three days from this period were selected, and will be discussed in the following. These days were selected with clear sky, and representative enough of the period attending to the levels of South Vertical Global Solar Radiation: High, moderate and low, near the beginning of winter, spring and summer respectively.

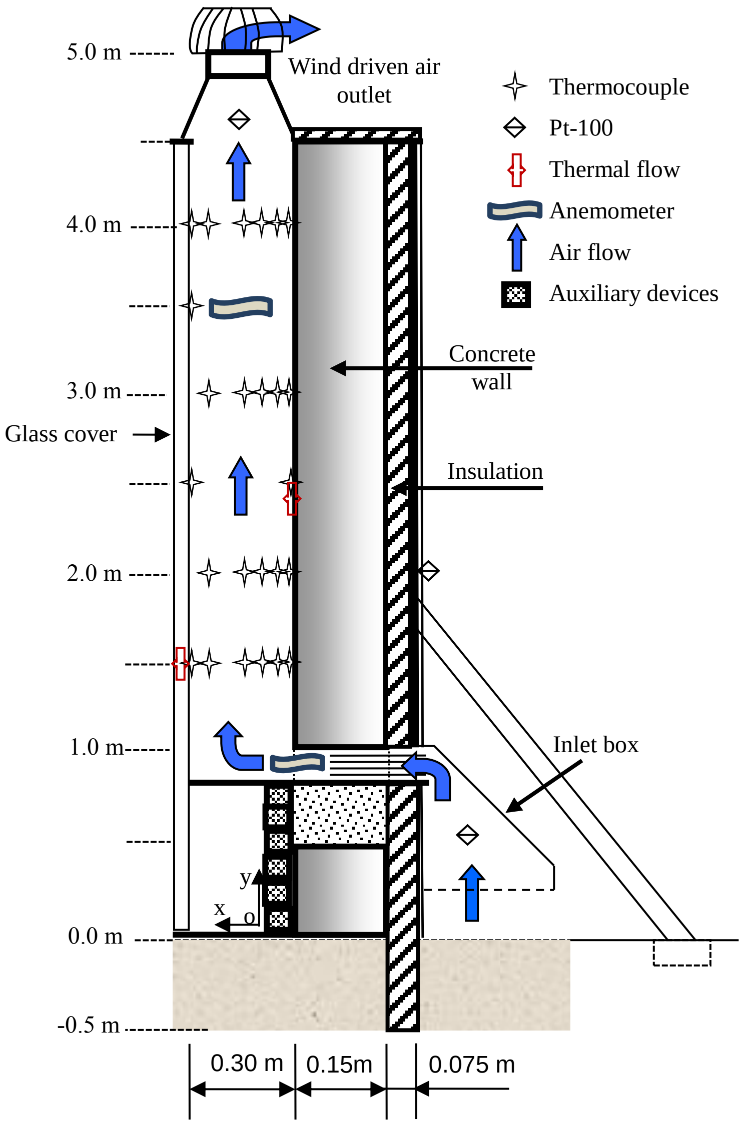


Figure 2. Lateral view of the solar chimney and the instrumentation set up.

4.1 Irradiance

Figure 3-a shows, from high to low intensity, the South Vertical Global Solar Radiation (SVGSR), the Horizontal Global Solar Radiation (HGSR), and the Diffuse Solar Radiation (DSR), on a clear day of winter (26/12/2014).

The SVGSR and the HGSR, on a clear day near the beginning of spring (05/03/2014), are shown in Figure 3-b. Both irradiances are quite near, due to the location and the day of the year. Furthermore, in Figure 3-b, the DSR is shown , which is smaller than that shown in Figure 3-a.

Near the beginning of summer (20/06/2014) HGSR, SVGSR and DSR, from high to low intensity, are shown in Figure 3-c. It is observed that the magnitudes of SVGSR and HGSR are inverted as the days and seasons of the year progresses.

The effects produced by the SVGSR, and by the (DSR), will be noticed on the surface temperatures of the absorber plate, in the air temperature in the channel flow and on the flow rate through the system. This will be discussed in the next paragraph.

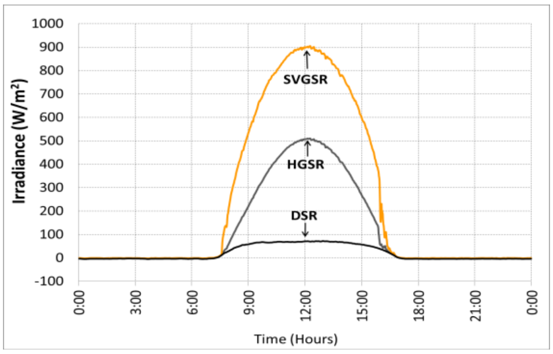


Figure 3-a. Solar radiation (VGSR, HGSR and DSR) on December 26th, 2014 (~winter).

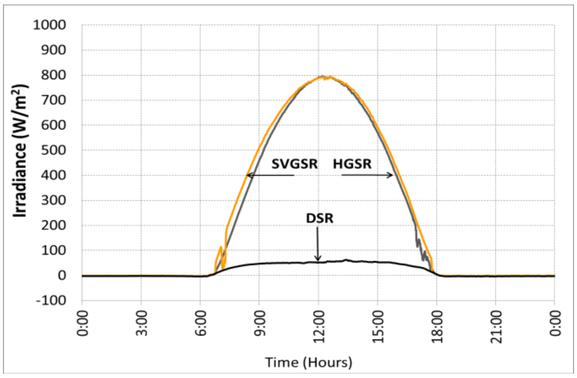


Figure 3-b. Solar radiation (SVGSR, HGSR, and DSR) on March 5th, 2014 (~spring).

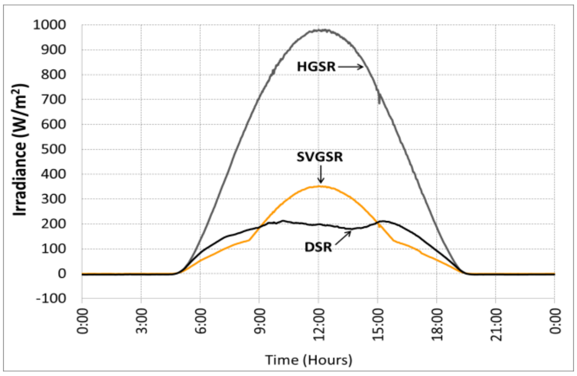


Figure 3-c. Solar radiation (HGSR, SVGSR and DSR) on June 20th, 2014 (~summer).

4.2 Wall temperatures

Similarly, for the same days, tendencies of the surface temperature for the absorber plate of the solar chimney are shown in Figure 4-a, Figure 4-b, and Figure 4-c. The surface temperature sensors are located at four different heights 1.5, 2.0, 3.0 and 4.0 m. Small differences in some cases and big ones are observed in others for each figure. For example, in Figure 4-a, the biggest differences of temperature among those tendencies are approximately 3 ºC. The highest value is the surface temperature located at 3.0 m, which is 52 ºC around 15:00 hours. The smallest values observed are approximately 15 ºC located between 6:00 and 8:00 hours. This fact produces a maximum surface temperature increment of approximately 37 ºC, because of the SVGSR received on the absorber surface.

Figure 4-b shows the surface temperatures on March 5th, 2014. The magnitudes of the trends are also very similar to each other, except that temperature at 4.0 m high, which is noticeably minor due to a shade on the zone of the surface caused by the top part of the chimney itself. Maximum values of 50 ºC between 14:00 and 15:00 hours and minimum values of approximately 17 ºC between 6:00 and 8:00 hours are observed. This fact produces a maximum surface temperature increment of approximately 33 ºC, 4 ºC lower than the case for winter due to the minor SVGSR received on the absorber surface.

Figure 4-c, shows the surface temperature on June 20th, 2014, trends are also very similar at each other, maximum values between 32 and 35 ºC around 13:00 hours are observed. The minimum values found are 24 ºC between 5:00 and 6:00 hours, this fact produces a surface temperature increment of 9.5 ºC approximately, due to minor solar radiation received (SVGSR) on the absorber surface.

The effect of the increases on surface temperatures in the absorber plate will be observed in the average increase of the air temperatures in the channel of the chimney, and at the same time in the air flow rate across the system. The above, will be described in the following paragraphs.

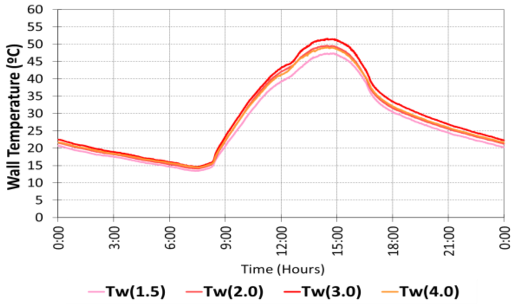


Figure 4-a. Wall temperatures at four different heights on December 26th, 2014 (~winter).

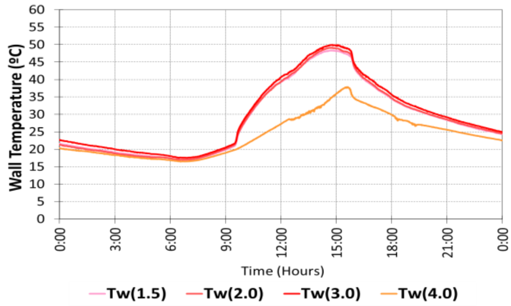


Figure 4-b. Wall temperatures at four different heights on March 5th, 2014 (~spring).

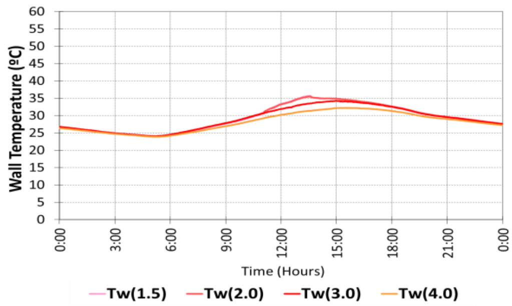


Figure 4-c. Wall temperatures at four different heights on June 20th, 2014 (~summer).

4.3 Air temperatures

The energy stored in the absorber surface in the solar chimney is transferred to the adjacent fluid, first by conduction and then by convection, so that, the greater difference between the average temperature of the surface and the temperature of the air at the entry, the greater will be the energy gained by the air inside the channel and the greater its increase of temperature. Therefore, on December 26th, 2014 (Figure 5–a), presents an average daily increase of 7.3 ºC, and a maximum increase of 14.0 ºC around 14:00 hours, approximately two hours after receiving the maximum solar radiation, due to thermal inertia of the system.

An average daily increase of 4.7 ºC is found on March 5th, 2014 (Figure 5-b) and 1.7 ºC on June 20th, 2014 (Figure 5-c). The above temperature increments will influence the air flow rate across the system, which will be discussed in the following paragraph.

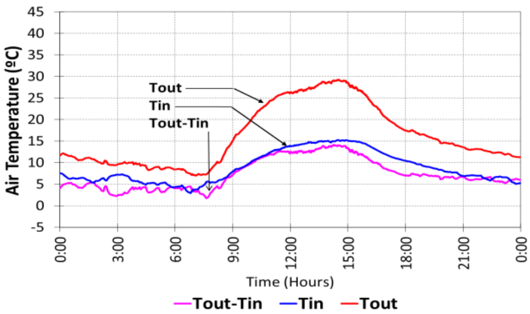


Figure 5-a. Air temperatures on December 26th, 2014 (~winter).

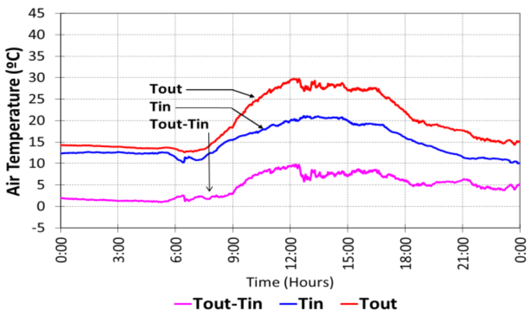


Figure 5-b. Air temperatures on March 5th, 2014 (~spring).

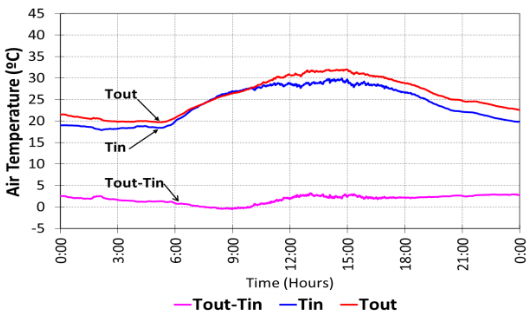


Figure 5-c. Air temperatures on June 20th, 2014 (~summer).

4.4 Air flow rate

In this section the air flow rate across the system is analysed, which is associated with the energy acquired while it is flowing along the channel of the chimney, and due to this fact it presents an associated increase of air temperature.

In the Figure 6-a, an average air flow rate of 91.5 m^3/h daily is obtained, with an outdoor wind speed of 1.4 m/s. In case of Figure 6-b, for the same schedule, there is an air flow rate of 96.5 m^3/h, and a corresponding averaged outdoor wind speed of 4.3 m/s. Analogously, in the Figure 6-c, in the same schedule, there is an air flow rate of 100.1 m^3/h, and a corresponding outdoor wind speed of 2.9 m/s.

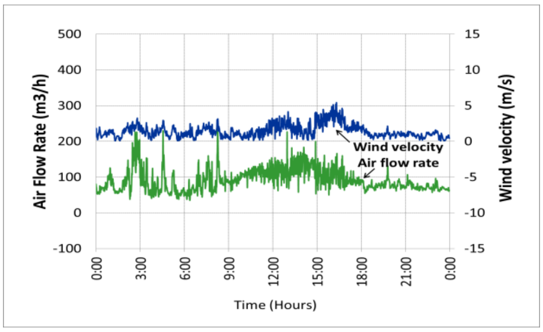


Figure 6-a. Air flow rate and outdoor wind velocity on December 26th, 2014 (~winter).

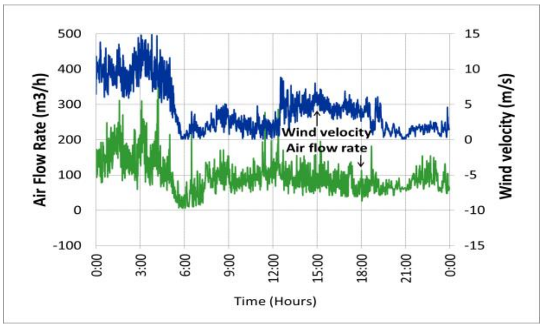


Figure 6-b. Air flow rate and outdoor wind velocity on March 5th, 2014 (~spring).

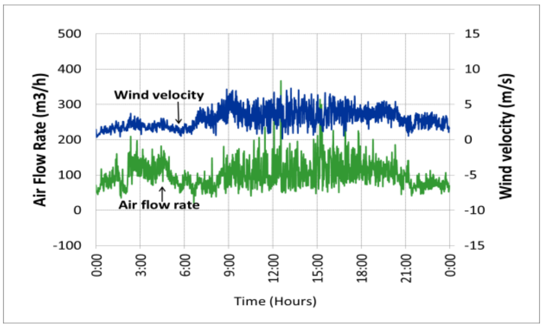


Figure 6-c. Air flow rate and outdoor wind velocity on June 20th, 2014 (~summer).

The results are summarized in Table 1, by the daily averaged values. The increase of the air temperature (ΔT_{air}), the mean air temperature (Tm_{air}), the outdoor wind speed 10 meters height (V_{wind}), and the Air Flow Rate (AFR). In the case of solar radiation, the maximum values are just at noon, in solar time.

Table 1: Average values from 00:00 hrs to 24:00 hrs

Parameter	Winter (26/12/2014)	~ Spring (05/03/2014)	~ Summer (20/06/2014)
Max.-SVGSR (W/m^2)	900	800	350
Max.-HGSR (W/m^2)	500	800	980
ΔT_{air} (ºC)	7.3	4.7	1.7
Tm_{air} (ºC)	12.5	17.2	24.8
V_{wind} (m/s)	1.4	4.3	2.9
AFR (m^3/h)	91.5	96.5	100.1

It was expected that the air flow rate was associated only with the increase of air temperature. Nevertheless we can realize that it is also influenced by the outdoor wind speed mainly, and also, it may be influenced by wind direction, which must be proved.

5. CONCLUSIONS

In spite of having a higher solar radiation (SVGSR) of 900 W/m^2 on the glass surface of the solar chimney in the considered winter day, and consequently of having a maximum increase in the temperature of the air in the channel of the chimney (7.3 ºC), we have a lower air flow rate in the system (91.5 m^3/h) with regard to the ~spring case (96.5 m^3/h) when there is a lower average increase daily of temperature (4.7 ºC) and there is lower average solar radiation (800 W/m^2) on the glass surface. The previous fact is attributed to the low daily average wind speed of 1.4 m/s in the first case, while in the second case such speed is 4.3 m/s. Whereas in ~summer with a wind speed of 2.9 m/s, a value between that one of ~spring and winter, an air flow rate of 100.1 m^3/h is obtained, remarkably higher than that one in winter and even noticeably higher that one in ~spring. This may be due to the higher mean air temperature (Tm_{air}=24.8 ºC) against 17.2 ºC and 12.5 ºC of ~spring and winter respectively.

According to the results, the air flow rate in a solar chimney is associated not only with the thermal effect and wind speed, but also with the combination of both effects on the system, and very probably with the wind direction.

6. ACKNOWLEDGEMENTS

The authors are grateful for the support of CIEMAT facilities at PSA in Spain, the support of Cenidet-TNM and the financial support of Conacyt-México for a stay abroad by a Mexican researcher.

7. REFERENCES

Martí, J. (2006). *Caracterización de una chimenea solar a través de parámetros físicos como sistema de ventilación natural*, Tesis doctoral. Departamento de Física de los Materiales, Facultad de Ciencias. Universidad Nacional de Educación a Distancia, Madrid-España.

Martí, J., Heras, M.R. (2006). *Dynamic physical model for a solar chimney,* Energy Efficiency in Building. Renewable Energy Department. Ciemat. Avd. Complutense 22, Madrid 28040, Solar Energy.

M. J. Jiménez. LECE-UiE3-CIEMAT Test Facilities. 2011. *Full scale test facilities for evaluation of energy and hygrothermal performances*. (A Janssens, S. Roels, L. Vandaele, ed.), ISBN 978-94-9069-584-2, BFG, UGent, Belgium.

Arce J., Jiménez M.J., Guzmán J.D., Heras M.R., Álvarez G., Xamán J. (2009). Experimental study for natural ventilation on a solar Chimney. *Renewable Energy*. 34 (12), 2928-2934.

Bansal N., Mathur R., Bhandari M. (1994). A study of solar chimney assisted wind tower system for natural ventilation in buildings. *Building and Environment*. 29 (4), 495-500.

Ben R., Bilgen E. (1991). Natural convection and conduction in Trombe wall systems. *International Journal Heat Mass Transfer*, 34-4/5, 1237-1248.

Bouchair A. (1994). Solar chimney for promoting cooling ventilation in southern Algeria. *Building Serv. Eng. Res. Technol*. 15 (2), 81-93.

Khedari J., Boonsri B., Hirunlabh J., (2000). Ventilation impact of a solar chimney on indoor temperature fluctuation and air change in a school building. *Energy and Building*. 32, 89-93.

Chen Z., Bandopadhayay P., Halldorsson J., Byrjalsen C., Heiselberb P., Li Y. (2003). An experimental investigation of a solar chimney model with uniform wall heat flux. *Building and Environment*, 38, 893-906.

Khedari J., Rachapradit N., Hirunlabh J. (2003). Field study of performance of solar chimney with air-conditioned building. *Energy*, 28, 1099-1114.

Chakraborty J., Fonseca E. (2005). *Analysis and evaluation of a passive evaporative cool tower in conjuntion with a solar chimney*. Plea2005-The 22nd Conference on Passive and Low Energy Architecture. Beirut, Lebanon. College of Architecture and Environmental Design, Arizona State University, Tempe Arizona, USA.

Emad H. (2006). Passive options for solar cooling of buildings in arid areas. *Energy*. 31, 1332-1344.

Chungloo S., Limmeechokchai B., (2007). Application of passive cooling systems in the hot and humid climate: The case study of solar chimney and wetted roof in Thailand. *Building and Environment*, 42, 3341-3351.

Burek, S.A.M., Habeb, A., (2007). Air flow and thermal efficiency characteristics in solar chimneys and Trombe Wall. *Energy and Building*, 39, 128-135.

Arce J., Jiménez M.J., Guzmán J.D., Heras M.R., Alvarez G., Xamán J. (2009). Experimental study for natural ventilation on a solar chimney. *Renewable Energy*. 34, 2928-2934.

Jiménez M.J., Guzmán J.D., Heras M.R., Arce J., Xamán J., Alvarez G., (2010). *Thermal performance of a natural ventilation system*. ASME-2010, Phoenix, Arizona, U.S.A.

CONTROL STRATEGIES OF THE NATURAL VENTILATION FOR PASSIVE COOLING FOR AN EXISTING RESIDENTIAL BUILDING IN MEDITERRANEAN CLIMATE

Guido Raffaele Dell'Osso[1], Francesco Iannone[2], Alessandra Pierucci[3], Alessandro Rinaldi[*4]

1,2,3 DICATECh - Department of Civil, Environmental, Building Engineering and Chemistry
Polythecnic of Bari,
via E. Orabona, 4 - 70125 Bari (BA) - ITALY

4 DICATECh - Department of Civil, Environmental, Building Engineering and Chemistry
Polythecnic of Bari,
via E. Orabona, 4 - 70125 Bari (BA) - ITALY
**Corresponding author: alessandro.rinaldi@poliba.it*

ABSTRACT

Natural ventilation is increasingly considered one of the most efficient passive solutions to improve thermal comfort in buildings. However in order to support its planning and implementation, quantitative analysis on airflow paths and heat-airflow building interactions are needed. This requires an adequate accounting of both internal effects, from building layout and structure, and external forcings from atmospheric factors.
This paper has dealt to analyze the potential of building automation systems for ventilative cooling of residential buildings.
The case study focused on a Italian typical building of the '60s, situated in the Mediterranean climatic context (Bari - Italy, 41° 07'31 "N 16 ° 52'00" E, 5 m asl), with windowed sides faced to northwest and southeast.
Various operating controls of ventilation have been developed to reduce energy consumption for cooling, ensuring adequate levels of indoor comfort. In particular, automated bottom hinged windows have been hypothesized.
Different design solutions have been simulated in order to choose the optimal control of ventilation in relation to internal and external temperature and humidity (relative and absolute).
The air flows in the building have been calculated with a multizone airflow model, performed by TRNFLOW within the TRNSYS software.
Thermal comfort analysis, according to the adaptive thermal comfort theory (EN 15251-2007), have shown that a natural ventilation system, calibrated on a variable set-point based on the optimal temperatures (according to the theory of adaptive comfort) determines a significant reduction of overheating during the occupation hours.
Despite the benefits of such control logic, it has been necessary to provide a second control logic, calibrated on internal and external humidity (relative and absolute). The simulations results have underlined a significant reduction in relative humidity levels (generally in range 40- 60%), although some undercooling occurs due to the opening of the windows when the control on the temperature is set off.
Others simulations have regarded the combination of the above described natural ventilation system, with an air to air heat pump. The simulation results have showed that the control strategies of ventilation for passive cooling enable significant reduction in energy consumption.
This study have underlined that ventilation strategies for passive cooling of existing buildings in Mediterranean climate, can contribute even more effectively to the improvement of the behavior of the building envelope,

integrating or replacing the conventional efficiency strategies, if properly integrated with adequate control systems.

KEYWORDS

Natural ventilation, thermal comfort, passive cooling, multizone airflow models, building automation

1 INTRODUCTION

Natural ventilation could significantly reduce building energy consumption for cooling and improve thermal comfort with the indoor environment (Brager et al., 2011).
Despite the natural ventilation is necessary to ensure adequate Indoor Air Quality levels and to dilute the pollutants originating in the building, it has the potential to save significant amounts of energy by reducing the demand for mechanical ventilation and air conditioning (Rowe, 1996; Zhao and Xia, 1998).
In residential buildings the ventilation is generally manual and not always aimed at cooling needs. Building automation systems and suitable control logics based on the building profiles of use and on the comfort performance are required.
In order to value indoor air temperatures in buildings, it is essential to have suitable methods to predict and valuate ventilation performance. The prediction of natural ventilation effects is complicated by the dynamic nature of ventilation itself; variables include change of wind speed and direction as well as the context, more or less densely populated, and the influence of users behaviour on the ventilation through building openings (Yan Li and Xiaofeng Li, 2014).
Several building simulation are able to integrate building's thermal model with a multi-zone airflow network model. We selected the TRNFLOW tool within the TRNSYS software, because it allows to simulate different control strategies for the building automation.
The present paper analyses various operating controls of ventilation as retrofit solutions to improve thermal comfort in an existing residential building located in a suburban zone of the Bari's city (Italy).

2 METHODOLOGY

The following study has focused on the potential of Building Automation Systems (BA) for the control of ventilation for passive cooling of buildings.
Several control strategies of natural ventilation have been simulated to reduce energy consumptions for cooling and ensure adequate levels of indoor comfort.
After setting up the building's thermal model and the multi-zone airflow network model within the TRNFLOW–TRNSYS software, different ventilation strategies have been compared through:

- thermal comfort analysis, according to the standard UNI EN 15251, assuming the category n. II (relative to new construction and existing buildings subject to refurbishment)
- energetic analysis in dynamic regime.

Simulations are performed during the cooling season (1 June – 30 September) to analyse the passive behaviour of the building.
In particular, relatively to the occupation hours, the discomfort due to overheating and undercooling has been calculated, in reference to the upper and lower temperature limit.
In relation to internal and external temperature and humidity (relative and absolute), four design solutions have been simulated in order to choose the optimal control of ventilation. Three cases are performed without any active cooling system. In a case (Case 4) the

combination of a natural ventilation system with an air to air heat pump has been simulated to evaluate the reduction of cooling energy consumptions.

2.1 Building Description

The case study has regarded a multi-family residential building typical of the '60s. The apartment analyzed situated at the intermediate floor has a net floor area of about 100 m^2 and is characterized by:

- windowed sides faced to northwest and southeast;
- compactness index (ratio between enveloping surface and heated volume) equal to 0.65, as result of the building type;
- bedroom, kitchen and bathroom faces to northwest, and living/dining/ room, bedroom and study room faces to southeast (fig.1)

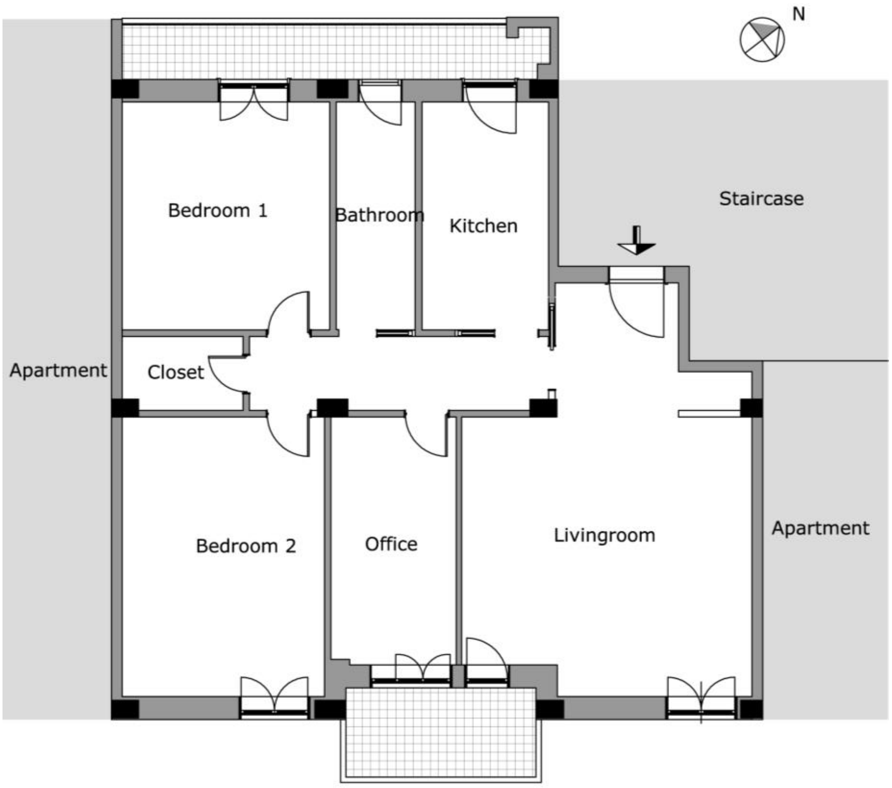


Figure 1: Apartment floor plan

The building envelope parameters (tab.1) were determined according to the typical envelope for the Italian residential buildings of '60s:

Table 1: Thermal characterstics of the envelopes

Items	U-value ($W/m^2 K$)
External wall	1,10
Internal wall	1,54
Roof	0,83
Window	5,6

In order to consider the *internal heat gains* for occupancy, lighting and domestic appliances, several typical schedules of a residential building have been implemented in the TRNSYS software.

None *solar shielding* system has been adopted in the various simulations for not alter the flow air when the windows are open and it will be the object of future studies. The adoption of solar shielding systems will be deepened in later stages of the search.
The type of windows in the rooms are *tilt-turn windows*, with possible bottom hinged opening automated.

2.2 Thermal - Multizone Airflow Simulation Model

The building thermal model has been set up in TRNSYS v.17 and the multi-zone air flow model has been integrated with TRNFLOW.
TRNFLOW integrates the multi-zone air flow model COMIS into the thermal building module of TRNSYS (Type 56). With TRNFLOW the air flows between airnodes (coupling), from outside into the building (infiltration) and from the ventilation system (ventilation) can be calculated.
Multizone air flow models idealize the building as a network of nodes and airflow links. The nodes represent the rooms and the building surrounding. The links depict openings, doors, cracks, window joints and shafts as well as ventilation components like air inlets, outlets, ducts and fans. The wind pressures on the façade and the indoor and outdoor air temperatures are the important boundary conditions.

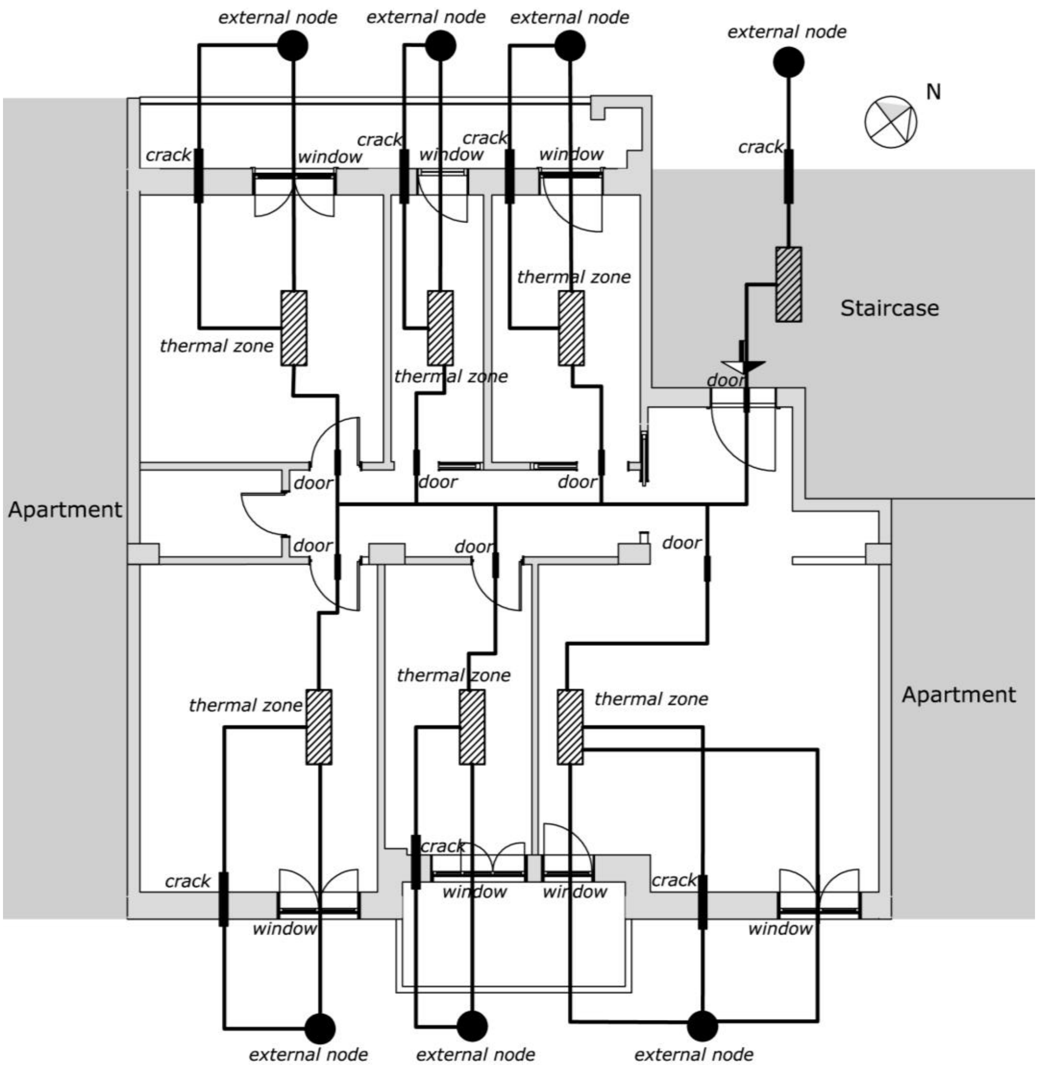


Figure 2: Network model of the building

The air permeability characteristics used in the various simulations that describe a low air tightness of the building envelope are shown in the tab.2

Table 2: Air permeability characteristic of building envelope

Item	Air Mass Flow Coefficient Cs (Kg/sPa) crack (Kg/s m Pa) large opening	Air Flow Exponent n	Discharge Coefficient Cd
Crack _External Wall	0,00002	0,85	-
Large Opening _Window	0,0003	0,6	0,6
Large Opening _Door	0,0015	0,6	0,6

Weather data for the city of Bari are used and wind speed profile has been modified by terrain roughness parameters for suburbs.
In the various simulations the air change rate depends on the logics of window opening and on the air permeability of windows. In particular, to ensure conditions of Indoor Air Quality (IAQ), it has been assumed a schedule opening-windows at certain hours (8 a.m. - 10 a.m.; 1 p.m.- 2 p.m.; 8 p.m. - 9 p.m.). These periods correspond to the activities of preparing and cooking foods and of household cleaning. During these hours it has been hypothesized a bottom hinged opening. Thus, the *opening factor of window* has been multiplied for a factor (Ck) depending on the maximum window tilt according to the UNI EN 15242 – 2008.

Table 3: Opening window parameters

Item	Maximum window tilt	Opening Coefficient Ck
Window_(h=2,2 mt)	13°	0,22
Window_(h=1,2 mt)	25°	0,39

The IAQ opening during these hours, managed directly by the user and independent by automation systems, it is the same in all the cases simulated.

2.3 Natural Ventilation Strategies and Automation Systems

Energy efficiency solutions have involved the installation of a network of sensors (wireless low-power) and actuators for the implementation of natural ventilation strategies.
In order to keep a comfortable temperature and humidity, four control strategies of natural ventilation are simulated. Opening actuators can be applied to existing windows, controlled by temperatures and humidity sensors.
Except Case 0 (with opening windows only for IAQ and no cooling system), and the Case 5 (equipped with an air to air heat pump for cooling, that operate only in presence of users with setpoint temperature of 26 °C), during the hours not included for IAQ, the windows are opened if:

Case I (actuators operated by temperature sensors)

- T indoor > T optimal (valuated according UNI EN 15251);
- T indoor – 3°C < T outdoor < T indoor (in order to avoid undercooling discomfort).

Case II (actuators operated by temperature or humidity sensors)

- T indoor > T optimal;
- T indoor – 3°C < T outdoor < T indoor.

  Or if:
- R.H. indoor (relative indoor humidity) > 70%;

- absolute indoor humidity > absolute outdoor humidity.

Case III (actuators operated by temperature and humidity sensors)

- T indoor > T optimal;
- T indoor – 3°C < T outdoor < T indoor.

  Or if:
- R.H. indoor (relative indoor humidity) > 70%;
- absolute indoor humidity > absolute outdoor humidity.
- T indoor > T optimal;

Case IV (hybrid system with cooling system)

- T indoor > 26°C;
- T indoor – 3°C < T outdoor < 26°C.

  The operation of air to air heat pump for cooling, during the hours of occupation, if:
- T indoor > 26°C;
- T outdoor > 26°C.

2.4 Results

The first three cases proposed have been compared in terms of adaptive thermal comfort and relative humidity conditions.
According the European standard UNI EN 15251, three comfort categories limited by three temperatures ranges are defined. Thermal comfort is evaluated on the difference between the optimal operative temperature and the simulated operative temperatures. The operative temperatures outputs during the occupation hours are compared with the Upper Temperature limit and Lower Temperature limit.
Simulation results have shown the efficacy of the proposed ventilative cooling strategies. A natural ventilation system, calibrated on a variable set-point based on the optimal temperatures (according to the theory of adaptive comfort) determines a significant reduction of overheating during the occupation hours.
The Bedroom2 has presented the more situations of discomfort for overheating. The orientation non-optimal and the absence of any shielding system, the night- ventilation lack and the high internal gains are the main causes. Furthermore the simulations showed high levels of relative humidity (>70%) in the bedrooms.
In order to reduce the high level of relative indoor humidity (>70 %), a natural ventilation strategy controlled by humidity sensor has been necessary. In fact in the Case 2, when the control of relative humidity is independent by indoor temperature, the discomfort situations for high levels of relative humidity have halved, although some undercooling occurs due to the opening of the windows when the control on the temperature is set off. In the other cases relative humidity discomfort percentages were almost the same.

Table 4: Thermal comfort simulation percentages.

Room	Case 0		Case 1		Case 2		Case 3	
	Overheat.	Undercool.	Overheat.	Undercool.	Overheat.	Undercool.	Overheat.	Undercool.
Bedroom1	2,3 %	3,7%	0,0 %	3,9%	0.0 %	9,3%	0,0 %	4,4%
Bedroom2	10,7%	0,9%	2,5%	0,9%	4,7%	5,4%	2,3%	1,6%
Study room	9,7%	1,6%	3,5%	1,6%	5,4%	5,1%	3,4%	2,0%
Living room	10,7%	1,2%	2,9%	1,3%	5,0%	4,7%	2,9%	1,8%

Table 5: Relative Humidity discomfort percentages (R.H.>70%)

Room	Case 0	Case 1	Case 2	Case 3
Bedroom1	50,3 %	51,3%	23,3%	50,4%
Bedroom2	37,1%	38,6%	17,2%	37,9%
Study room	10,0%	11,7%	4,6%	10,7%
Living room	15,6	18,9%	3,8%	17,7%

Activation system controlled by humidity and temperatures sensors with logics above described (Case 3) have allowed significant overheating discomfort reductions as shown in following figures.

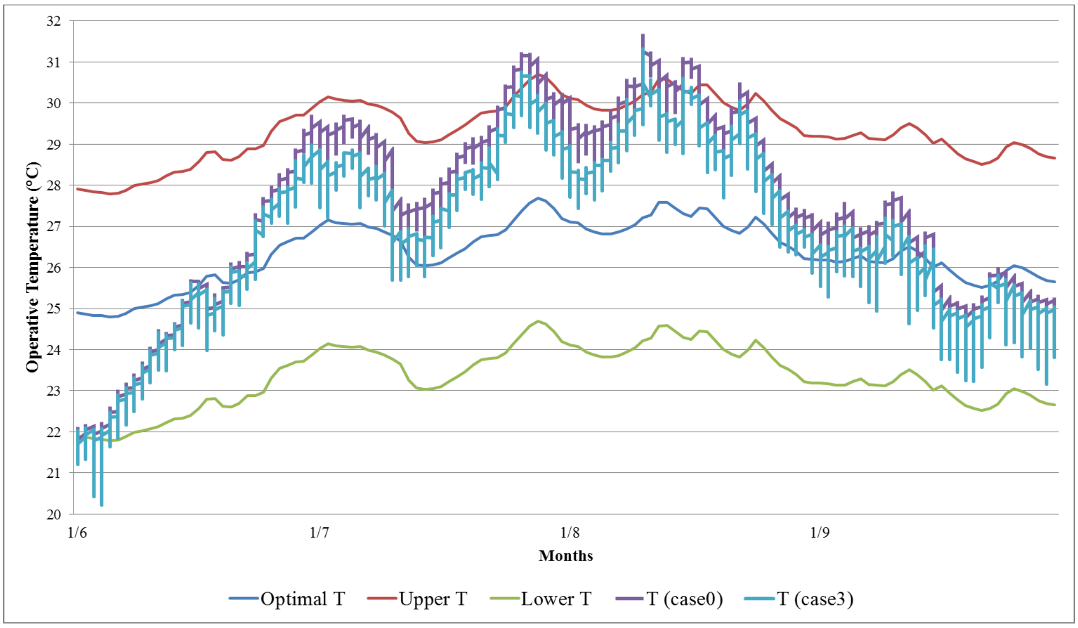


Figure 3: Thermal comfort simulation results – Bedroom2 (Case 0 – Case 3)

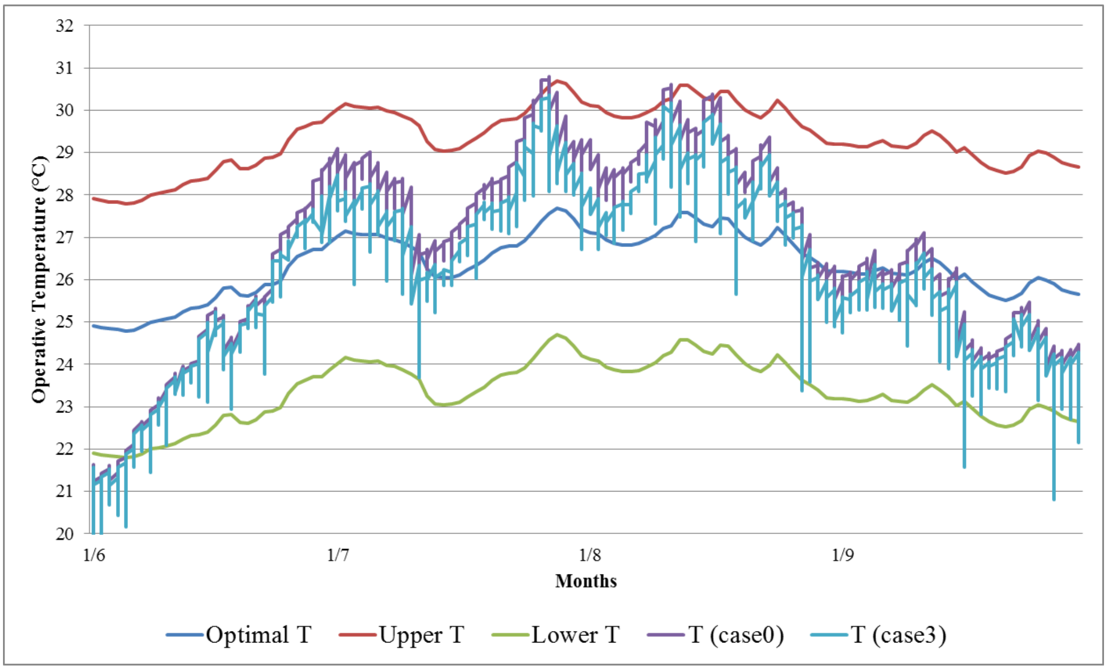


Figure 4: Thermal comfort simulation results – Bedroom1 (Case 0 – Case 3)

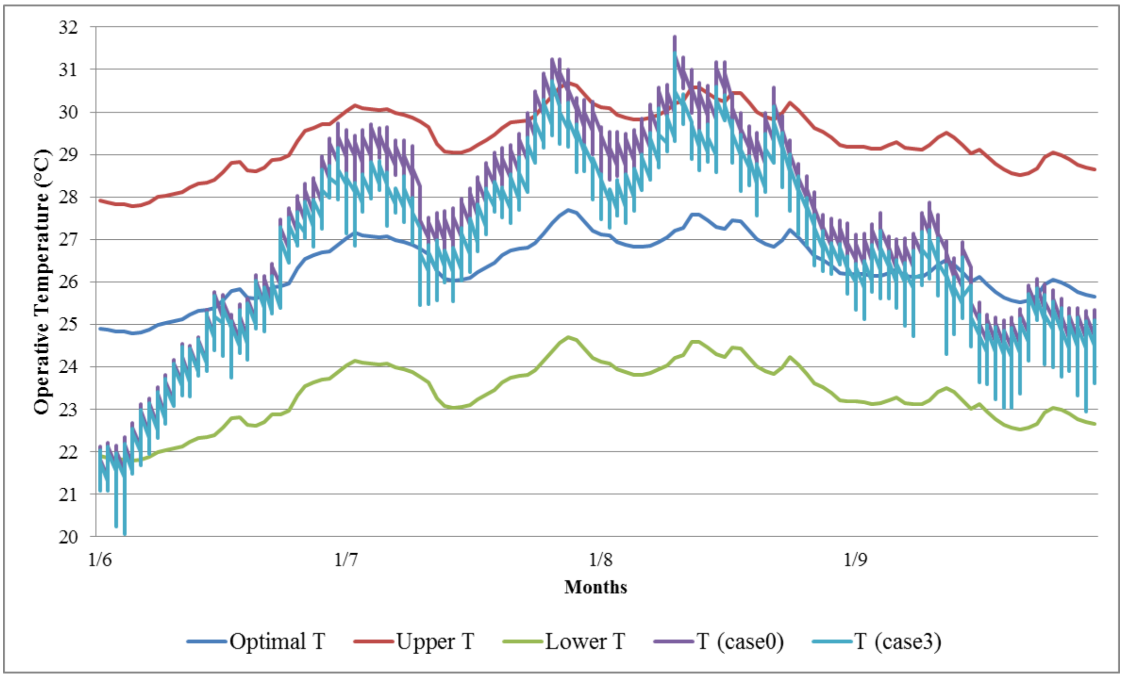


Figure 5: Thermal comfort simulation results – Living room (Case 0 – Case 3)

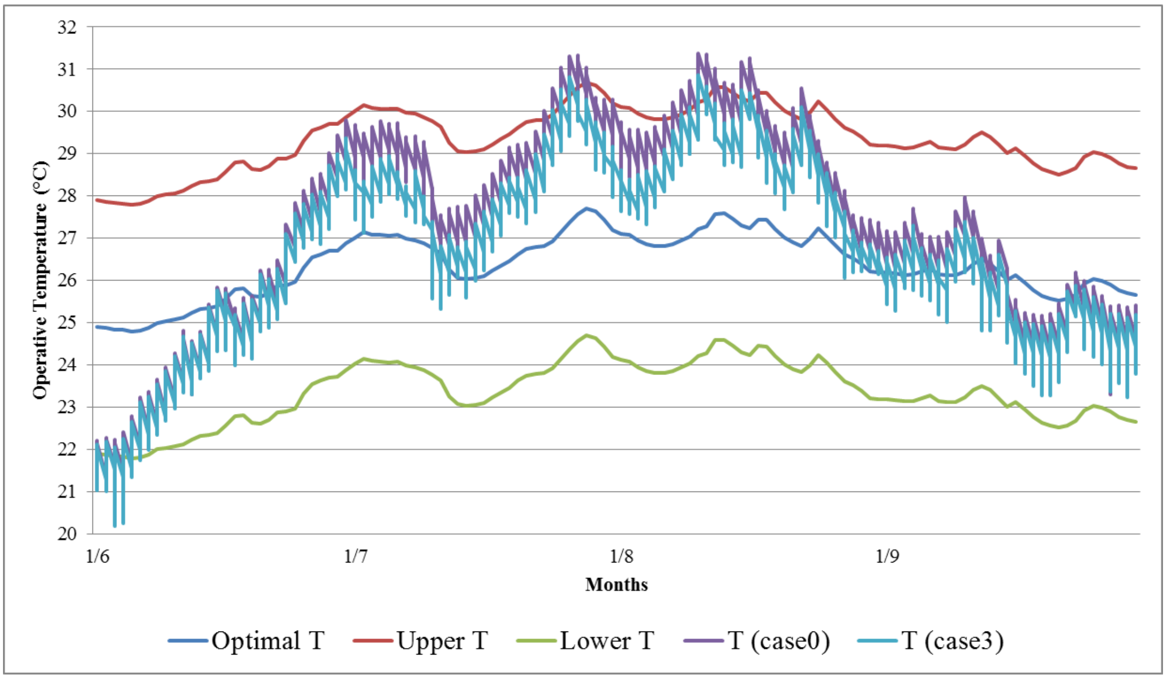


Figure 6: Thermal comfort simulation results – Study room (Case 0 – Case 3)

For the Case 4, where natural ventilation strategies are combined with air to air heat pump of cooling system, energy needs for cooling have been evaluated and compared with the Case 5. The simulation results have showed that the control strategies of ventilation for passive cooling enable a 50% reduction of energy consumption, from 905 kWh to 445 kWh, in similar comfort conditions.

3 CONCLUSIONS

In this paper the potential of natural ventilation in an existing residential building in Mediterranean climate has been studied.

With logics based on temperature and humidity, four strategies have been proposed to ensure adequate comfort conditions. Air flow dynamic simulation models have been created in order to choose the optimal control of ventilation. Assuming a likely occupation profile for the various rooms, adaptive thermal comfort analyses have been conducted according UNI EN15251.

The natural ventilation activation logics for passive cooling allow significant reductions in hours of discomfort due to overheating. To reduce the conditions of discomfort induced by high humidity levels, sensors for humidity control are necessary that activate the windows opening.

The study have underlined that ventilation strategies for passive cooling, can contribute even more effectively to the improvement of the behaviour of the building envelope, integrating or replacing the conventional efficiency strategies, if properly integrated with adequate control systems. With low investment costs the natural ventilation could reduce the high energy consumptions of cooling systems

Passive cooling through automatically controlled natural ventilation could allow the reduction of energy consumptions for cooling, reducing peak cooling power.

Future developments will concern the integration of control strategies with those relating to solar shielding systems.

4 REFERENCES

Brager, G., Borgeson, S. (2011). *Comfort standard and variation in exceedance for mixed-mode buildings.* Build Res Inf, 39(2), 118-33.

Rowe, D., (1996). *Mxed mode climate control: some hands-on experience.* AIRAH Journal, 50 (12), 19-23

Zhao, R., Xia, Y. (1998). *Effective non-isothermal and intermittent air movement on human thermal responses.* In: Proceedings of 6th International Conference on Air Distribution in Rooms, Stockholm, Sweden, pp.351-357.

Willmert, T. (2001). *The return of natural ventilation.* Architectural Record, 189 (7), 137-146.

Krishan, A., Baker, N., Yannas, S., Szokolay, S. (2001). *Climate responsive Architecture: a design Handbook for Energy Efficient Buildings.* New Delhi: Tata McGraw – Hill Education.

Yan, L., Xiaofeng, L., (2014). *Natural ventilation potential of high.rise residential buildings in northern China using coupling thermal and airflow simulations.* Building Simulation. vol.2, issue 1, 51-64.

Lion, L., Belleri, A., Lollini, R., Zardi, D., Giovannini, L. (2012). *Natural ventilation strategy potential analysis in an exisitng school building.* AIVC – Tightvent conference, Copenhagen.

Belleri, A., Lollini, R., Dutton, S M., (2014). *Natural ventilation design: an analysis of predicted and measured performance.* Building and Environment, 81, 123-138.

TRNFLOW Manual, A module of an air flow network for coupled simulation with TYPE 56 (multi-zone building of TRNSYS).

UNI EN 15251 (2007). *Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics.*

EVAPORATIVE COOLING AND VENTILATION CONTROL STRATEGIES FOR A KINDERGARTEN IN MEDITERRANEAN CLIMATE

Graziano Addante[1], Francesco Iannone[2], Alessandro Rinaldi[*3]

1,2 DICATECh - Department of Civil, Environmental, Building Engineering and Chemistry
Polythecnic of Bari,
via E. Orabona, 4 - 70125 Bari (BA) - ITALY

3 DICATECh - Department of Civil, Environmental, Building Engineering and Chemistry
Polythecnic of Bari,
via E. Orabona, 4 - 70125 Bari (BA) - ITALY
**Corresponding author: alessandro.rinaldi@poliba.it*

ABSTRACT

Aim of this work has been to determine the effectiveness of evaporative cooling and ventilation control strategies on a case study to ensure an adequate combination between energy efficiency and high levels of indoor comfort.

The case study has been a kindergarten, situated in the context of the climate continental Mediterranean area (Cerignola, Italy, 41°16'00"N, 15°54'00"E, 120 m asl), oriented on an east/west axis, classrooms south faced, and the services zone to north.

Several strategies for passive and hybrid cooling of the classrooms have been simulated in order to reduce the overheating in the summer season and to reach high levels of air quality.

Different solutions have been simulated to evaluate the optimum operative conditions, through the TRNSYS simulation software.

The first design strategy have regarded a night hybrid ventilation (from 10 p.m to 6 a.m.). The thermal comfort analysis, according to the adaptive thermal comfort (EN 15251-2007), have shown the reduction of overheating and the need to introduce a control logic, in relation to the outdoor temperature, to reduce the undercooling in the early hours of occupancy.

It has been necessary to provide a ventilative cooling strategy also during daytime in order to obtain a significant reduction of overheating.

The second design strategy has involved the adoption of a direct evaporative cooling system. The simulation have concerned different combinations of evaporation efficiency and air flow rates. The results have shown the optimal air flow rate for different evaporation efficiencies. Analysis on relative humidity levels have shown that the evaporative cooling system did not significantly alter the levels of relative humidity and there was a positive increase in the minimum levels of relative humidity.

This study have underlined that passive cooling systems, operated by a suitable control logic, can provide performance levels almost comparable with those of an air conditioning system, ensuring a significant reduction of energy consumption.

KEYWORDS

Natural ventilation, thermal comfort, passive cooling, evaporative cooling, building automation

1 INTRODUCTION

The increased usage of air conditioning systems in buildings has caused an higher energy consumption, an increase in peak electricity demand and significant environmental impact such as global warming and ozone depletion (M. Santamouris, 2007).

As a result, the recovery of natural techniques of cooling can be sufficient to restore conditions comfort inside buildings, reducing or eliminating the use of conventional mechanical cooling systems.

Passive cooling systems are one of the well-known strategies that can maintaining indoor comfort reducing energy consumption related to space cooling. Generally, these strategies consist in the solar heat gains control and in the internal heat gains reduction by altering building envelope elements. Some passive cooling strategies aim at lowering the indoor temperatures by utilizing natural heat sinks such as ambient air, sky, ground, and water (B. Givoni, 2011).

Natural ventilation is considered one of the most effective passive cooling techniques to reduce energy consumption of buildings and improve indoor air quality.

There were few studies, especially experimental work, that have evaluated the potential of passive cooling techniques in hot climates. For example, Bajwa (Bajwa, 1993) et al. have investigated the effectiveness of passive **evaporative cooling techniques** through experiments on two-story full-scale test house in the eastern region of Saudi Arabia.

Passive evaporative cooling is typically suitable for hot and dry climates. The passive cooling system is able to cool the indoor spaces with minimal use of energy (Alaidroos, 2015).

In this paper two passive cooling techniques have been evaluated including natural ventilation systems, in particular night cooling ventilation, and an evaporative cooling system, as retrofit solutions to improve thermal comfort in a kindergarten located in a rural zone of the Southern Italy.

2 METHODOLOGY AND PHASES OF THE WORK

This study has focused on evaporative cooling and ventilation control strategies to achieve high levels of energy efficiency and indoor comfort during summer period for a kindergarten located in Cerignola (Italy- Apulia).

In the first phase building thermal and visual performances have been evaluated to identify the zones with worse conditions, through the software Ecotect (Autodesk). Thermal comfort analysis have been conducted according to the standard UNI EN 15251. This standard defines an upper and lower comfort temperature related to the external conditions.

In particular, relatively to the occupation hours, the discomfort due to overheating and undercooling has been calculated, in reference to the upper and lower temperature limit.

The analyses focused on the southern building part, where the classrooms are located, that are characterized by the higher percentage of indoor thermal discomfort.

In the second phase, the thermal model has been implemented within the TRNSYS software and different design solutions are simulated for energy savings and for improve thermal comfort conditions. At first, ventilation control strategies has been hypothesized due to high thermal loads and the high levels of Indoor Air Quality (IAQ) required for kindergartens. A night-hybrid ventilation has been simulated for the building, occupied only in the daytime and with high inertia of the opaque components such as the roof slab.

Despite the benefits from the night ventilation system, it has been necessary to provide a more efficient cooling strategy under daytime. An evaporative cooling system has been chosen due to the high external air temperatures and due to the low external relative humidity. Several

evaporative cooling strategies were simulated by varying the characteristic parameters such as efficiency and air flow rates.
Finally, regarding the optimal design of the evaporative cooling system, internal relative humidity levels were checked to assess whether they were acceptable.

3 CASE STUDY

The case study has regarded an existing kindergarten (under completion), situated in a rural context of Cerignola (southern Italy).
It is subdivided in three sections (baby aged 3-12 months, 12-24 months, 24-36 months), each articulated in classrooms, quiet rooms, play activities room and services area.
The building has one basement and one floor above ground, 6 meters high, floor area of about 795 m^2 and green zone of 1100 m^2. It is characterized by (fig.1):

- windowed sides facing to south (classrooms) and to north small (services area);
- compactness index (ratio between enveloping surface and heated volume) equal to 0.64, as result of the building type;
- an overhang to south that shields the window areas (fig.2);
- a horizontal skylight placed on ceiling of classrooms.

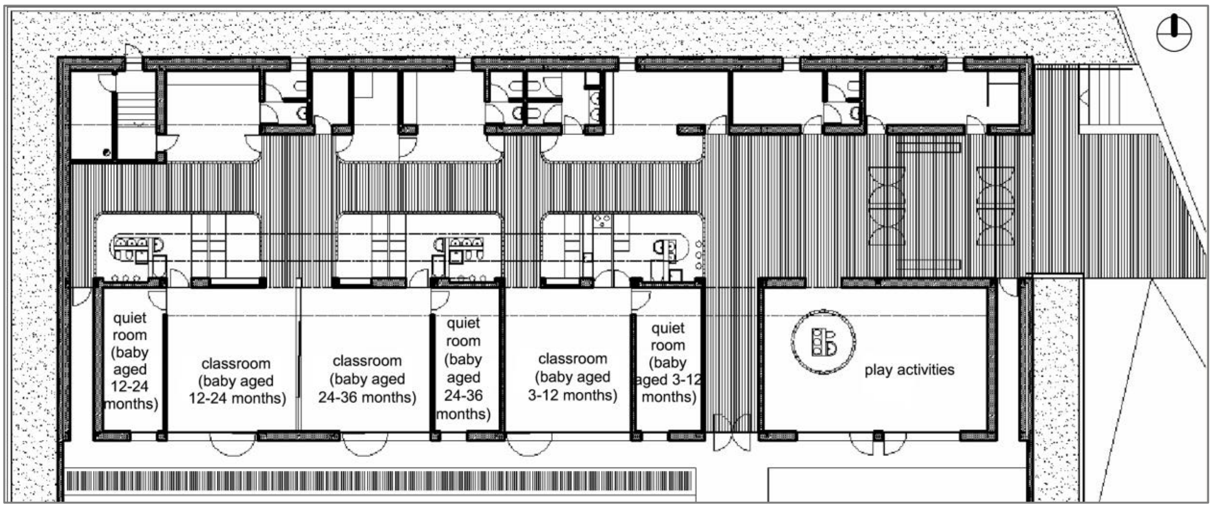


Figure 1: Ground floor plan

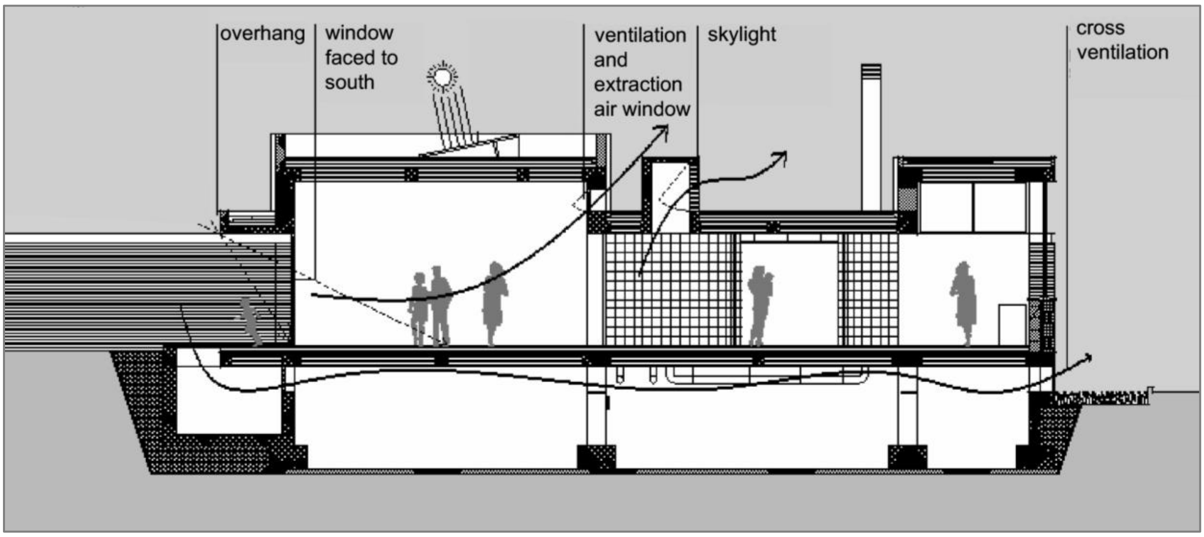


Figure 2: Sections of the building

Thermal characteristics of the building envelope are shown in the following table (tab.1):

Table 1: Thermal characterstics of the envelopes

Items	Thickness (cm)	U-value ($W/m^2 K$)	Thermal lag (hr)
External northern wall	57,5	0,37	18,7
External southern wall	41,5	0,39	12,5
Roof	45,0	0,46	13,1
Ground floor	38,5	0,48	11,8
Window	4-12-4	2,90	-

As regard the HVAC systems, there is no cooling systems or mechanical ventilation plants.

3.1 Internal Thermal Loads of the Building Model

In order to consider the *internal heat gains* for occupancy, lighting and equipment, several typical schedules have been implemented in the TRNSYS software. In particular, according to Edyth Boyd (E. Boyd, 1935), the internal heat gains (IHG) due to the baby has been estimated on height, weight and body surface area (BSA):

$$\text{BSA } (m^2) = 0{,}0003207 * \text{height (cm)}^{0,3} * \text{weight (gr)}^{(0,7285 - (0,0188*\text{LOGweight(gr)})} \quad (1)$$

$$\text{IHG (w)} = \text{metabolic rate} * \text{BSA} \quad (2)$$

For determine height and weight data of babies, reference was made to the anthropometric data provided by medical studies, particularly those of Tanner, Whitehouse and Takashi.
The metabolic rate has been estimated according to ISO 7730:2005 for kindergartens.
The following table summarizes the thermal loads determined by each occupant in relation to the type of activity.

Table 2: Internal Heat Gains

User	Height (mt)	Weight (kg)	B.S.A. (m^2)	Activity	Metabolic rate (W/m^2)	I.H.G. (W)	Sensible I.H.G. (60%) (W)	Latent I.H.G. (40%) (W)
Baby aged 3-12 months	0,67	7,65	0,39	classroom	81	32,16	19,30	12,86
				quiet room	66	26,20	15,72	10,48
Baby aged 12-24 months	0,81	11,35	0,53	classroom	81	42,93	25,76	17,17
				quiet room	66	34,98	20,99	13,99
Baby aged 24-36 months	0,89	13,5	0,60	classroom	81	48,76	29,26	19,50
				quiet room	66	39,73	23,84	15,89
Teacher	1,70	60,0	1,60	classroom	93	148,80	89,28	59,52

The profiles of occupation (fig.3) in the various rooms are determined according to other studies present in literature and to the data provided by teachers.

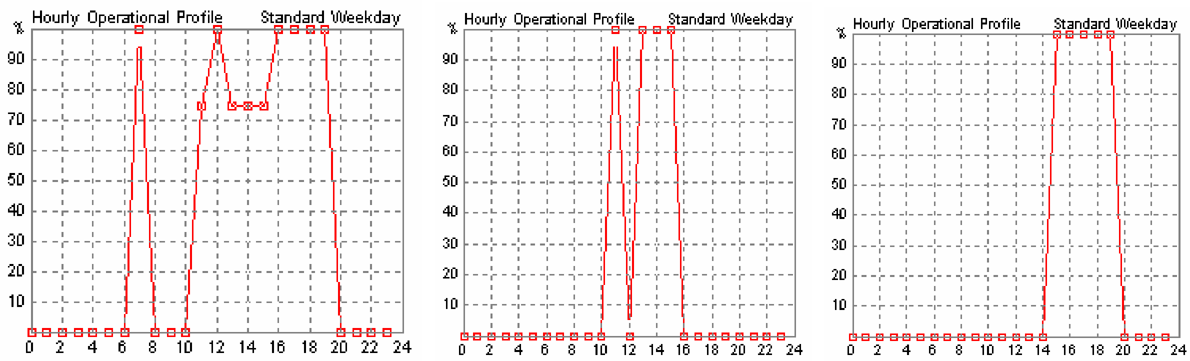


Figure 3: Hourly profile of occupation (classroom, quiet room, common space for recreational activities)

The three classrooms (for babies aged 3-12, 12-24 and 24-36 months) are attended at most by 10 and 20 babies respectively. The common space for recreational activities can host at most 8 babies.

For estimating thermal loads due to *lighting*, high efficiency fluorescent tube have been hypothesized with switching on from 5 p.m. to 7 p.m.

Regarding *ventilation* and *infiltration*, air changes have been defined equal to 0.004 m3/s*person, as defined by the UNI 10339: 1995, and a constant infiltration of 0,5 ACH.

4 CONTROL STRATEGIES FOR VENTILATIVE COOLING

Through the Ecotect software, a thermal-loads analysis of the building and their impact was carried out to identify the most critical elements of the project and provide suitable alternative solutions.

The analysis of heat flows and the internal operating temperatures have shown that the daytime heat gains, responsible of rising temperatures themselves, remained stored within the environment.

It resulted as high levels of discomfort are reached in the classrooms (above all the classroom of baby aged 24-36 months). Indeed this room is characterized by maximum number of occupants, equal to 20, and thermal loads greater for effect of a larger "body surface area", as previously described.

So in this paper cooling strategies for passive ventilation were deepened, evaluating different control logics of the external air flows. For the classroom cited before, several control strategies for ventilative cooling have been analyzed. The strategies and design solutions have been simulated in TRNSYS, Transient SYstem Simulation tool, that performs transient systems analysis using a modular approach able to model most of the envelope and HVAC subsystems (TRNSYS 16, 2004).

4.1 Night Cooling Ventilation System

The first design solution has regarded the adoption of a night cooling ventilation system. For this design strategy there are no limits of flow rate, speed and temperature of the inlet air, due to the absence of users.

A direct hybrid ventilation has been hypothesized, with the inlet openings on the south walls, arranged close to the ceiling with entry assisted by mechanical ventilation (helical ventilator) and extraction for stack effect through the window at the top of the north face.

It was therefore a horizontal ventilation crossing the building, with inlet and air outlet openings placed on opposite walls with incidence of the wind on the plane of the openings and pressure difference between the two openings.
Several simulations are conducted varying the air change flow rate between 2 ACH and 15 ACH in order to obtain the optimal value.
The mechanical ventilation for night cooling was controlled by an internal and external thermostat.
A first activation logic of the ventilator and the window opening will occur when (Case 1):

- time period: 10 p.m. – 6 a.m.
- external temperatures < internal temperatures.

After, in order to reduce the discomfort for undercooling, the activation logic has been implemented as (Case 2):

- time period: 10 p.m. – 6 a.m.
- lower limit temperatures (evaluated according UNI EN 15251) < external temperatures < internal temperatures.

The discussion of the results of these assumptions are reported in the paragraph 4.3, noting that the night-cooling improves but does not solve overheating discomfort.

4.2 Evaporative Cooling System

As underlined in the following paragraph, it resulted the need to integrate the night cooling with a passive cooling during daytime. For the hot-dry climate, typical of the Mediterranean continental climate, during daytime the design solution has concerned on evaporative cooling system. Indeed this cooling is more sensitive if the air is able to make evaporate much water, namely the more the air is dry initially.
During evaporative cooling, the air undergoes a process of humidification and cooling. In energy balance, the reduction of the air temperature does not involve a variation of the enthalpy content of the air. It is an adiabatic cooling where the reduction of air temperature is compensated with the increase of water steam.
Evaporative cooling can be considered, therefore, a process of passive cooling, where the effect of the evaporation of water present in the air is used as natural heat sink (Alaidroos, 2015).
In this paper, a direct evaporative cooling system has been simulated, where the air comes in contact with water and increases RH.
Effectiveness and benefits of such cooling system are function of two parameters:

- efficiency-of evaporation (percentage of saturation);
- airflow rate.

Regarding the efficiency, several simulations have been conducted considering as reference values 30%, 50% and 70%.
For the classroom above analyzed, the air flow has been varied between a minimum value of 2.15 ACH (according UNI 10339 that defines 0.004 m3/s*person) and a maximum value of 13 ACH (by evaluating a maximum velocity of 0.11 m/s according ISO 7730:2005 for Kindergarten). In particular, five air flow rates have been considered: 2,16 ACH, 5 ACH, 7,5 ACH, 10 ACH and 13 ACH.
Different combinations of air flow and efficiency were evaluated during the occupation hours 7:00 am-7:00 p.m, to assess the benefits obtained from the evaporative cooling system by varying the two fundamental parameters and to identify an optimized solution.
In a first set of simulations (obtained combined the three air flow rates with the five efficiency values), the cooling system operated if the inlet temperature was lower than the indoor temperature.

Subsequently in other simulations (obtained combined the air flow rates upper than 5 ACH with the efficiency values greater than 50%), in order to reduce the discomfort for undercooling, the system turned on if:

- indoor operative temperature > optimal temperature;
- inlet temperature < indoor temperature.

4.3 Results

As regarding the **night cooling ventilation system**, the two cases proposed have been compared in terms of adaptive thermal comfort (UNI EN 15251).
The simulations results have shown the efficacy of the proposed ventilative cooling strategies, determining a significant reduction of overheating during the occupation hours. Comparing the Case 0 and Case 1, there was a decrease of overheating discomfort hours of about 30 %, but an increase of undercooling discomfort hours. Therefore, the activation logic of the Case 2 had needed to reduce the undercooling hours without excessively limiting the cooling efficiency of the system.

Table 3: Thermal Comfort Analysis

Case	Overheating discomfort (hr)	Overheating discomfort (degree hours)	Undercooling discomfort (hr)	Undercooling discomfort (d.h.)
Case 0	385	1040	19	21
Case 1	260	483	74	140
Case 2	347	666	28	24

In order to reduce even more the thermal discomfort conditions, by operating also during daytime, the first set of **evaporative cooling system** proposed have underlined as (fig.4) :

- by increasing the air flow rate with equal efficiency, the benefits were lower than those obtainable by increasing efficiency at equal flow;
- for a very low efficiency, inferior to 50%, even at high flow rates the overheating discomforts were considerable, ie the increase of flow was not sufficient to compensate the low efficiency of the system;
- the optimal air flow rate was between 5 ACH and 7 ACH: the slope was maxim before 5 ACH and almost constant after 7 ACH.

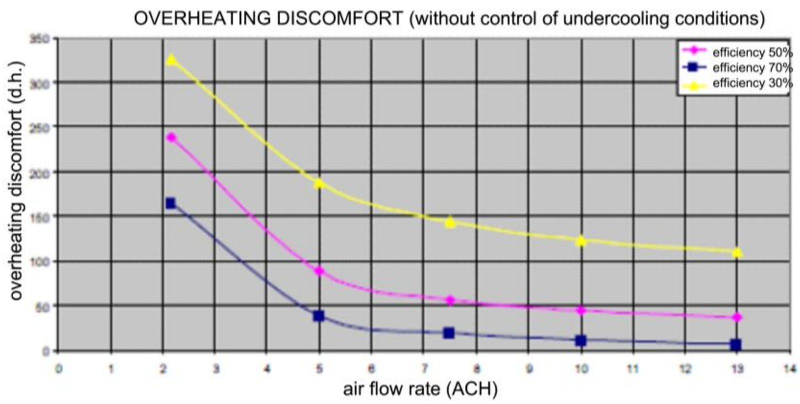


Figure 4: Thermal Comfort Analysis in regime of constant activation

The other set of **evaporative cooling system** proposed have shown as (fig.5) :

- with a greater system efficiency (70%) the operating temperatures had a daily fluctuation mostly contained within the band of comfort;

- high air flow rates, even at high efficiency, were not sufficient to total elimination of overheating discomfort and caused more discomfort conditions for undercooling;
- the optimal solution was represented by 7.5 ACH and efficiency equal to 70%, where occurred only 60 degree hours of discomfort for overheating and undercooling.

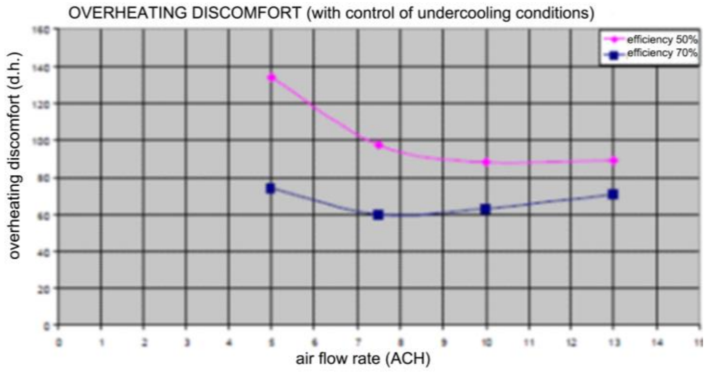


Figure 5: Thermal Comfort Analysis with control logic

The combination of natural cooling ventilation and evaporative cooling system, with control logics above examined, caused a significant reductions of discomfort conditions (> 80%) respect to base case.

Table 4: Thermal Comfort Analysis

Case	Overheating discomfort (hr)	Degree hours for overheating discomfort (d.h.)	Undercooling discomfort (hr)	Degree hours for undercooling discomfort (d.h.)	Total discomfort hours (hr)
Case 0	385	1040	19	21	404
Optimal Case	28	22	48	38	76

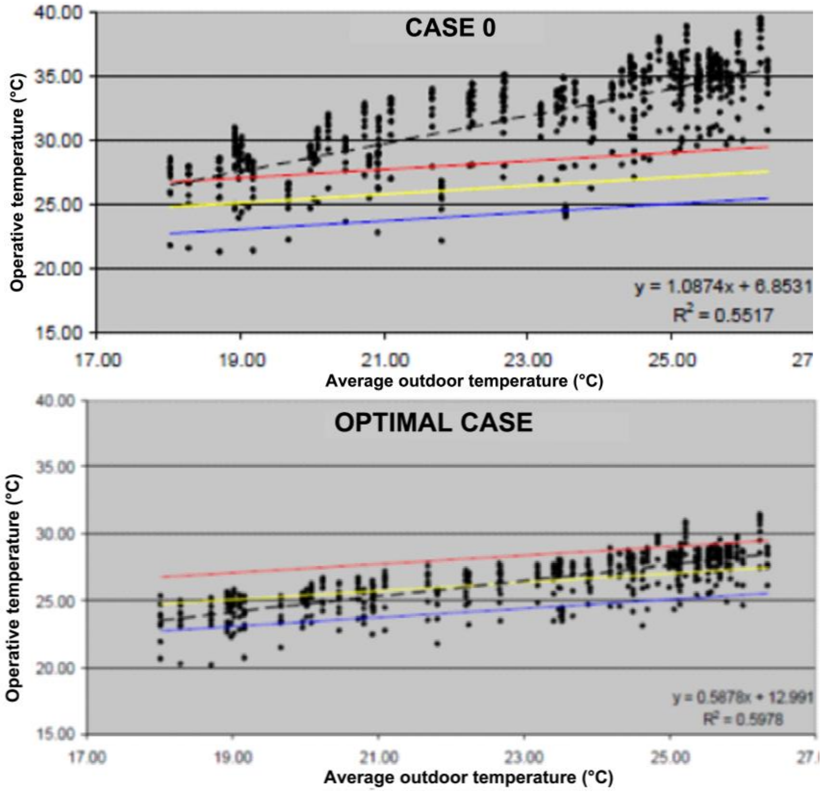


Figure 6: Thermal Comfort Analysis (CASE 0 – OPTIMAL CASE)

Finally, the comparison between the conditions of internal relative humidity in the absence and in the presence of evaporative cooling system was carried out. It is possible to note that the evaporative cooling system:

- did not alter the levels of comfort to a relative humidity internal;
- determined a positive increase in the minimum levels of relative humidity (on hot days and muggy), reducing of the hours when the level of internal humidity was lower than 30%;
- has caused a rise of maximum levels of relative humidity above 70%, however, limited to 5% of the total occupation hours.

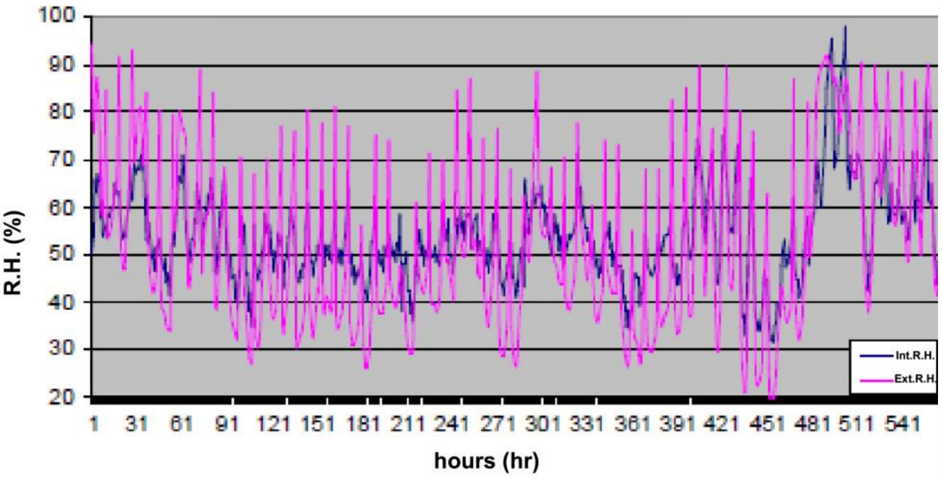


Figure 7: Relative Humidity

Table 5: Analysis of relative humidity levels

Case	40%<R.H.<60%	30%<R.H.<70%	R.H.<30% - R.H.>70%	R.H.<30%	R.H.>70%
Case 0	414 hr	544 hr	23 hr	8 hr	15 hr
	73%	96%	4%	1%	3%
Case optimal	409 hr	536 hr	31 hr	0 hr	31 hr
	72 %	95%	5%	0%	5 %

5 CONCLUSIONS

In this paper evaporative cooling system and ventilation control strategies has been studied to ensure an adequate combination between energy efficiency and high levels of indoor comfort for an existing kindergarten in Mediterranean climate.

The simulations have shown the effectiveness of the night cooling ventilation, particularly in climates with high daily thermal oscillation. Through the control logics described above, it was possible to calibrate the ventilation in relation to the cooling needs, greatly reducing the discomfort for undercooling. Despite the benefits of the night cooling ventilation, a cooling strategy daytime was necessary.

The different strategies for evaporative cooling, obtained by varying the characteristic parameters such as efficiency and flow rate, have shown how it is possible to obtain a high comfort level by adopting reduced flow, if the system has high efficiency.

The analysis showed that the evaporative cooling does not significantly alter the comfort levels for indoor relative humidity. The indoor relative humidity is maintained in the optimal

intervals generally 40-60%, and moreover there is a positive increase in the minimum levels of relative humidity.
This study have underlined that passive cooling systems, operated by a suitable control logic, can provide performance levels almost comparable with those of an air conditioning system, ensuring a significant reduction of energy consumption.

6 REFERENCES

M. Santamouris, *Advances In Passive Cooling*. 2007.

B. Givoni, "Indoor temperature reduction by passive cooling systems," Sol. Energy, vol. 85, no. 8, pp. 1692–1726, Aug. 2011.

Willmert, T. (2001). *The return of natural ventilation.* Architectural Record, 189 (7), 137-146.

M. Bajwa, E. Aksugur, and G. Al-Otaibi, (1993) "The Potential of the Evaporative Cooling Techniques in the Gulf Region of the Kingdom of Saudi Arabia," Renew. Energy, vol.3, no. 1, pp. 15–29.

A. Alaidroos, M. Krarti, (2015), *"Impact of passive cooling strategies on energy consumption reduction of residential buildings in the kingdom of Saudi Arabia"*, ASME 2015 9th International Conference on Energy Sustainability, San Diego, California.

Boyd E., (1935), "The growth of the surface area of the human body", Minneapolis: University of Minnesota Press.

UNI 10339:1995, *Air-conditioning systems for thermal comfort in buildings – General, classification and requirements – Offer,order and supply specifications.*

ISO 7730:2005, *Ergonomics of the thermal environment -- Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria*

TRNSYS 16 (2004) Volume 05 – Mathematical Reference.

UNI EN 15251 (2007). *Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics.*

A MODULAR, OPEN SYSTEM FOR TESTING VENTILATION AND COOLING STRATEGIES IN EXTREMELY LOW ENERGY LECTURE ROOMS

Ando Ludovic Andriamamonjy*, Ralf Klein

Sustainable Buildings Research Group, Technology Campus Ghent, University Leuven
Gebroeders De Smetstraat 1
9000 Gent, Belgium
**Corresponding author: razakaniainaandoludovic.andriamamonjy@kuleuven.be*

ABSTRACT

Lecture rooms with their high, quickly fluctuating internal gains, e.g. changing from no occupation to full occupation within some minutes, are quite challenging when good indoor air-quality and thermal comfort is required in an extremely low energy building context.
One essential aspect is the perfect control of air flow and temperature based on reliable, continuous measurement in all relevant parts of the ventilation system.
This paper describes a case study that combines real building operation on a university campus with an advanced, modular test platform covering all aspects of the real operational performance of extremely low energy lecture rooms.
A full-scale Passive House test facility was constructed at Technology Campus Gent, KU Leuven, Belgium. The building is part of the campus and has two lecture rooms for 80 students each. Designed and certified according to the Passive House standard, the new facility consists of two levels, constructed on top of an existing building (ground-floor only).
Thermal insulation was placed also in-between the two lecture rooms and in the internal walls towards the staircase. This results in a layout with two identical, box-shaped volumes with different thermal mass (one with a timber-frame structure, one with brick-walls).
The facility includes a high performance AHU, with frequency controlled fans and two sets of VAV-boxes, providing ventilation, heating and cooling, a wood pellet boiler, motor-controlled exterior sun-shading and windows and high performance lighting fixtures with daylight control.
A key feature is the integrated system for monitoring and control based on open standards: BACnet for communication with the AHU, KNX, DALI and EtherCAT to link decentralized IO-units with the Building controller. This PLC based, distributed PC environment provides detailed control of the building equipment and real-time, long-term monitoring of all building parameters and the outdoor climate.
It provides also a very flexible and powerful platform for the implementation and testing of new strategies for model based predictive control (MPC) and fault detection and diagnosis (FDD). The Modelica language is used for building simulation during operation.
A detailed Building Information Model (BIM) was created and all relevant elements of the equipment and the BMS will be added. The BIM will be used to manage measured data and provide integration between simulation and measurement.
Results from detailed air flow measurements at different fan speeds are provided. These initial measurements show good general agreement and provide deeper insight in the dynamic behaviour of the ventilation system. Beside the air flow sensors of the AHU and the VAV boxes, Venturi tubes are integrated in the supply- and return-air duct of each lecture room. The modular monitoring system provides the possibility of easy integration

of additional sensors (e.g. thermo-anemometers for temporary measurement of velocities and calculation of the air flow based on the Log-Chebychev method)..

KEYWORDS

nZEB, case study, building monitoring system, air flow measurement

1 INTRODUCTION

Indoor air quality (IAQ) and thermal comfort are essential parameters to ensure an optimal indoor environment for people living or working in buildings. This emphasizes the importance of an adequate ventilation strategy where the main purpose is to provide fresh and clean air while removing harmful air contaminant (Dimitroulopoulou, 2012). This aspect of ventilation is of importance as a correlation between occupant's health and IAQ has been established. The association of ventilation rate and occupants health in commercial buildings was investigated (Seppänen, Fisk, & Mendell, 1999; Seppänen & Fisk, 2004). These studies concluded that at low ventilation rate, building occupants show the symptoms of the sick building syndrome (SBS) which is due to the high concentration of carbon dioxide within the environment. Moreover, (Emenius et al., 2004) examined the impact of building characteristics and IAQ on infants in Stockholm, during the first 2 years of their life. They concluded that indoor humidity superior to 45% and the age of the building are related to a recurrent wheezing in children. Dimitroulopoulou (Dimitroulopoulou, 2012) summarizes in his review study, the likely health effects (asthma, allergies, inflammation) on vulnerable persons such as infants and elderly of an inadequate ventilation system in buildings.
Alongside of maintaining an optimal air quality, the ventilation system is also used for cooling purposes to achieve an optimal indoor thermal comfort (Emmerich, Dols, & Axley, 2001). It can be used to cool directly building interiors by replacing the warm indoor air with cooler outdoor air or to cool directly building occupants by directing cool outdoor air over building occupants at sufficient velocity. Another ventilation aspect is the night ventilation which constitutes an indirect way of cooling building interiors by pre-cooling thermally massive components of the building with cool nighttime outdoor air.
Two main ventilation technologies are currently used: the passive, natural ventilation and the mechanical ventilation. The choice of technology depends on miscellaneous factors ranging from the local climate to the building characteristics and the envelope's air tightness.
Natural ventilation is defined as ventilation provided by thermal, wind or diffusion effects through doors, windows, or other intentional openings in the building. It has the potential to significantly reduce the energy cost required for mechanical ventilation and constitutes the primary choice for building ventilation as long as the building characteristics and local conditions permit its safe and (energy-) efficient implementation.
Three (3) fundamental approaches of natural ventilation are to be considered (Emmerich et al., 2001):

- The wind-driven cross ventilation which occurs while openings on opposite sides are kept open
- The buoyancy driven stack ventilation which relies on the temperature differences of indoor vertical air layer stratification to draw cool outside air.
- The single sided ventilation where the openings from one side of the room are used.

Most of the currently used natural ventilation strategies are variants of the above-mentioned approaches and are enhanced by using controlled openings.
On the other hand, mechanical ventilation implies the use of mechanical fans connected to ductworks to supply fresh outdoor air to the building interiors. An obvious and major drawback of these kind of systems is the high use of energy to run the mechanical components. According to (Pérez-Lombard, Ortiz, & Pout, 2008), about 50% of the total

primary energy used within buildings are used for mechanical ventilation systems. This situation points out the necessity of energy efficient ventilation strategies which leads to an intensive research for optimised design and control of mechanical ventilation systems and their integration into the building context. Three basic types of mechanical ventilation strategy are to be considered (Roberson et al., 1995):

- The exhaust only ventilation where fans are used to extract indoor air from the building while outdoor air is entering through openings and buildings leakage.
- The supply only ventilation where fans are used to push air into the building and induce a positive pressure. This constitutes the main advantage of this strategy as it prevents outdoor pollutants to flow into the building.
- The balanced ventilation strategy where the supply and exhaust fans provide similar volume flows. This type of ventilation system is typically equipped with heat recovery to improve indoor thermal comfort and energy efficiency.

Considering these fundamental approaches for natural and mechanical ventilation strategies, a large range of combination and hybrid ventilation strategies can be implemented. However, prior to any implementation in real buildings of any strategy, a thorough study needs to be undertaken to find out its suitability considering local climate and building characteristics and its efficiency in term of health, comfort and energy use. Moreover, alongside an in depth theoretical study, an experimental validation and testing is a mandatory step to investigate the effectiveness of a strategy in the real world. This situation emphasizes the need for an experimental facility having the capability to provide an adequate environment such as an accurate and reliable air flow measurement system to carry out such experimentation.

This work presents a high performance test facility having the capability to host such testing and experimentation due to its unique features. The building is a passive house located in a temperate climate and fully equipped to perform natural ventilation, as well as mechanical ventilation strategy testing. Four main features are:

- Its highly insulated and air-tight building envelop and motorized windows and exterior shading provide a unique environment for natural ventilation scenario testing.
- Its integrated computer based monitoring and control system provide a modular, real time testing framework.
- Its high performance airflow measurement system provides continuous and reliable airflow data of the mechanical ventilation system's key components as well as the indoor and outdoor air velocity and direction.
- Its comprehensive set of installed sensors which covers all key aspects of buildings.

Within a first section, the main characteristics of the facility will be presented. The focus will be set on the facility's unique features related to ventilation testing. A particularly relevant point is the choice of a continuous air flow measurement system. As velocity based air flow measurement technologies such as the log Chebyshev method are strongly influenced by local perturbancies (fans, bends, …) and less convenient to set up (De Zutter & De Bock, 2014), the facility is equipped with Venturi-tubes (ISO 5167-4) with differential pressure sensors to measure air flow within the ducts of the ventilation system. Their measurement performance will be emphasized by showing their behavior towards an identical and controlled excitation and results are checked against the values provided by the sensors of the Air Handling Unit (AHU) and the Variable Air Volume (VAV) boxes.

A second section will be dedicated to the prospective ventilation strategy experimentation possibilities that will be implemented by using the presented facility as test case. A non-exhaustive review of existing studies and research will be used to demonstrate the facility's capabilities towards ventilation strategy implementation.

2 TEST FACILITY ASSETS FOR VENTILATION STRATEGIES DEVELOPPMENT AND OPTIMISATION

The facility is part of the civil engineering department of the Technology campus Gent of Leuven University (Belgium) and consists of two lecture rooms for 80 students each (lecture room E220 and E120) (see figure 1). The building was designed and certified according to the passive house standard with a total energy demand of 11 kWh/m^2y and 6.10 kWh/m^2y respectively for space heating and cooling. Moreover, the air tightness of the lecture rooms reaches a value of $n_{50} \leq 0.6\ h^{-1}$. Alongside of being used as normal lecture room for courses, the facility is also used by students, researchers and projects for research on building energy-efficiency strategies development in a "real use" environment. Its main components which are the building envelop, the Air Handling Unit and the integrated monitoring and control system have specific features which facilitate the testing of such strategies and especially ventilation strategies.

2.1 Facility envelope features for ventilation strategy implementation

The test facility was constructed on top of an existing university building (see Figure 1.a) and has four zones which are: the two lecture rooms (E120 and E220), the staircase and the technical room. The lecture rooms form two cuboids with a volume of 380 m^3 each and are thermally insulated from the outside and the neighbouring zones. The E120 room's external wall was designed as brick wall with exterior insulation (cellulose fibre) with an average u-value of 0.148 w/m^2K while the lecture room E220 was designed with an 0.346 m thick insulated lightweight timber frame wall with an u-value of 0.155 w/m^2K. This design of the lecture rooms leads to two identical box shaped zones with different thermal mass. Such characteristic could be emphasized in a ventilation strategy where the thermal mass plays an important role (e.g: Night ventilation). Moreover, for ventilation strategy study involving numerical modelling, the simple architectural shape of the building and the thermal separation of the lecture rooms facilitates the model implementation and calibration.

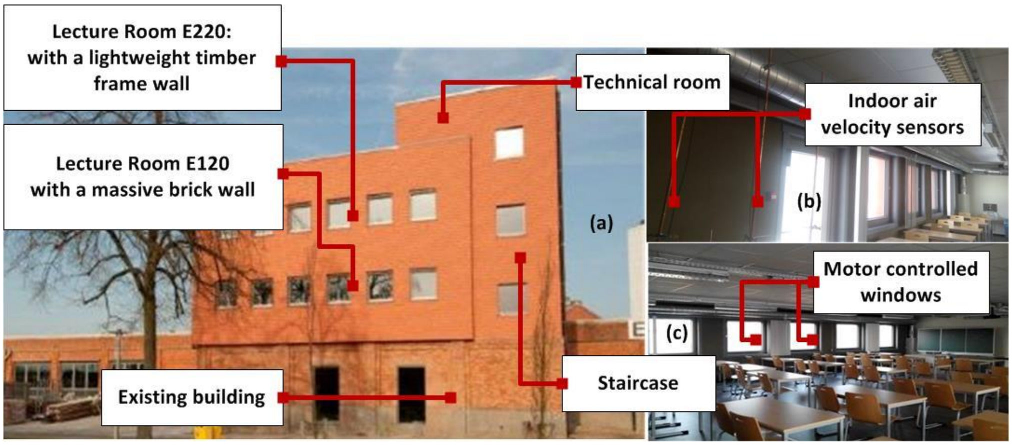


Figure 1: Building envelop presentation and brief description

Each classroom has the same surface and type of windows which are high performance triple glazing windows filled with argon with a solar heat gain value g-value of 0.52 and an u-value of 0.6 w/m^2K (see Figure 1.c). Each window's opening rate is driven by motors controlled by the computer based control system and can be opened or closed automatically and independently. Electricity counters monitor also the energy consumption of these motors. Such feature constitutes an important asset for natural ventilation scenario study and implementation.

A set of sensors has been installed to monitor indoor and outdoor conditions. The facility has its own weather station which monitors the main outdoor parameters such as the solar radiation, the outdoor temperature, the outdoor humidity and the wind speed and direction. For the indoor conditions, the indoor temperature, the CO_2 concentration and the indoor humidity are continuously monitored. However, a set of indoor high precision air velocity sensors is available to monitor the distribution of indoor air velocity and direction (figure 1.b). The data from these sensors are displayed in real time and recorded within the computer based monitoring and control system. Among the above mentioned sensors, those of interest for ventilation strategy implementation are the outdoor wind speed and direction sensors and the indoor air velocity sensors. These sensors are to be used to investigate the outdoor and indoor air speed and motion. Moreover, as the knowledge of the occupancy's level is an important parameter for mechanical ventilation control, the facility, alongside of the CO_2 sensors, has motion detection sensors and high definition camera with face recognition installed within each classroom. They provide the occupancy level in real time.

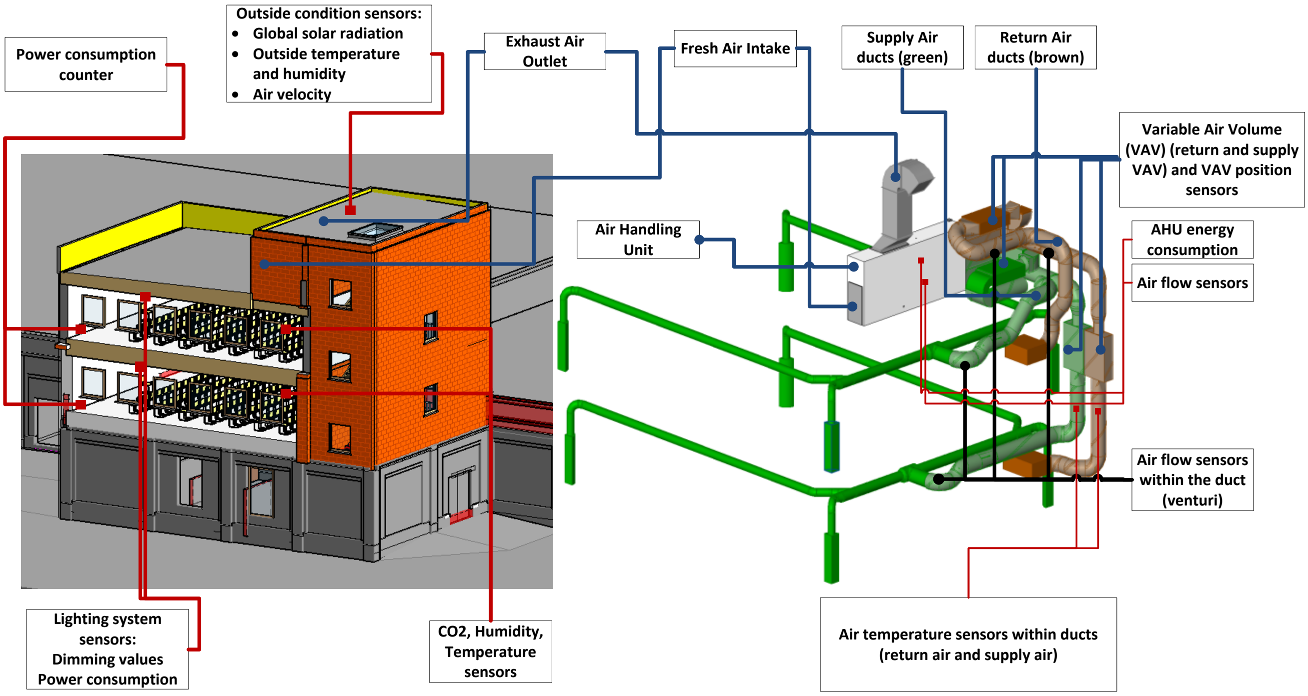


Figure 2: Ventilation system description and sensors placement description

2.2 Air Handling Unit (AHU) features for ventilation strategy implementation

The AHU is a high-performance unit capable of a maximum air flow rate of 5100 m^3/h and has an air flow of 4000 m^3/h at normal condition (see figure 2). It has a coefficient of power efficiency of 72 % (EN 13053:2012) (European Standard, 2012) and is equipped with a double heat exchanger in polypropylene and an Indirect Evaporative Cooling (IEC) system. The heat recovery system can reach an efficiency superior to 75% and is categorized as H1 class (EN 13053:2012). The AHU unit is powered by two energy-efficient fan units with respectively a Specific Fan Power (SFP) category 1 for the air supply fan and category 2 for the Return Air (RA). The fans' speed can be controlled externally and independently to one another by introducing the fans' motor desired angular frequency. Figure 3 (left curve) presents a fan speed control test which emphasizes two distinct periods. A first period where the supply air fan is controlled and a second period where the return air fan is controlled. One can notice the stair shape curve of the controlled air flow which testify the change in angular

frequency operated by the controller during the test. This control feature shows a large range of possibilities for mechanical ventilation strategies testing and implementation as it can mimic an over pressurization and a depressurization phenomenon within the lecture rooms.
To regulate the supply and return airflow rates of the two lecture rooms separately, the ventilation system is equipped with (4) four Variable Air Volume (VAV) where the dampers' position can be controlled by the BMS through the BACnet interface of the AHU. Hence, by closing both return and supply VAV of a lecture room, we can implement natural ventilation testing in one room while using the other room for mechanical or hybrid ventilation.
Knowing the importance of airflow within ventilation strategies, two distinct airflow measurement processes are present within the ventilation system:

- An airflow measurement system embedded within the AHU provides the total supply and return volume flow from the AHU (figure 3, left curve) and the linear position of the four VAV. The combination of these parameters allows us to compute the air flow entering and outgoing the lecture rooms.
- A second airflow measurement system is composed by four differential pressure sensors installed into Venturi tubes which are integrated into the supply- and return-air duct of each lecture room (see figure 2 black lines). These additional sensors are installed to continuously monitor the air flow rates of both lecture rooms. The figure 3 (red and blue curves) shows the supply air entering the lecture rooms (E120 and E220) recorded with the "Venturi sensors" while the black curves present a comparison of the supply flow rate values monitored with the two distinct systems. One can notice the good agreement between the two airflow measurement systems.

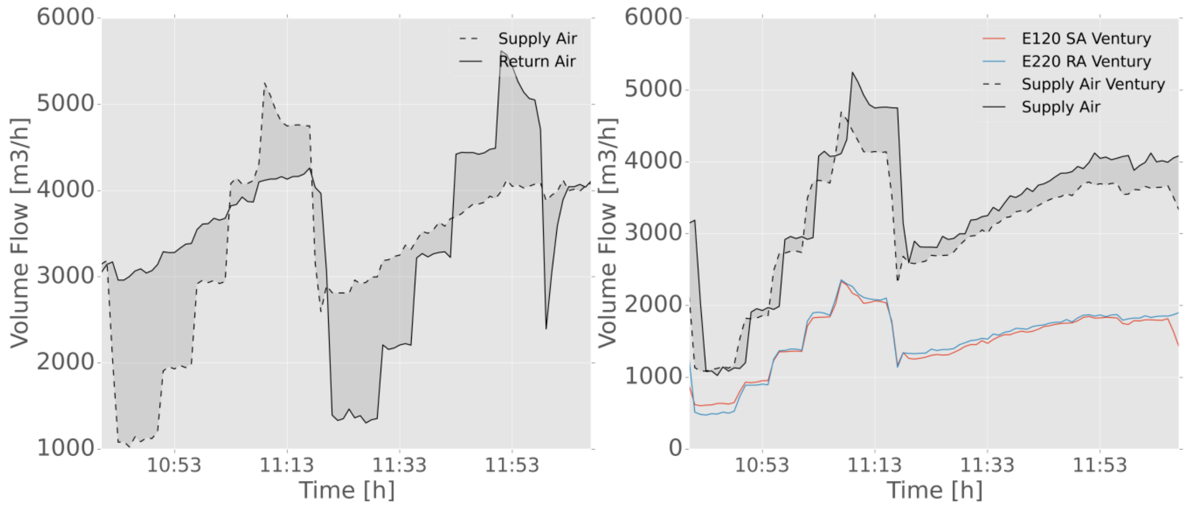


Figure 3: Airflow measurement system comparison

Separate electricity counters are also installed to monitor the energy consumption of the Air Handling Unit.

2.3 Monitoring system features

The monitoring and control system is one of the main assets of the facility. The system relies on the use of open and standard protocols to establish the communication between the main computer based Programmable Logic Controller (PLC) and the system components such as the sensors, the Air Handling Unit and the controllable devices. The open protocol BACnet is used for the fan speed and VAV position control and to communicate with the AHU's embedded sensors while the KNX standard and the EtherCAT protocol are respectively used for controllable devices and decentralised input/output (sensors) in the facility. The monitoring system and control structure can be represented as a two layers structure (see figure 4).

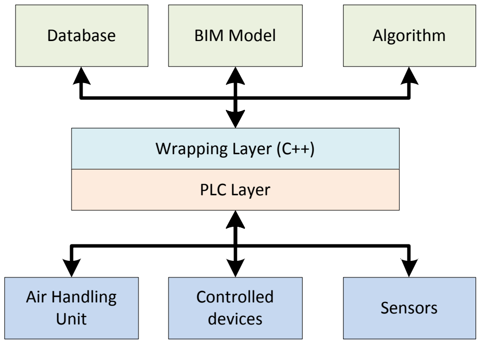


Figure 4: Simplified scheme of the monitoring system

The main task of the bottom real-time PLC layer is to establish and coordinate the communication between the hardware components and the upper software layer while the wrapping layer which is mainly coded in C++ has three (3) main tasks: (1) It establishes the communication between the bottom layer and the database where the monitored data is recorded with a one minute time resolution. (2) It constitutes the link between the system and the Building Information Model (BIM) of the facility. To improve measured data management, the monitoring system was coupled with the existing BIM model of the facility (figure 2 shows a screenshot of the BIM). Each physical sensor is represented within the BIM by its own virtual representation and BIM characteristics are used to couple the virtual sensor with its corresponding measured data to facilitate data access. This method enhances the measured data access for numerical simulation validation. (3) It constitutes a link between the system and the user's control algorithm. Hence, users and researchers have access to real time measured data and control. The system was designed in a way that users can implement their algorithm or scenario using their "favourite" programming language for control. Such configuration confers to the users a large panel of possibility for ventilation cooling strategy implementation testing.

3 PROSPECTIVE CONTRIBUTION TOWARDS VENTILATION STRATEGY TESTING

Considering its features, the test facility provides an optimal, controlled laboratory environment to perform ventilation strategies. The following section emphasizes the advantages provided by the facility characteristics towards ventilation strategies implementation based on a non-exhaustive literature review.
Within the last decade, the high energy use in mechanical ventilation system has raised concern and leads to a regain of attention towards the natural ventilation systems. Passive ventilation could be used either as the main ventilation system or coupled with mechanical ventilation system to reduce energy use. Research involving experimental investigation as well as numerical study of natural ventilation has been undertaken to enhance its efficiency. Among them, (Heiselberg & Perino, 2010) evaluate the airing performance of buoyancy driven and single-sided natural ventilation in terms of ventilation characteristics, IAQ and thermal comfort. They investigate the consequences of opening time, opening frequency, opening area, ventilation efficiency and thermal comfort within a fully monitored test facility equipped with controlled windows located in Aalborg (Denmark). The importance of these parameters has been confirmed by (Karava, Stathopoulos, & Athienitis, 2007). In their work they also point out the necessity of further experimental work before generalization of the conclusions. In this scope, the presented facility has the required technical features to host

such experiments. As the facility is equipped with motor driven windows located on the two opposite façades of each lecture room, one can investigate various combination of windows opening rate of the windows while their effect is recorded and monitored by the indoor sensors installed. Hence the facility provides an ideal environment to perform such test.

Alongside experimental studies, numerical modelling and especially computational fluid dynamics modelling is widely used for natural ventilation performance study (Hu, Ohba, & Yoshie, 2008; Jiang & Chen, 2001; Ramponi & Blocken, 2012). This implies the need of high precision monitoring of the indoor air velocity and direction for model validation. For this purpose, as the sensors are already available, air velocity and air temperature at various heights within the rooms can be measured as needed. The modularity and flexibility of the computer based system allows to easily install, remove or change sensors for a specific test.

Being a major energy use item within buildings, the enhancement of mechanical ventilation systems efficiency has drawn researcher's attention. Due to the control of the fan speed and VAV position of the Air Handling Unit of the presented facility, one can test and investigate the performance of mechanical ventilation strategies by creating similar environment such as over pressurization for supply only ventilation strategy or under pressurization for exhaust only. The facility features fulfill all technical requirements to investigate the advantageous and disadvantageous aspects of these fundamental approaches of mechanical ventilation. However, although testing and implementing new mechanical ventilation strategies is possible within the facility, the current focus within this field is set on the enhancement of the control strategy. (Erickson & Cerpa, 2010; Karava et al., 2007) propose a demand response HVAC control strategy that uses real time occupancy monitoring to achieve efficient conditioning. They conclude that such control can reach a 14% to 20% energy usage reduction of the Air Handling Unit system. An important challenge in the implementation of such method is the real time occupancy detection method. Having CO2 sensors, motion detection sensors, camera with face recognition installed within each lecture rooms and two air flow measurement system, the test facility has the capability to provide the real time occupancy level. Therefore, it can be used to implement the existing methods or improve their efficiency by investigating the possibility of an enhanced control of the AHU coupled with (controlled) natural ventilation.

Another ventilation aspect with promising implementation within buildings is the night ventilation process. This process relies on the use of the cool ambient air as a heat sink, to decrease the indoor air temperature and the temperature of the building's structure. The cooling efficiency of this technique is mainly based on the air flow rate as well as on the thermal capacity of the building and the efficient coupling of air flow and thermal mass (Geros, Santamouris, Tsangrasoulis, & Guarracino, 1999). The importance of air flow within this method and thus air flow measurement constitutes one of reasons of the special attention given to the air flow measurement system within the facility. Night ventilation could be coupled with natural ventilation, mechanical ventilation or hybrid strategies. The importance of thermal mass for night ventilation is emphasized by (Givoni, 1998) within his study where he investigates the effect of thermal mass on night ventilation by considering three houses with different thermal mass located in Pala, South California (USA). He concludes that night ventilation is efficient for buildings with high thermal mass during the summer. As the two classrooms of the facility has different thermal mass, such experiments initiated by (Givoni, 1998) could be undertaken to investigate the effect of the thermal mass difference between the two classroom in a temperate climate especially during summer periods with high occupation and high outdoor temperature during day time.

In addition to the above-mentioned non exhaustive list of possibilities, one of the most important characteristic of the system is its open, integrated and modular monitoring and control system, implemented as soft-PLC in a standard computer environment (IPC) . This feature allows users to interact directly with the control features and real time monitored data and avoid the time consuming process of learning new software and languages to communicate with the control system and software limitations of a traditional hardware PLC. Being a recently built facility, a set of measurements and studies need to be initiate on the facility either within the ventilation strategies implementation field or building physics field. However, some projects such as the “Optimization of demand control ventilation in tertiary buildings nZEB project” already use the facility as test case (VraagVent, n.d.).

4 CONCLUSION

The constant evolution of research on buildings performance efficiency raises the need for full-scale test facilities case for experimentation, method testing and validation. This work presents a test facility with dedicated features for natural ventilation as well as for mechanical ventilation strategy implementation and control. Its main characteristics lie in its envelope features, AHU controllability and air flow measurement system, and its modular, flexible and integrated monitoring and control system. Special attention was paid to airflow measurement where a first experiment considering various imposed steps of constant airflow testified the accuracy of the measurement process. A non-exhaustive review of the testing possibility that can be carried out within the facility has been presented based on existing studies and research on ventilation strategies. Regarding its characteristics, this demonstrated the large range of experiments that can be hosted by the facility and open new perspectives and possibilities for ventilation strategy implementation using the facility as a test case.

5 ACKNOWLEDGEMENTS

This work was carried out under a grant from the University of Leuven. The authors acknowledge this support.

6 REFERENCES

De zutter, t., & de bock, h.-j. (2014). Verdere uitbouw testgebouw duurzaam bouwen monitoring ventilatiesysteem : uitbouw testopstelling.

Dimitroulopoulou, c. (2012). Ventilation in european dwellings: a review. Building and environment, 47(1), 109–125. Http://doi.org/10.1016/j.buildenv.2011.07.016

Emenius, g., svartengren, m., korsgaard, j., nordvall, l., pershagen, g., & wickman, m. (2004). Indoor exposures and recurrent wheezing in infants: a study in the bamse cohort. Acta paediatrica, 93(7), 899–905. Http://doi.org/10.1080/08035250410025582

Emmerich, s., dols, w., & axley, j. (2001). Natural ventilation review and plan for design and analysis tools. National institute of standards and technology, nist. Retrieved from http://www.bfrl.nist.gov/iaqanalysis/docs/nistir_6781_natventtool.pdf

Erickson, v. L., & cerpa, a. E. (2010). Occupancy based demand response hvac control strategy. Buildsys, 7–12. Http://doi.org/10.1145/1878431.1878434

European standard. (2012). Din en 13053:2012-02. Retrieved from http://www.beuth.de/en/standard/din-en-13053/143293044

Geros, v., santamouris, m., tsangrasoulis, a, & guarracino, g. (1999). Experimental evaluation of night ventilation phenomena. Energy and buildings, 29(2), 141–154.

Givoni, b. (1998). Effectiveness of mass and night ventilation in lowering the indoor daytime temperatures. Part i: 1993 experimental periods. Energy and buildings, 28(1), 25–32. Http://doi.org/10.1016/s0378-7788(97)00056-x

Heiselberg, p., & perino, m. (2010). Short-term airing by natural ventilation - implication on iaq and thermal comfort. Indoor air, 20(2), 126–140. Http://doi.org/10.1111/j.1600-0668.2009.00630.x

Hu, c. H., ohba, m., & yoshie, r. (2008). Cfd modelling of unsteady cross ventilation flows using les. Journal of wind engineering and industrial aerodynamics, 96(10-11), 1692–1706. Http://doi.org/10.1016/j.jweia.2008.02.031

Jiang, y., & chen, q. (2001). Study of natural ventilation in buildings by large eddy simulation. Journal of wind engineering and industrial aerodynamics, 89(13), 1155–1178.

Karava, p., stathopoulos, t., & athienitis, a. K. (2007). Wind-induced natural ventilation analysis. Solar energy, 81(1), 20–30. Http://doi.org/10.1016/j.solener.2006.06.013

Pérez-lombard, l., ortiz, j., & pout, c. (2008). A review on buildings energy consumption information. Energy and buildings, 40(3), 394–398. Http://doi.org/10.1016/j.enbuild.2007.03.007

Ramponi, r., & blocken, b. (2012). Cfd simulation of cross-ventilation for a generic isolated building: impact of computational parameters. Building and environment, 53, 34–48. Http://doi.org/10.1016/j.buildenv.2012.01.004

Roberson, j. A, brown, r. E., koomey, j. G., warner, j. L., greenberg, s. E. (1995). Ventilation strategies for energy-efficient production homes. Lbnl, (december), 287–298.

Seppänen, o. A, fisk, w. J., & mendell, m. J. (1999). Association of ventilation rates and co2 concentrations with health and other responses in commercial and institutional buildings. Indoor air, 9(4), 226–252. Http://doi.org/10.1111/j.1600-0668.1999.00003.x

Seppänen, o. A., & fisk, w. J. (2004). Summary of human responses to ventilation. Indoor air, supplement, 14(suppl. 7), 102–118. Http://doi.org/10.1111/j.1600-0668.2004.00279.x

Vraagvent. (n.d.). Vraagvent. Retrieved june 10, 2015, from https://iiw.kuleuven.be/onderzoek/vraagvent

COLD STORAGE IN THE THERMAL MASS OF BUILDINGS USING NIGHT VENTILATION. EXPERIMENTAL ANALYSIS

Rafael Salmerón Lissen[1], Francisco José Sanchez de la Flor[2], Alejandro Rincón Casado [2], Álvaro Ruiz-Pardo[*2] and Servando Álvarez Domínguez [3]

1 SAMLER Arquitectura
ETSI, Camino de los descubrimientos S/N
Seville, Spain

2 University of Cadiz,
Escuela Superior de Ingeniería, Av Argentina, 1
Puerto Real Cádiz, Spain
**Corresponding author: alvaro.ruiz@uca.es*

3 University of Seville,
ETSI, Camino de los descubrimientos S/N
Seville, Spain

ABSTRACT

An experimental analysis of the night ventilation technique for cooling in buildings, was performed in a test cell with the aim of establish the potential of this technique in two scenarios: a) when the air-stream is in poor contact with the thermal mass and b) when the air-stream is in close contact with the thermal mass of the test cell. The test cell is a small one-room building equipped with instrumentation for measurement and control the night ventilation following a strategy based in the values of indoor and outdoor temperatures.

The results showed that similar mean daily temperatures are reached when the air stream is in poor contact with thermal mass and when is in good contact. Nevertheless, lower variation in the interior temperature is obtained when the air stream is in good contact with thermal mass, producing better comfort conditions since no very high and low temperatures are achieved in this case.

KEYWORDS

Night ventilation, Thermal mass, Mediterranean Climates, Free cooling

1 INTRODUCTION

In this paper we present experiments conducted within the framework of the project "Analysis of the energy performance of concrete envelopes based on maximizing the benefits of its thermal inertia" (Análisis del comportamiento energético de los cerramientos de hormigón en base a la maximización de las ventajas derivadas de su inercia térmica). The experimental setup consists of a test cells representing a building of concrete that has been instrumented with sensors to record the evolution of the surfaces and air temperatures in the interior; and weather station for recording external conditions. The test cell is located in Alcala de Guadaira, a little town near to Seville Spain.

Night ventilation potential for improving comfort has mainly been investigated by numerical means(Pfafferott et al. 2003). (Artmann et al. 2007) evaluated the potential of passive cooling by night-time ventilation in Europe; they conclude that in the Northern Europe the potential of passive cooling is very significant and it decreases with the latitude being a limited potential in the southern Europe. Nevertheless, several studies shows that this technique can be useful in Mediterranean climates (Irulegi et al. 2014), (Macias et al. 2006), and some of them indicate that the use of the thermal inertia is a key variable in order to obtain a significant reductión of temperatura and cooling loads (Corgnati & Kindinis 2007), (Kolokotroni & Aronis 1999), (Geros et al. 1999).

Experiments performed in the test cell are focused to evaluate the importance of the thermal mass in the effeteness of the night ventilation technique. .

2 EXPERIMENTAL SETUP

The experimental setup consists of a stand or test cell, made of concrete, located in the premises of a cement factory near the city of Seville, Spain, (37º21'35 "N, 5º51'50" O) about 40m altitude. This town is classified as extreme summer and mild winter in Spain.
The interior dimensions of the cell are 2.9m wide, 2.40m deep and 2.40m high. The walls of the north facade, east and west have two layers: the outer concrete is 2500 kg / m3 density and 12cm thick; the inner polystyrene layer is 8 cm thick. The door is located on the north side and is internally insulated by a layer of polystyrene of the same thickness as the walls. The cover is made of concrete with a layer of expanded polystyrene insulation inside 5cm thick.

Ventilation is carried out by mechanical fans drive where 8 147 m3 / h peak flow each, are located in the door of the test cell by way of extraction as shown in figure 1. The consumption of these fans is 140W, though It has confirmed that exist in the market more efficient options that can lower the power consumption to about one tenth.

figure 1 Exhaust fans installed at the door of the test cell

figure 2, schematically shows the outside air, passes first air chamber of the south wall upwards until holes on top communicating with the inside of the building, where the air runs inside of it out by fans located in the rear door (north side).

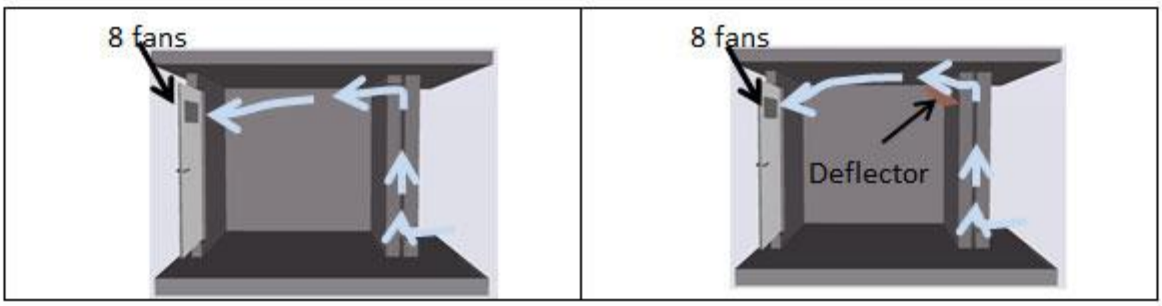


figure 2. Two flow-patterns considered in order to evaluate tye effect of the thermal storage in the roof

In order to evaluate the effect of thermal storage in the concrete slab, two flow patterns were tested: one in which the air enters through the holes directly, and another in which a baffle is positioned to enhance contact between the fresh air and the concrete slab. The baffle was constructed of plywood and installed as shown in figure 3.

The main points of temperature measurement are shown in figure 3. In each of these points a T type thermocouple was connected to a CompactDAQ data acquisition system which provides an accuracy of +/- 0.5 ° C.

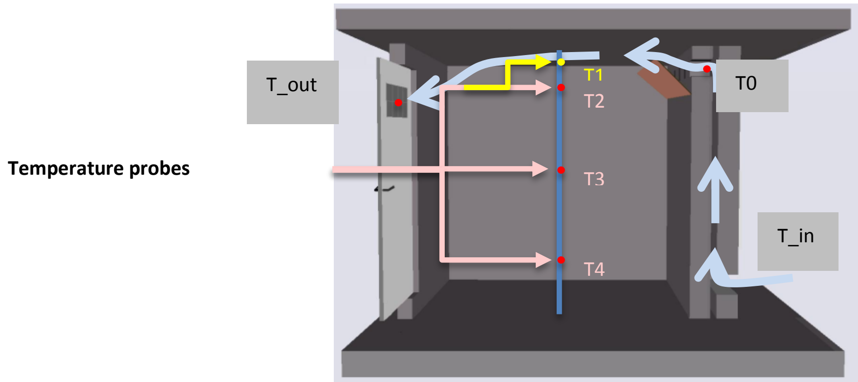


figure 3 Temperature sensors to measure the effect of night ventilation

The starting and stopping of ventilation was controlled by the difference of temperature between the indoor and outdoor temperature, if this was with a value higher than 4ºC, the fans start operation, and stop when the outdoor temperature is inferior to 2ºC.

3 METHODOLOGY

The experiments were performed between 02.09.2014 and 06.10.2014, and consisted of observing the indoor temperature, under different operation modes.

The first experiment, was to let the test cell temperature swings freely without the presence of any forced night ventilation. This took place between 09.02.2014 and 08.09.2014. The purpose of this experiment was to determine the performance of the test cell without night ventilation to evaluate later the effect of ventilation.

The second experiment was to operate the cell at night ventilation scheme without deflector. Under this scheme, air enters and leaves the test cell without having a good contact with the inner surfaces. This experiment took place from 08 to 15 September 2014.

The third experiment is identical to the second except for the placement of a deflector to forces the air stream have a better contact with the ceiling. This experiment was conducted 15.09.2015 and 06.10.2015. The objective of this experiment was to evaluate the effect of cold accumulation in the structure.

In this study, we used a single cell to perform each of these experiments. It has an obvious advantage of cost savings. The disadvantage of using a single test cell is that the excitation conditions are different in each experiment as can be seen in figure 4. The consequence is that the results cannot be compared directly. However, it has carried out an analysis and comparison of the results in relative terms, obtaining consistent conclusions.

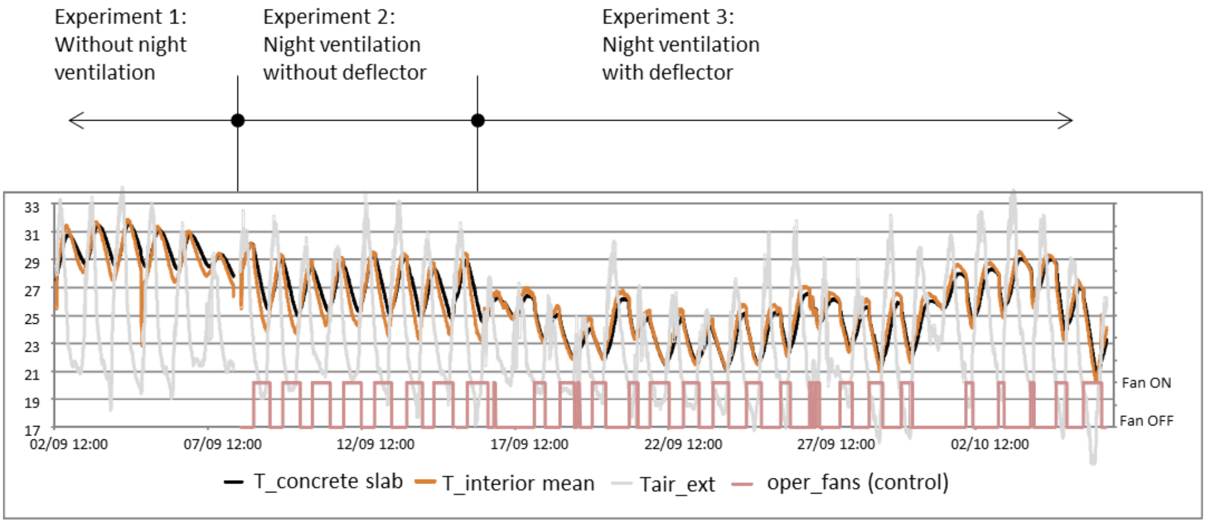


figure 4. Main temperatures of the three experiments performed, and fans operation

The method of analysis of the results was performed in two ways:

Analysis 1: Assessment of the difference between the outside temperature and the temperature inside the test cell

Analysis 2: Evaluation of the daily fluctuation of the temperature inside.

The first analysis gives an indication of the total heat quantity in evacuated form of each operation. Analysis 2 gives an indication of the amount of cold stored in the structure, since a low temperature fluctuation indicates that the cold is stored and transferred slowly during the day.

4 RESULTS AND DISCUSION

For each of the first two experiments, graphics corresponding to three days were selected. The selection of these days was done with the intention that external conditions were similar to each other.

For the third experiment, although the same similarity of external conditions was sought, it was not possible to find a nearby sufficiently clear and for that reason, two periods of three days are shown. In the first period the similarity with experiment 2 is that the operating times of ventilation are similar. With experiment 3, the similarity is that mean temperatures are approximately equal to those of the first two experiments.

4.1 Experiment 1. Without night ventilation

In this experiment the test cell works in free floating mode without any ventilation and air exchange with the exterior except infiltration.

It is worth mentioning that in this period of testing, it was necessary to enter the test cell for reasons of maintenance, resulting in a slight discontinuity in indoor temperatures, as will be seen in the graphs presented in this section. Still, the results are considered to be valid, because that the variations were very small.

The figure 5 shows that the average indoor air temperature and ceiling has an amplitude much smaller than the outside oscillation. This result is expected due to thermal inertia, which is also the one that occurs there is a delay between the highest and lowest points of the temperatures outside and inside.

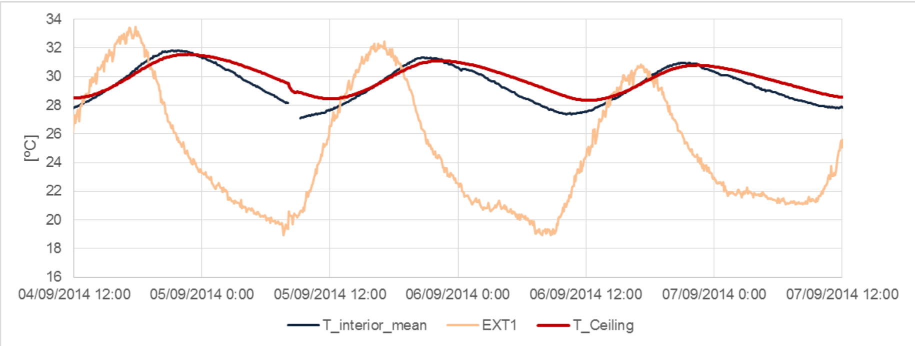


figure 5. Evolution of the interior, exterior and ceiling temperatures in the experiment without night ventilation

The temperature of the ceiling shows that almost always remains above the interior, indicating that the slab provides heat to the test cell, as a result, mainly, of solar radiation incident on its outer surface. It is also evident that the concrete slab has a high thermal inertia due to the thermal oscillation amplitude is less than the interior and their higher delay.

Furthermore, it was recorded in the cell a temperature stratification inside, as shown in figure 6, where it is seen that between the highest and nearest to ground sensor, T4 (0.4m) and T1 (2.32m) the difference is about 2 ° C to remain constant in all the registered period.

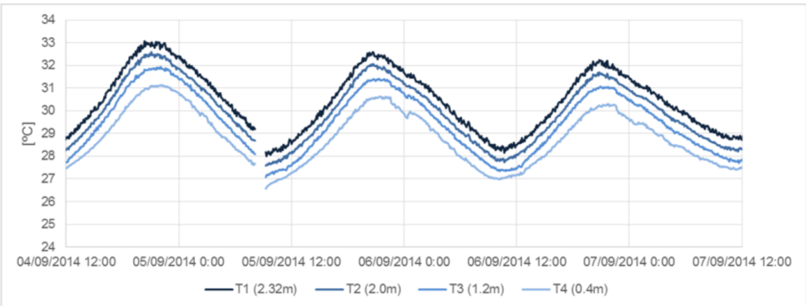


figure 6. Evolution of the interior temperatures at four levels

4.2 Experiment 2. Night ventilation without deflector

This experiment develops a night ventilation controlled by the temperature difference between the outside and inside of the test cell following the criteria mentioned above. The air stream in this experiment is not forced to come in contact with the ceiling.

The behavior of the temperatures shown in figure 7 displays that the average interior temperature descends significantly when fans works, but this also rise sharply when it stops working as result of internal gains produced by equipment inside the tests cell and heat gains through the envelope, where solar radiation plays an important role.

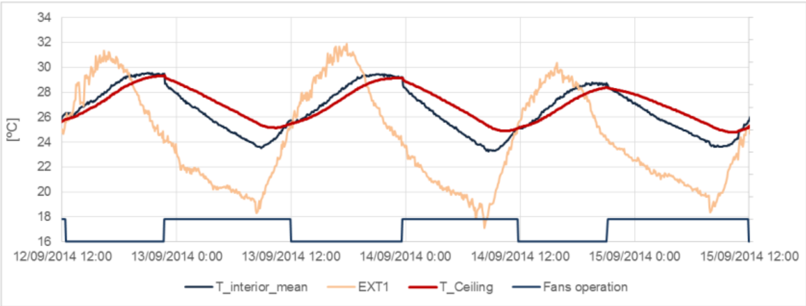


figure 7. Evolution of the interior, exterior and ceiling temperatures in the night ventilation experiment without deflector

The temperature of the ceiling shows a clear signal not to be being modified directly by the ventilation, which can be interpreted as the existence of a poor contact between the air stream and this surface.

From the behavior of air temperatures at different heights shown in figure 8, we can see that in the absence of ventilation, the behavior is similar to that in experiment 1, in other words, if the height is bigger, higher temperature. However, when there is ventilation, the temperature behavior is inverse, that is, the hottest area is the bottom and the cooler is higher; the temperature difference between the upper two points is negligible, it is suggesting that the air stream passes through these two points.

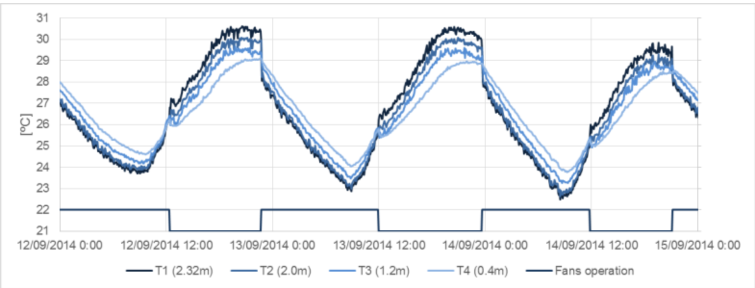


figure 8. Evolution of the interior temperatures at four levels

4.3 Experiment 3. Night ventilation with deflector

The only difference between this experiment and experiment 2 is the installation of a deflector to force the airflow to come into contact with the ceiling.

As stated earlier, it was not possible to find a sequence of three days similar to those of the first two experiments conditions. In this case, two sequences of three days each were selected.

The similarity of the first selected set of three days, is that the periods of night ventilation operation are similar to those of Experiment 2. In this case, the temperature of the ceiling is lower than the mean indoor temperature (figure 9) when the ventilation is off, indicating that cold is accumulated in the slab; and not as in experiment 2 in which the temperature of the ceiling were superior.

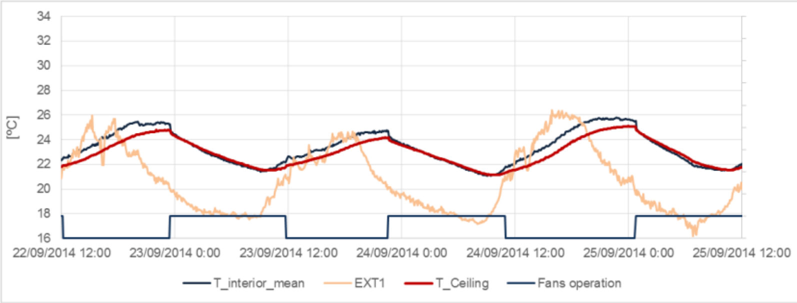


figure 9. Evolution of the interior, exterior and ceiling temperatures in the night ventilation experiment with deflector. First series of days selected in experiment 3

Regarding the temperature stratification shown in figure 10, having a similar behavior of experiment 2 (figure 8), with the difference that in this case the sensor (T1) shows a temperature below the other sensor (T2), which means that the air stream passes to greater heights in this case, as expected.

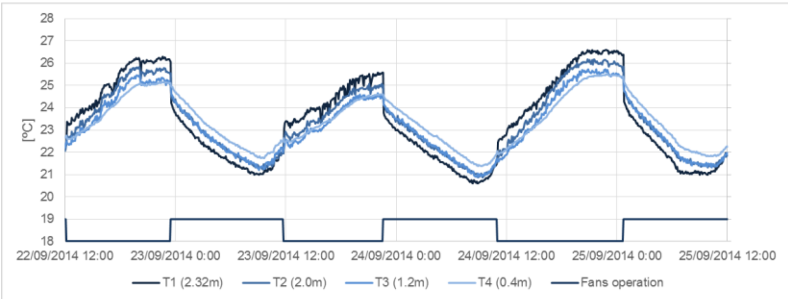


figure 10 Evolution of the interior temperatures at four levels in the first series of days selected in experiment 3

In order to search for lower power consumption of the fans, the value of the temperature difference widened so that it would began to ventilation, the initial value of 4 ° C was passed to 6C, that is, the fans only turned on if the outside temperature was 6 ° C lower than the interior.

The behaviour of the average test cell temperature and ceiling with the new mode of operation is shown in figure 11. It is noted that the operation time of cooling is strongly reduced, but also the decrease of the indoor temperature when ventilation works, is much lower than the previous cases. The oscillation amplitude of the indoor temperature is similar to the case of Experiment 1, namely, without night ventilation values.

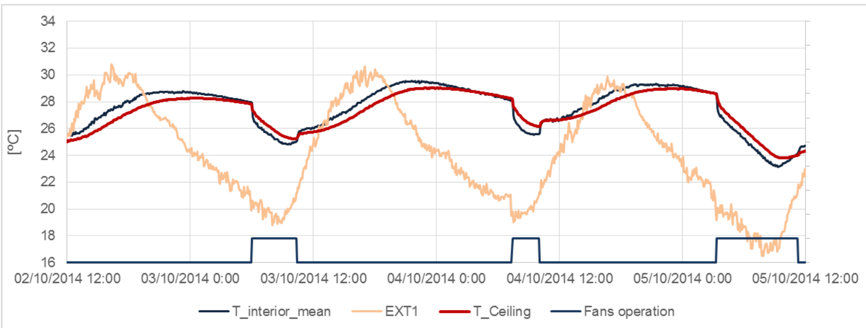


figure 11 Evolution of the interior, exterior and ceiling temperatures in the night ventilation experiment with deflector. Second series of days selected in experiment 3

The stratification of indoor temperatures shows an analogous behavior to that obtained in the case where the control strategy kept the start of fans set for a 4 ° C difference between outside and inside, as can be seen in the figure 12.

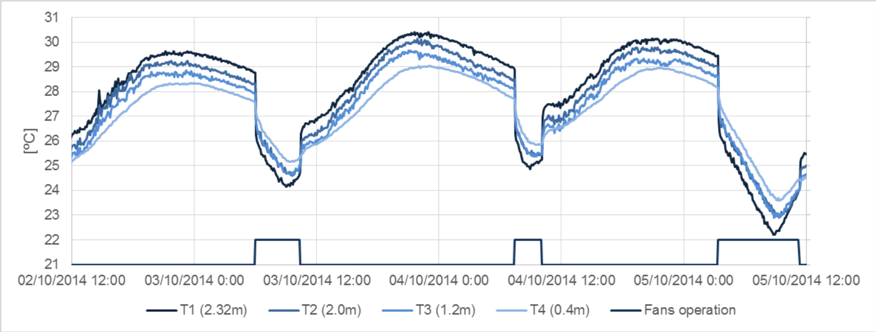


figure 12 Evolution of the interior temperatures at four levels in the first series of days selected in experiment 3

4.4 Summary of the three experiments

In order to obtain an overview of the three experiments and their results, we have condensed the data of the most relevant variables to daily values.

Daily exterior and interior average temperatures are shown in figure 13. The first and obvious observation was that the average temperature of the interior test cell is always higher than the outside. This is because the internal and solar gains through walls and roof are greater than the heat removed by ventilation. Therefore, the smaller the difference between the mean outside temperature and the inside, the better the comfort inside the test cell.

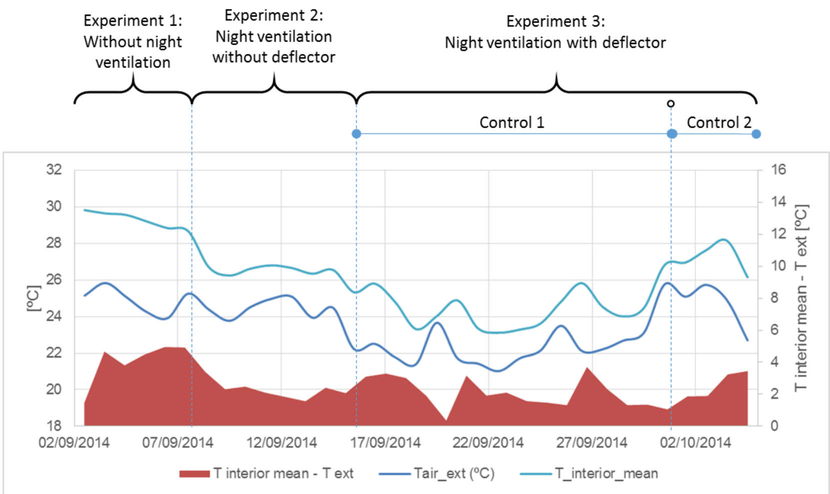


figure 13. Daily mean interior and exterior temperatures

figure 13 also it shows the difference between indoor and outdoor temperature. This temperature difference is lower when night ventilation is applied. In this figure there is not clear difference between the average results of Experiment 2 and Experiment 3, which means that the average daily temperature obtained with and without deflector (with and without thermal inertia) is very similar.

In figure 14, the daily average temperatures of ceiling and interior the test cell is show. In this case, the temperature of the ceiling in experiment 1 and experiment 2 shows higher values than the indoor air. In experiment 3, the ceiling temperature is inferior than that of air. This indicates two things:

1. In experiments 1 and 2, the ceiling is an element that provides heat to the interior while in experiment 3 the cover removed heat.
2. The only plausible explanation for the roof show a lower temperature than that shown in the interior of the cell in Experiment 3, it is that the cover has cooled during the hours of night ventilation. On another hand, because on Experiment 1 is impossible the cold accumulation, and because in Experiment 2 the temperature difference between ceiling and indoor is similar, we can say that the cold storage in experiment 2 is very poor.

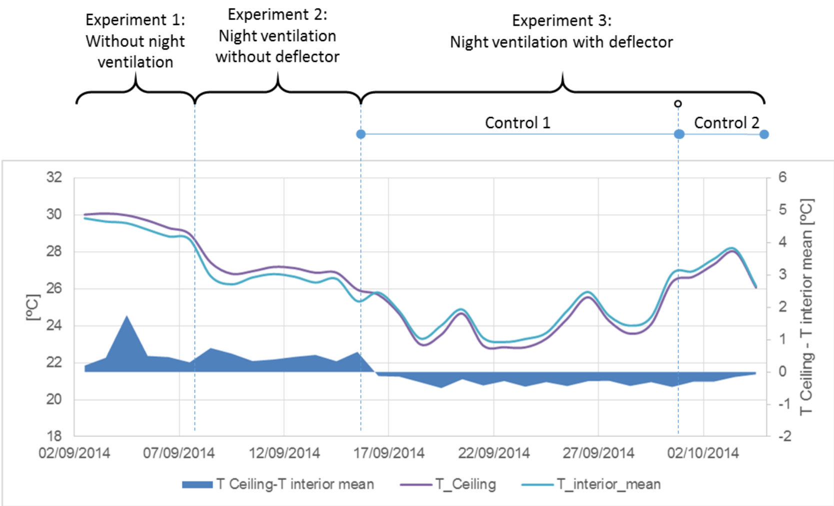


figure 14. Daily mean interior and ceiling temperatures

An extra benefit of the ceiling having inferior temperatures than the indoor air, is the increased comfort conditions, since the operating temperature of the building would be lower in the case of Experiment 3.

An overall summary of three experiments is shown in Table 1. These data ratify the observations done before, that is, the mean temperature in the test cells is lower with night ventilation and this temperature is approximately the same with or without cold storage on the slab. Also, with the experiment 3. The ceiling temperature is lower, indicating that there has been cold storage and in experiment 2, this storage is poor.

Table 1. Global summary of mean temperature differences and temperature oscillation for the three experiments carried out

	Mean Text - Tint	Mean Tceil - Tint	Mean exterior daily temperature oscillation	Mean interior daily temperature oscillation
	[ºC]	[ºC]	[ºC]	[ºC]
Without night ventilation	4.6	0.7	12.6	3.7
Night ventilation without deflector	2.1	0.5	12.5	5.4
Night ventilation with deflector (control 1)	2.2	-0.3	10.2	4.0
Night ventilation with deflector (control 2)	2.3	-0.3	12.7	4.5

However, observing the daily variation of indoor temperature, we see something that had not been evident previously, is that the temperature variation with cold storage on the slab (experiment 3), is significantly lower than without this storage. This is verified by observing Figures 7, 9 and 11. Therefore, although the daily mean indoor temperature is similar with and without cold storage, the difference between the minimum and maximum indoor temperature is lower when cold is storage, producing higher comfort in the occupants.

5 CONCLUSIONS

Three experiments performed in a tests cells located near to Seville Spain, has shown that night ventilation is useful to reduce the daily mean interior temperature approximately 2ºC and this value is independent of the activation of the thermal mass. Nevertheless, when the air stream allows to activate the thermal mass, the mean daily oscillation temperature is approximately 1ºC lower, indicating that daytime temperatures are around 0.5ºC inferior compared with the case in which thermal mass ion not activate.

6 ACKNOWLEDGEMENTS

The authors would like to thank the FEDER of European Union for financial support via project “Análisis del comportamiento energético de los cerramientos de hormigón en base a la maximización de las ventajas derivadas de su inercia térmica“ of the “Programa operativo FEDER de Andalucia 2012-2014”. We also thank all Agency of Public Works of Andalusia Regional Goberment staff and reserchers for their dedication and professionalism. Special thanks to "Grupo Cementos Valderrivas, Alcalá de Guadaira plant" for his selfless assistance in developing the experiments.

7 REFERENCES

Artmann, N., Manz, H. & Heiselberg, P., 2007. Climatic potential for passive cooling of buildings by night-time ventilation in Europe. *Applied Energy*, 84(2), pp.187–201..

Corgnati, S.P. & Kindinis, A., 2007. Thermal mass activation by hollow core slab coupled with night ventilation to reduce summer cooling loads. *Building and Environment*, 42(9), pp.3285–3297..

Geros, V. et al., 1999. Experimental evaluation of night ventilation phenomena. *Energy and Buildings*, 29(2), pp.141–154.

Irulegi, O. et al., 2014. Potential of Night Ventilative Cooling Strategies in Office Buildings in Spain - Comfort Analysis. *The International Journal of Ventilation*, pp.193–210.

Kolokotroni, M. & Aronis, A., 1999. Cooling-energy reduction in air-conditioned offices by using night ventilation. *Applied Energy*, 63(4), pp.241–253.

Macias, M. et al., 2006. Application of night cooling concept to social housing design in dry hot climate. *Energy and Buildings*, 38(9), pp.1104–1110.

Pfafferott, J., Herkel, S. & Jäschke, M., 2003. Design of passive cooling by night ventilation: evaluation of a parametric model and building simulation with measurements. *Energy and Buildings*, 35(11), pp.1129–1143.

HOW SAFE IS IT TO NEGLECT THERMAL RADIATION IN INDOOR ENVIRONMENT MODELING WITH HIGH VENTILATION RATES?

Sasan Sadrizadeh* and Sture Holmberg

Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Stockholm, Sweden
**Corresponding email: ssad@kth.se*

ABSTRACT

Typical heat sources in indoor environments include humans, electrical devices, and computers. The number of such sources in operating room environments is even higher due to the presence of surgical staff members and medical equipment. The exchange of thermal energy between indoor surfaces and air is usually modelled by considering contributions from both radiation and convection. Complete heat transfer simulations in indoor environments are normally difficult since radiation models have a tendency to generate numerical instability and, hence, problems with convergent solutions. In the past, thermal radiation influence on indoor airflow movement and contaminant distribution was rarely addressed. This study therefore focused on evaluating the influence of radiative heat transfer in an operating room with 40 air changes per hour.

For the purpose of the current study, an identical case with and without considering the radiative part of heat transfer was simulated and compared. In both simulated cases, the numerical results have shown negligible changes in terms of indoor temperature and air velocity, and thus contaminant distribution. It was also observed that temperature differences between heat-emitting surfaces were negligible. It is therefore reasonable to conclude that it is safe to disregard the radiative part of heat transfer in indoor air simulations with high ventilation rates and heat transfer can here be modelled as pure convection.

KEYWORDS: heat transfer, radiation modeling, operating room, contaminant (particle) distribution, computational fluid dynamics (CFD), numerical simulation

1. INTRODUCTION

Indoor airflow simulations can predict contaminant dispersion and provide valuable information regarding indoor air quality, especially in sensitive environments such as operating rooms (ORs). With recent advances in computer capability and speed, computational fluid dynamics (CFD) has become a powerful alternative for predicting airflows in enclosed environments. Simulation accuracy greatly depends on the proper setting of boundary conditions and numerical simulation parameters.

The precise setting of thermal boundary conditions is required for correct prediction of indoor environment temperature distribution. This may result in precise airflow calculation and thus airborne particle concentration. A local indoor temperature that is miscalculated by 2 °C can result in determining an incorrect airflow pattern, even opposite direction compared to the real airflow patterns (Yuan & Srebric, 2004). This may generate substantial errors in determining the distribution of indoor contaminants, which usually follow airflow streamlines closely (Sadrizadeh & Holmberg, 2014a; Sadrizadeh, Tammelin, Ekolind, & Holmberg, 2014). Typical CFD simulations were usually considered the sensible heat loads as 100 % convective (Sadrizadeh, Holmberg, & Tammelin, 2014). This was a simplified treatment since a portion of the sensible load should be radiative in nature.

Complete heat transfer simulations in indoor environments are difficult since radiation modelling has a tendency to generate numerical instability and thus problems with convergent solutions. Thus, thermal radiation influence on indoor airflow simulations was seldom addressed.

This study is therefore focused on evaluating the influence of radiative part of heat transfer on temperature and airflow velocity in an operating room with 40 air changes per hour (ACH).

2. METHODOLOGIES

A single-zone standard OR, which was adapted from the authors' previous work (Sadrizadeh & Holmberg, 2014b; Sadrizadeh, Tammelin, Nielsen, & Holmberg, 2014), was chosen as the physical model for this study. The OR measured 8.5 m □ 7.7 m □ 3.2 m (H), with the physical configuration shown in Fig. 1.

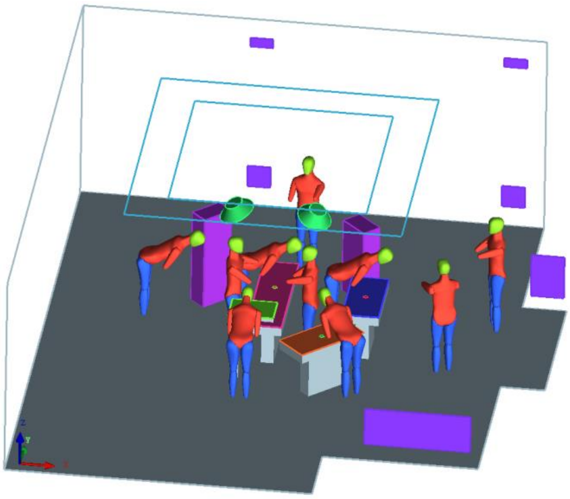


Fig. 1: Isometric view of operating room

The supply air was directed to the OR, with a total volume flow rate of 2,000 L/s, giving a design exchange rate of 40 h^{-1}. Supply air temperature and turbulence intensity were 20 °C and 5 percent respectively. Exhaust air was expelled out through six grilles placed on three parallel vertical walls. An operating table, two instrument tables, and one mayo stand table were considered within the surgical zone. Two pieces of medical equipment, each with thermal load of 255 W, were also included. Two medical lamps were fixed above surgical area; each emitted a heat load of 320 W. Ten staff members were considered in both "bend" and "straighten up" postures. Any of the simulated staff members with 1.6 m^2 total surface area emits a thermal heat load of 195 W.

The mean airflow field was solved using the realizable k-ε turbulence model. This model is widely used for indoor environments due to its relative simplicity and robustness for indoor airflow simulations. The governing equations can be written in the general format as follows:

$$\frac{\partial}{\partial \mathrm{t}}(\rho\varphi) + \nabla.\left(\rho\varphi\vec{V} - \Gamma_{\varphi}\nabla\varphi\right) = \mathrm{S}_{\varphi} \qquad \text{Eq. 1}$$

where ρ is the air density, φ represents each of the three velocity components u, v, w and $\vec{V}$ is the velocity vector. S_{φ} is the source term and Γ_{φ} is the effective diffusion coefficient for each dependent variable.

The above-mentioned equations are discretized into algebraic equations by the finite volume method and solved by ANSYS Fluent 15.0. The discretized method for convection terms is a second-order upwind scheme. The SIMPLE algorithm was adapted to couple pressure and velocity. Enhanced wall treatment was employed to treat the turbulent airflow properties in the near-wall regions. The numerical models were previously validated against experimental data (Sadrizadeh, Holmberg, et al., 2014; Sadrizadeh, Tammelin, Ekolind, et al., 2014) and are not repeated here.

In the current study, the sensible heat loads of an identical case were considered once as "100 % convective" and once as "combination of radiative and convective". For this purpose, radiative heat transfer was simulated using discrete ordinates (DO) radiation model theory, which solves the radiation transfer equation for a finite number of discrete solid angles (in this study 32). Here, all rigid surfaces were treated as sources of opaque gray (emissivity of 0.95) radiation.

3. RESULTS AND DISCUSSIONS

In general, particle distribution is highly dependent on Stokes number (Stk). Stk is defined as the ratio of the characteristic time of a droplet to a characteristic time of the airflow or of an obstacle. Particles with Stk < 0.1 will follow airflow streamlines closely (Gao & Niu, 2007). The authors have previously shown that the Stocks number of common particles with the surgical area have the stocks number of less than 10^{-3} (Sadrizadeh & Holmberg, 2014a; Sadrizadeh, Tammelin, Ekolind, et al., 2014); therefore, the contaminants will follow the airflow streamlines.

Fig. 2 shows the location of eight lines across the OR, which the airflow temperature and velocities exported.

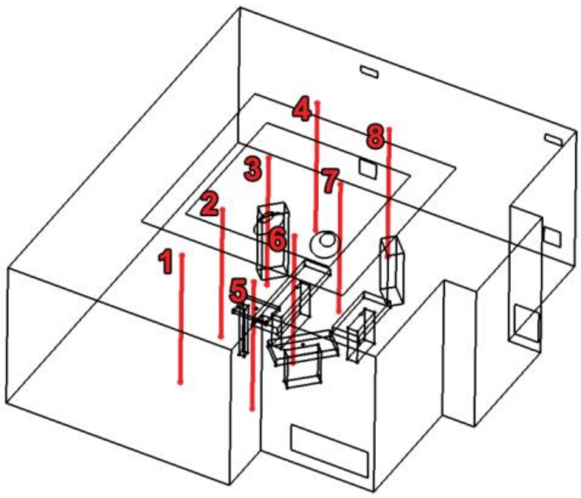


Fig. 2: Eight vertical lines representing the location of exported temperature and velocity data

Fig. 3 shows the ratio of the temperature and velocity magnitude, with (T_r, V_r) and without (T, V) considering the radiative part of heat transfer.

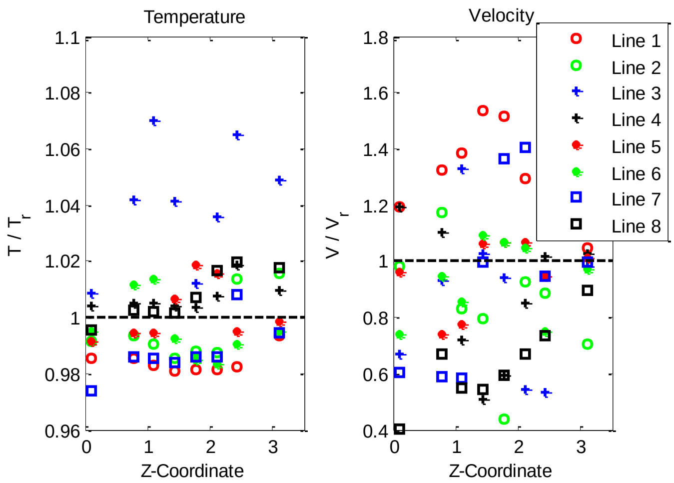


Fig. 3: Ratio of temperatures, one with considering the radiative heat transfer (T_r) and one without (T) [left]; and ratio of velocities with (V_r) and without (V) considering the radiative heat transfer [right]

It is clear that the temperature difference between the results obtained with and without considered the radiative part of heat transfer is negligible. This difference is much higher in case of the velocity ratio (V/V_r). This might be due to the velocities being less than 0.4 m/s, as small differences may result in quite high ratios. In operating room environments, since the air exchange rate is usually very high compared to the normal indoor air situations, the convective part of heat transfer become highly dominant. Therefore, most of the heat transfer will be handled by convection. Radiative part heat transfer thus has a negligible effect and can be disregarded.

4. CONCLUSIONS

In order to access the effect of thermal radiation in indoor airflow simulations with high ventilation rates, an identical operating room was simulated. The sensible heat loads were either taken as 100 percent convective or a combination of radiative and convective. The discrete radiation model coordinates handle the thermal radiation effect. In both cases, the numerical results indicate negligible changes in indoor air velocity and temperature distribution and thus on particle concentration. However, simulation of velocity and contaminants may require further assessment in establishing the influence of simplifications.

5. REFERENCES

Gao, N. P., & Niu, J. L. (2007). Modeling particle dispersion and deposition in indoor environments. Atmospheric Environment, 41(18), 3862–3876.

Sadrizadeh, S., & Holmberg, S. (2014a). Effect of a portable ultra-clean exponential airflow unit on the particle distribution in an operating room. Particuology, DOI: 10.1016/j.partic.2014.06.002, [In Press].

Sadrizadeh, S., & Holmberg, S. (2014b). Surgical clothing systems in laminar airflow operating room: a numerical assessment. Journal of Infection and Public Health, 7(6), 508–516.

Sadrizadeh, S., Holmberg, S., & Tammelin, A. (2014). A comparison of vertical and horizontal laminar ventilation systems in an operating room: A numerical study. Building and Environment, 82, 517–525.

Sadrizadeh, S., Tammelin, A., Ekolind, P., & Holmberg, S. (2014). Influence of staff number and internal constellation on surgical site infection in an operating room. Particuology, 13, 42–51.

Sadrizadeh, S., Tammelin, A., Nielsen, P. V, & Holmberg, S. (2014). Does a mobile laminar airflow screen reduce bacterial contamination in the operating room? a numerical study using computational fluid dynamics technique. Patient Safety in Surgery, 8(1), 27.

Yuan, J., & Srebric, J. (2004). Transient prediction of contaminant distribution by introducing energy load calculations into multi-zone modeling. In CIB World Building Congress 2004 (Vol. 148, pp. 2–7). Toronto, Canada.

3D FLUID DYNAMIC ANALYSES OF OPEN JOINT VENTILATED FACADES APPLYING EXPERIMENTAL STEREO-PIV TECHNIQUES.

M. Nuria Sánchez [*1], Cristina Sanjuan [1] and M. Rosario Heras [1]

1 Department of Energy, Energy Efficiency in Buildings Unit, CIEMAT
Av. Complutense, 40. Madrid 28040, Spain
**Corresponding author: nuria.sanchez@ciemat.es*

ABSTRACT

Open joint ventilated façade (OJVF) is a passive constructive system widely used to ameliorate envelop of buildings, improving their energy efficiency. A laboratory model of a ventilated façade with horizontal and vertical open joints was used to study thermal and fluid-dynamic behaviour of this system. Experimental Stereo Particle Image Velocimetry (Stereo-PIV 2D-3C) was applied to study the fluid dynamic performance of this model. In addition, thermal analyses were performed applying infrared thermography and air temperature monitoring inside the ventilated cavity. Experimental measurements were carried out inside the air cavity in a vertical joint plane and in a middle plane of panels, corresponding to horizontal joints. Radiation condition corresponds to $1x10^9$ Ra and wind effect is not considered.
The analysis of the data confirms previous results of velocity and temperature patterns for OJVF with horizontal joints and leads to identify two different velocity patterns corresponding to the airflow entrance through horizontal and vertical joints. Experimental results will be used for validate computational fluid dynamics (CFD) models of the OJVF system.

KEYWORDS

Ventilated facade; natural convection; Stereo-PIV.

1 INTRODUCTION

The four major energy end-use sectors are commercial, residential, industrial and transportation. Commercial and residential buildings account for over 40% of total final energy consumption (EIA, 2014), representing cooling loads an important percentage of it. To reduce energy consumption in this sector, a combination of best available technologies and public policies has to be adopted.
Different passive systems are implemented in the built environment for enhancement of sustainability, achieving the goal of saving energy while maintaining thermal comfort conditions (GhaffarianHoseini et al., 2012 and Romila et al., 2012).
A commonly used system are Open joint ventilated façades (OJVF), reducing the heat transferred to the building and consequently the cooling loads. The solar radiation incident on the external surface of the OJVF produces an ascending airflow, induced by the buoyancy effect, which extracts part of the heat to the ambient air.

The system, composed by multiple layers, is characterized by an external opaque coating and a ventilated air cavity formed between the envelope and the insulation layer attached to outer side of the wall mass (Figure 1). The exterior coat is made of tiles hanged to the structure forming open joints between them, horizontally or vertically oriented.

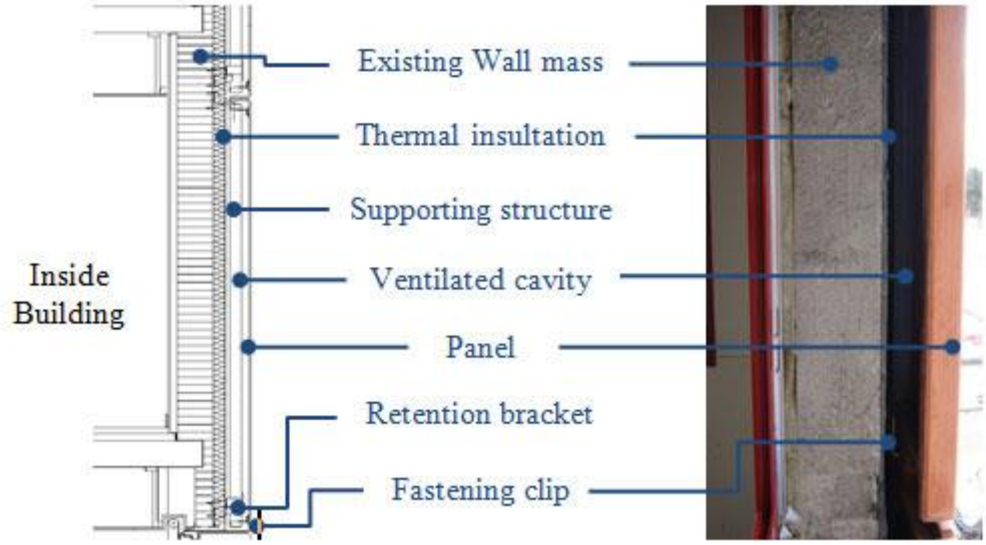


Figure 1: From left to right: open joint ventilated façade design and image of the component installation during building refurbishment.

Several authors assess experimentally the effect on the performance of the system of different constructive parameters: height of the ventilated camara, facade orientation, environmental conditions or material of panels (Marinosci et al., 2014 and Stazi et al., 2014). Moreover, several studies evaluate the performance of OJVF with horizontal joints based on 2D-PIV, a non-intrusive particle based technique used in flow velocity measurement (Sánchez et al., 2013 and Sanjuan et al., 2011 and 2012). This optical method of flow visualization and quantification of instantaneous velocity fields, measures two velocity components in the area of analysis. The measurement principles and technology description have been reviewed by Cao et al., (2014) dealing with the applications in indoor environment. The current research is focused on analyse the thermal and energy behaviour of OJVF with vertical and horizontal open joints, determining the role of orientation in the performance of the component.

2 EXPERIMENTAL SETUP

2.1 Model Design

An experimental model of a ventilated façade with horizontal and vertical open joints was designed and constructed to ensure the natural ventilation flow simulation in analogous conditions to real façades. In this model, the exterior coat is made up of 16 metallic panels disposed in a symmetrical distribution of 4 rows and 4 columns with 5 mm horizontal and vertical open joints (Figure 2).

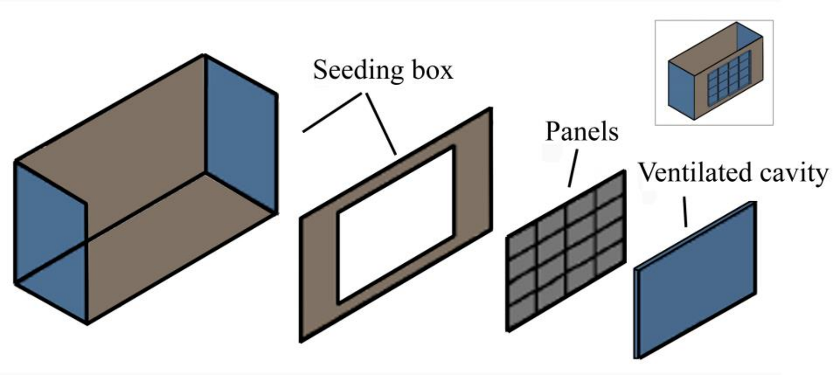


Figure 2: Exploded diagram of the experimental model of a ventilated façade with horizontal and vertical open joints.

Overall dimensions of the facade were 0.825 m high and 1.225 m wide with a ventilated cavity of 0.045 m. Additionally, a seeding box was rigidly attached to the ventilated facade on its outer surface, simulating the outdoor environment.

The design is based on simplified real facades required for experiments in the laboratory setting. The dimensions of the cavity and the joints correspond to real façades dimensions while the height of the model has been reduced to a third of the distance between window panes in real buildings. Moreover, thermal insulation and wall mass layers were replaced by a glass layer to allow optical access to the ventilated cavity. These limitations are mainly due to the height of the laboratory and the specific requirements of the experimental technique used for the characterization of the ventilation flow.

The natural ventilation airflow inside the camera was induced by heating the external surface of the panels using auto-adhesive electrical heater mats; simulating the solar radiation effect.

2.2 Experimental measurement techniques

Particle Image Velocimetry measures instantaneous flow fields capturing two images shortly, one after each other. Statistical correlations are used to find average tracer particle displacement (illuminated by a sheet of light) within this time. From the known time difference and the measured displacement, the instantaneous velocity field is calculated. Stereo-PIV (2D3C) unlike PIV (2D2C) uses two points of view (two cameras) and images are recorded simultaneously by left and right cameras. A numerical model describing how objects in 3-dimensional space are mapped onto the 2-dimensional image (recorded by each of the cameras) is used to estimate the third velocity component in the area of analysis (Hinsch, 1995).

The Stereo-PIV technique was applied to study the fluid dynamic performance of the constructed OJVF model. The three velocity components of the ventilation airflow were measured, obtaining the instantaneous velocity field. A rotation configuration arrangement with cameras on either side of this plane was performed. In order to achieve a maximal overlap of the area covered by both cameras in the stereoscopic PIV 3D-reconstruction, viewing angles of the cameras were set to 45º. The true 3D displacement (ΔX, ΔY,ΔZ) is estimated from a pair of 2D displacements (Δx,Δy) as seen from left and right camera respectively (Figure 3).

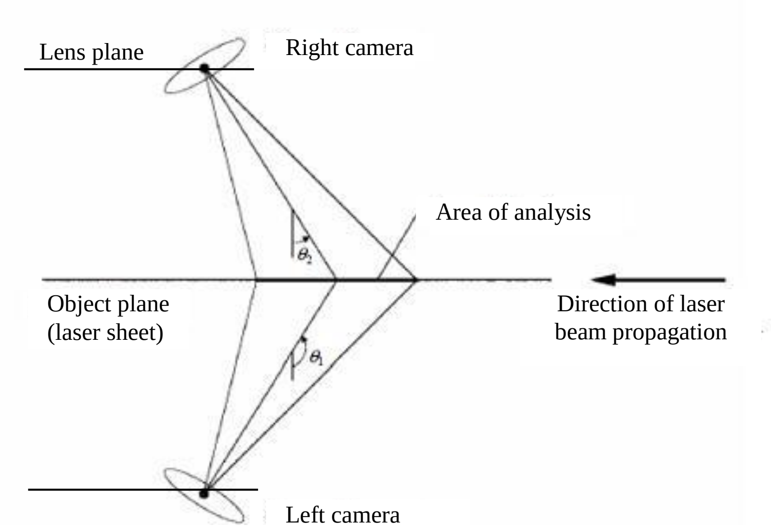


Figure 3: Schematic of the angular displacement system in Stereo-PIV.

One micron olive oil droplets were used as tracer particles; introduced into the air inside the seeding box. The image density was larger than 15 particles and air flow was homogeneously seeding. The value of the Stokes numbers for the experiments is in the range of 10^{-6} indicating that the particles follow the fluid flow.

Vertical dimension of the active area of the cameras was almost one order of magnitude lower than the cavity height (15cm compared with 82cm). As a result, the experimental device was eight times vertically displaced 10cm up to the maximum height of the air cavity. Measurements registered at the eight different runs were reconstructed, obtaining the whole velocity field in vertical plane. 250 snapshots were recorded for each run with a frequency of 17 Hz. The time between pulses was 10^{-4} s.

A correct calibration is an essential prerequisite for measuring accurately the three velocity components. Thus, Stereo Automapping was applied in order to correct the misalignment between calibration plate and laser light sheet, resulting one of the major sources of error in Stereo-PIV. This self-calibration procedure computes the cross-correlation and disparity map between images of left and right camera recorded at the same time.

Image processing was done using multi-grid algorithm with 64x64 pixel initial window size and 32x32 pixel final window size with an overlap of 50%. At a later stage, spurious vectors were eliminated from the flow field in the post-processing.

In addition, thermal analyses were performed applying infrared thermography and air temperature monitoring. The temperature measurements were performed using an infrared thermograph camera and thermocouple sensors, respectively.

Both, temperature and velocity measurements, were obtained in two different vertical planes of the air cavity, corresponding to a middle plane characterized by the horizontal joints and a vertical joint plane.

3. RESULTS

This research is conducted considering solar radiation corresponding to $1x10^9$ Rayleigh and without considering the wind effect. Relevant thermal experimental conditions are: the heater mats powered by 10 V dc within a power rating of 28W simulating an absorbed solar radiation of 460W/m2, the 51ºC mean temperature of slabs, the 29ºC seeding temperature and the calculated Stokes number of $1.01x10^9$.

3.1 Mean Velocity Field

Based on real-time field velocity, it is possible to calculate other factors involved in fluid dynamics such as average velocity field. Instantaneous velocity can be regarded as consisting of an average value u indicated by the dashed line, plus a random fluctuation u_i' (Equation 1).

$$u_i(x,y,z) = \bar{u}(x,y,z) + u_i'(x,y,z) \tag{1}$$

For computing the averaged velocity vector fields, only vectors from snapshots with time correlation factors higher that 99% were used in the mean flow calculation.

Figure 4 shows time-averaged velocity field in a middle-panel plane of measurement at three different cavity heights (bottom, middle and top). Similarly, Figure 5 shows time-averaged velocity field in a joint plane of measurement at three different cavity heights (bottom, middle and top). Maximum time-averaged velocity field was 0.35m/s, at the bottom of the air cavity in the joint plane.

Significantly, two different patterns were identified for the same Rayleigh value, corresponding to the airflow entrance through horizontal and vertical joints. Figure 4 and Figure 5 shows these two patterns especially at the bottom of the cavity. In both patterns the air enters the ventilated cavity through the lower open joints and leaves it through the upper joints. At half height of the air cavity, the pressure equilibrium prevents the exterior air not enter or leave the cavity through the joints.

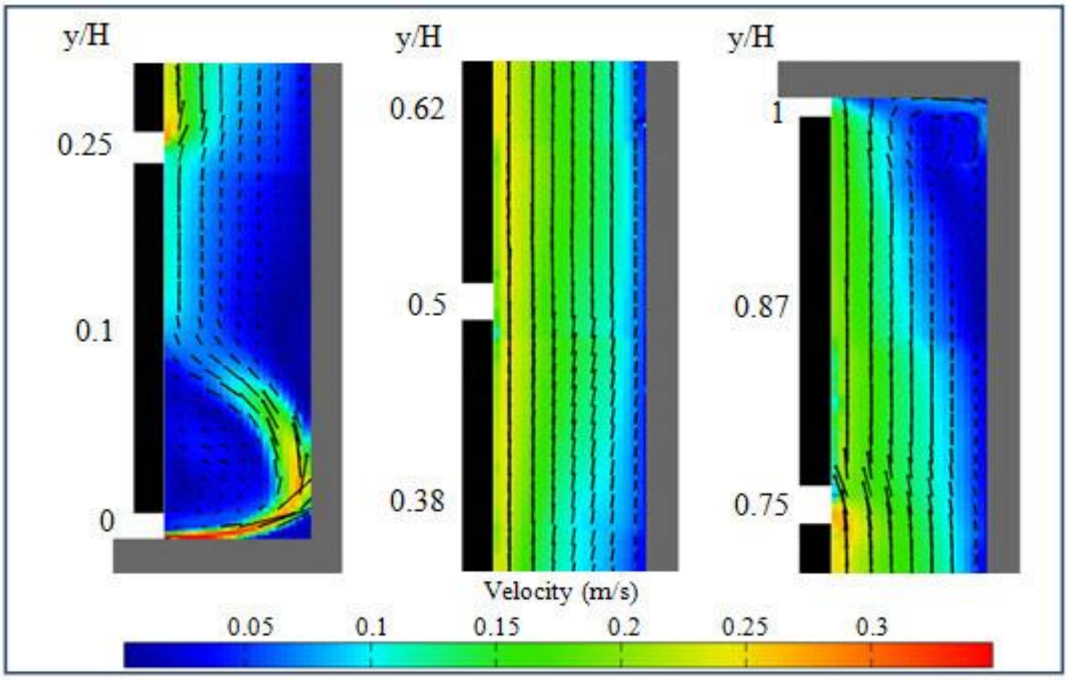


Figure 4: From left to right: time-averaged velocity field in a middle plane of measurement at three different cavity heights.

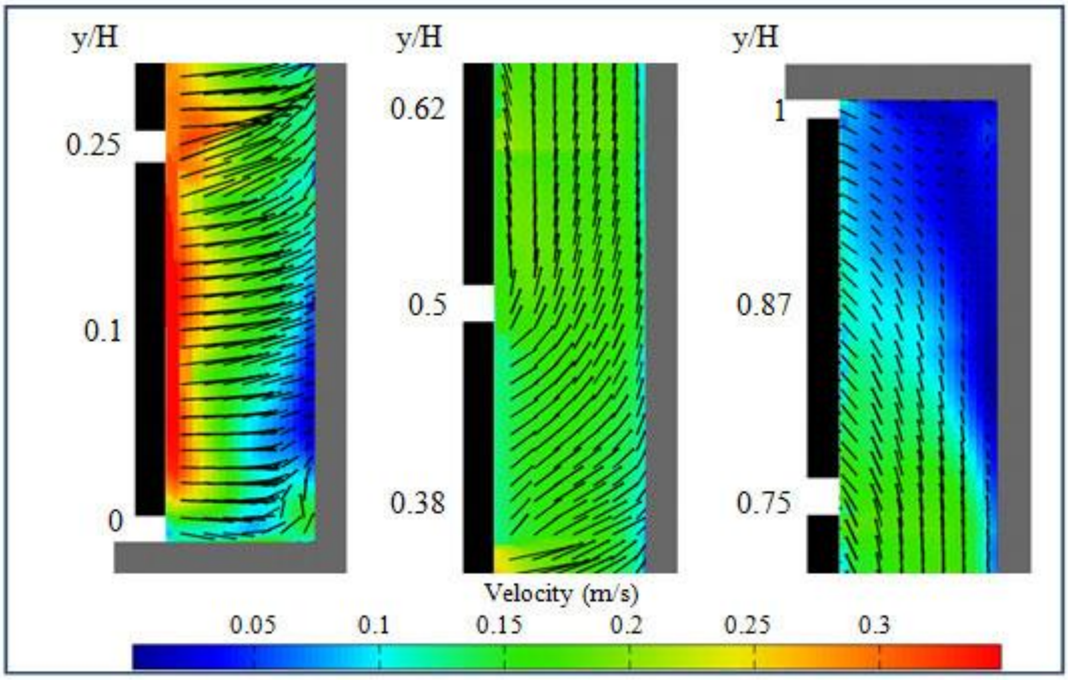


Figure 5: From left to right: time-averaged velocity field in a joint plane of measurement at three different cavity heights.

The buoyancy effect created inside the air cavity by heating the external surface of the panels produces an ascending ventilation flow that circulates along the cavity at an inhomogeneous rate. The abrupt entrance of the airflow through the lower horizontal joints generates recirculation vortexes (Figure 4); complex fluid structures due to the presence of slabs along the wall. However no recirculation vortexes were identified at the lower vertical joints because the fluid enters the cavity without interruption throughout the open vertical joints (Figure 5).

3.2 Turbulence

Derived magnitudes from velocity such as turbulence were calculated to further characterize the air movement inside the cavity (Equation 2).

$$Tu(x,y,z) = \sqrt{\frac{2}{3}K(x,y,z)} \qquad (2)$$

Kinetic energy corresponding to the random velocity component is referred to as turbulence kinetic energy (K) and calculated by the following equation (Equation 3).

$$K(x,y,z) = \frac{1}{2N}\sum_{i=1}^{N}[u'^2_i(x,y,z) + v'^2_i(x,y,z) + w'^2_i(x,y,z)] \qquad (2)$$

N is the number of snapshots.

Figure 6 shows turbulence level in a middle-panel plane of measurement at three different cavity heights (bottom, middle and top). Similarly, Figure 7 shows turbulence level in a joint plane of measurement at three different cavity heights (bottom, middle and top).

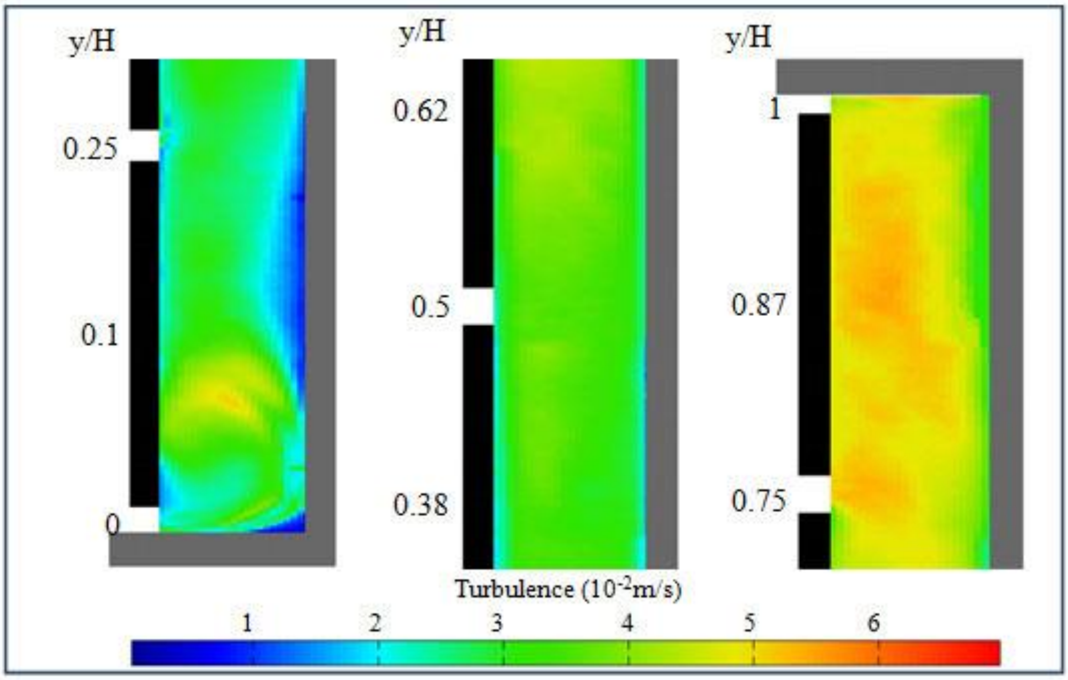


Figure 6: From left to right: turbulence in a middle plane of measurement at three different cavity heights.

As it would be expected, the turbulence level increases with the mean flow velocity, quantifying the oscillation part of the velocity. Maximum turbulence level was 0.07 m/s, determined at the bottom of the air cavity in the joint plane.

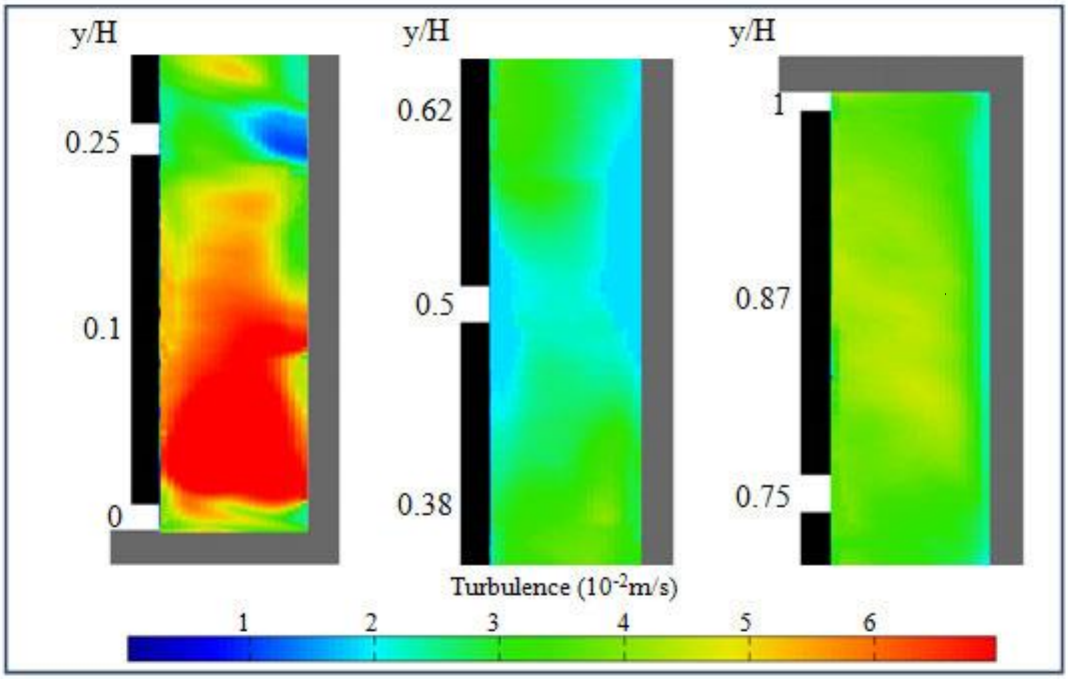


Figure 7: From left to right: turbulence in a joint plane of measurement at three different cavity heights.

3.3 Temperature measurements

Simultaneously to the performance of the Stereo-PIV experiments, the OJVF model was monitored measuring the temperatures in the outer surface, the interior air cavity, and the environmental conditions. Figure 8 represents summarized time-averaged temperatures measured during the experiments corresponding to the two vertical planes (middle-panel plane and joint plane).

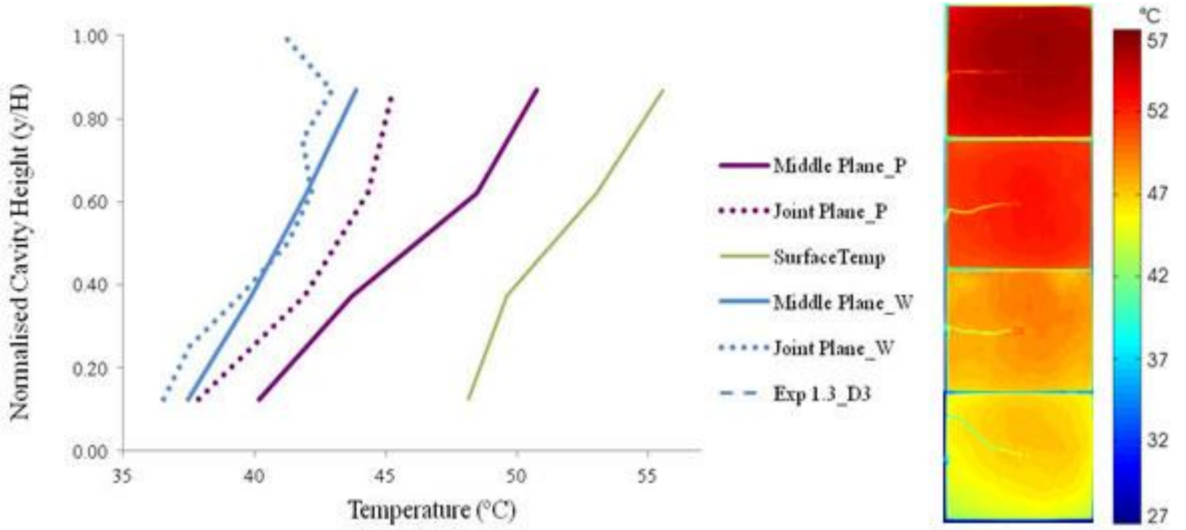


Figure 8: From left to right: air temperature at the centre of the ventilated cavity and surface temperature of the panels.

The temperatures monitored close to the entrance of the channel shows higher values of the air temperature at the middle plane compared to the joint plane. The temperatures monitored close to the wall mass of the channel show similar values of the air temperature at the middle plane and the joint plane The surface temperature at the centre of the panels ranged from 48ºC to 56ºC.

Additional measurements include infrared thermography in selected monitored panels. The surface temperature registered with a thermographic camera also shows higher temperature values at the centre of the plates according to the previous measurements. The distribution of superficial temperature in the panels is not homogeneous. Temperature values are lower near the edges due to the ventilation airflow through the joints.

4. CONCLUSIONS

This study concludes that airflow inside the cavity can be considered as a steady turbulent flow. Two different patterns were identified corresponding to the airflow entrance through horizontal and vertical joints with a maximum time-averaged velocity field of 0.35m/s. The evolution of the velocity profiles along the cavity was different. In both patterns the air enters the ventilated cavity through the lower open joints and leaves it through the upper joints. The abrupt entrance of the airflow through the lower horizontal joints generates recirculation vortexes. However, the evolution of the velocity profiles along the cavity was different.

The temperatures monitoring shows higher values of the air temperature at the middle plane compared to the joint plane. The air temperature increased due to the heat exchange of the panels and the ventilated cavity walls with the ascending flow. The surface temperature registered with a thermographic camera shows higher temperature values at the upper panels and higher temperature values at the centre of the panels.

The fluid pattern corresponding to the middle plane (horizontal joint) reproduces well the previous studies (Sánchez et al., 2013 and Sanjuan et al., 2011).
These experimental results help to validate CFD models to predict accurately the heat transferred to the building. Besides, additional studies considering different radiation conditions and several relevant constructive parameters have to be done.

5. ACKNOWLEDGEMENTS

This work has been developed in the framework of the OMEGA-CM programme and the experimental facility was initially developed under the PSE-ARFRISOL project. The OMEGA-CM programme, ref. P2013/MAE2835, is a multidisciplinary R&D programme supported by the Madrid Regional Government and co-financed by EU Structural Funds. The PSE-ARFRISOL, ref. PSE-120000-2005-1, is a scientific-technical research project of singular character, supported by the Spanish Ministry of Science and Innovation and co-financed by FEDER funds. The authors thank OMEGA-CM and ARFRISOL members.

6. REFERENCES

EIA. (2014). Independent Statistics and Analysis U.S. Energy Information Administration. www.eia.gov (Access June 2015).

GhaffarianHoseini A.H., Berardi U., Ali GhaffarianHoseini A. and Makaremi N. (2012). Intelligent Facades in Low-Energy Buildings. *British Journal of Environment and Climate Change*, 2(4), 437-464.

Romila C., Popovici C.G., Cherecheş N.C. (2012). Reduction of building energy consumption using ventilated façades. *Environmental Engineering and Management Journal*, 11(4), 806-811.

Marinosci C., Semprini G., Morini G.L. (2014). Experimental analysis of the summer thermal performances of a naturally ventilated rainscreen façade building. *Energy and Buildings*, 72, 280–287.

Stazi F., Vegliòa A., Di Perna C. (2014). Experimental assessment of a zinc-titanium ventilated fac¸ ade in a Mediterranean climate. *Energy and Buildings*, 69, 525–534.

Sánchez M.N., Sanjuan C., Suárez M.J., Heras M.R. (2013). Experimental assessment of the performance of open joint ventilated façades with buoyancy-driven airflow. *Solar Energy*, 91, 131-144.

Sanjuan C., Sánchez M.N., Heras M.R., Blanco E. (2011). Experimental analysis of natural convection in open joint ventilated façades with 2D PIV. *Building & Environment*, 46(11), 2314-2325.

Sanjuan C., Sánchez M.N., Enríquez R., Heras M.R. (2012). Experimental PIV techniques applied to the analysis of natural convection in open joint ventilated facades. *Energy Procedia*, 30, 1216-1225.

Cao, X., Liu, J., Jiang, N., Chen, Q. (2014). Particle image velocimetry measurement of indoor airflow field: A review of the technologies and applications. *Energy and Buildings*, 69, 367-380.

Hinsch, K.D. (1995) Three-dimensional particle velocimetry. *Meas. Sci. Technol.*, 6, 742-753.

A STUDY ON THE REDUCTION EFFECT OF VENTILATION AND HEATING LOAD BY INSTALLING AIR-BASED SOLAR SYSTEM IN THE DETACHED HOUSES

Youngjin Choi*[1], Kozo Takase[2], Masayuki Mae[3], and Hyunwoo Roh[4]

1 The University of Tokyo
Hongo 7-3-1, Bunkyo-ku
Tokyo, Japan
**Corresponding author:choiyj11@gmail.com*

2 Tokyo University of Science
Yamazaki 2641, Noda
Chiba, Japan

3 The University of Tokyo
Hongo 7-3-1, Bunkyo-ku
Tokyo, Japan

4 OM solar
Murakushi -cho 4601, Hamamatsu
Shizuoka, Japan

ABSTRACT

Using solar heat energy has been paid attention to as effective natural energy use. In this study, we deal with air-based solar heat system, which is used for not only ventilation but heating and hot water supply by hot air.
In Japan, this system has been quite popular and installed into many buildings, but there is a few survey on the system by precise measurement. Then, we started the survey by building three huts with different specification and measure precise data of these in same weather condition. So we can analyse heat balance in these huts. Heat collection, thermal environment and heat storage are very important in the system. Especially, to use stored heat in night time is difficult because the solar heat is collected in the daytime only. The purpose of this research is to make some suggestion to improve the system by the measurement.

KEYWORDS

Air-based solar heat system, Experiment, Heat balance, Heat storage

1 INTRODUCTION

Using solar heat energy has been paid attention to as effective natural energy use. In this survey, we deal with air-based solar heat system, which is used for not only hot water supply but heating and ventilation by hot air. This system works as follows (shown in the Figure 1),

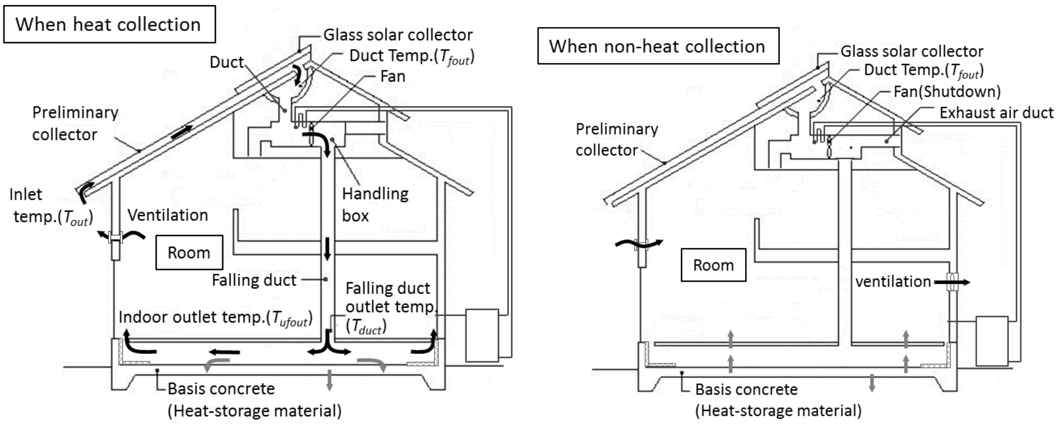


Figure 1: Conceptual diagram of the existing system

a) Heating outdoor air by solar collector on the whole roof.
b) Supply heat to water tank by antifreeze fluid which received heat by heat exchanger in the air handling box in the attic.
c) Blow hot air under the floor from the fan in the air handling box.
d) Store heat into the basis concrete under the floor by hot air.
e) Heating the whole room by supplying warm air from the under floor.

In this paper, we explain the system and the abstract of the measurement. The collector in air-based solar collect system is made up pre-heating collector of roof and glass collector for high temperature collection.

2 HEAT BALANCE AND THE POINTS OF IMPROVEMENT

2.1 Heat balance of the building with the system

The factors determining the performance of air-based collecting system, are the amount of collection, heat absorbing and radiating in the under floor, and heat loss by ventilation and heat flow-through. The performance of collector is calculated the difference of inlet (T_o) and outlet (T_{colout}) temperature and wind flow as shown in equation (1). The collection efficiency (η) is calculated by the ratio of solar radiation in the collector and the amount of collection (Q_{col}) as shown in equation (2).

$$Q_{col} = c\rho \cdot V \cdot (T_{colout} - T_o) \tag{1}$$
$$\eta = Q_{col}/(A_{col} \cdot I) \tag{2}$$
$$\rho = 353.25/(T_{colout} + 237.15) \tag{3}$$

In addition, the amount of collection of room temperature criteria, makes influence in the indoor heating load, is defined equation (4). In addition, the amount of collection of room temperature criteria, makes influence in the indoor heating load, is defined equation (4). As shown in equation (5), the absorbing heat into the heat storage material in the under floor is obtained from the difference of the temperature of under floor (T_{uf}) and the surface temperature of the heat storage material, and the convective heat transfer coefficient (α_c) of the heat storage surface.

$$Q_r = c\rho \cdot V \cdot (T_{colout} - T_r) \quad (4)$$

$$Q_{uf} = \alpha_c \cdot (T_{uf} - T_{hs}) \cdot A_{hs} \quad (5)$$
$$= F_{hs} \cdot A_{hs}$$

As shown in Figure 2, the heated air moves to the under floor, and then occurs the heat absorbing into the heat storage material as basis concrete (Q_{uf}), heat flow to the room (Q_f), and air flow into the room. The heat balance in the under floor is shown in equation (6).

$$Q_{vd} = c\rho \cdot V \cdot (T_{duct} - T_{ufout}) \quad (6)$$
$$= (c\rho \cdot Vol_{uf} \cdot \Delta T_{uf}) + Q_f + Q_{uf}$$

The heat from the floor surface into the room is used for room temperature rising, heat flow into the wall, ceiling, window (Q_{wall}, Q_{ceil}, Q_{win}), and the heat loss by ventilation (Q_v). The heat balance in the room in the absence of solar heat gain and internal heat generation is shown in equation (7).

$$Q_{vu} + Q_f = c\rho \cdot V \cdot (T_{ufout} - T_r) + Q_f \quad (7)$$
$$= (c\rho \cdot Vol_r \cdot \Delta T_r) + Q_{wall} + Q_{ceil} + Q_{win} + Q_v$$

If it is assumed that there is no heat loss in the falling duct and handling box, the outlet temperature of collector is same to duct temperature. As shown in equation (8), the amount of collection in the room temperature criteria is same to the sum of heat supply to the room (Q_{vu}) of equation (7) and heat supply to the under floor (Q_{vd}) of the equation (6).

$$Q_{vu} + Q_{vd} = c\rho \cdot V \cdot (T_{duct} - T_r) = Q_r \quad (8)$$

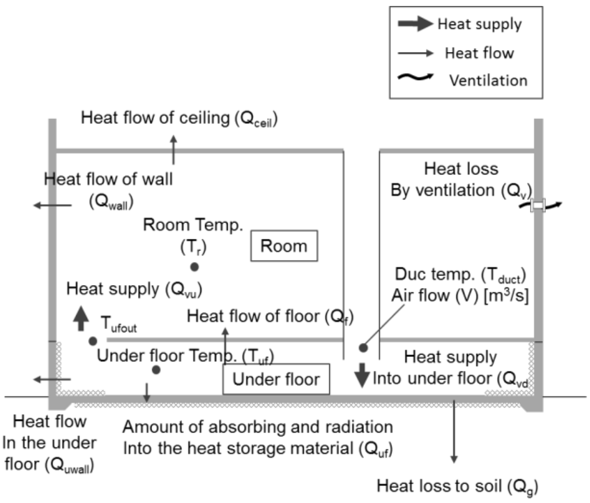


Figure 2: Definition of the heat flow

It is important to understand the heat balance in the building with this solar system in order to think about the improvement of the system. The heat balance diagram is shown in the Figure 3.

2.2 The points of improvement

There are four points for improvement as follows,
1) Increase the amount of solar heat collection.
2) Decrease the amount of heat loss from the basis concrete as the heat mass to the outdoor air and soil.

3) Increase the amount of heat absorption into the heat storage material.
4) Increase the amount of heat emission from the heat storage material.

In this study, as shown in Figure 4, (1) to improve the heat collection efficiency by improving the performance of collector and install the PV for electronic generation is target. The PV panel has same collection performance with existing pre-heat collector. And, (2) the method of insulation is considered for reducing heat loss to the outdoor and soil. The effective method of insulation is examined by measurement. (3) For increasing the heat absorbing and radiation of heat storage material by larger heat capacity, the water pack is installed as the additional heat storage material. (4) For increasing the heat radiation of heat storage material at the non-collection time, the forced air flow is adapted for increasing the convective heat transfer by changing the air of room and under floor. In the summer and intermediate season, the heated air is used for how water by heat exchange in the handling box.

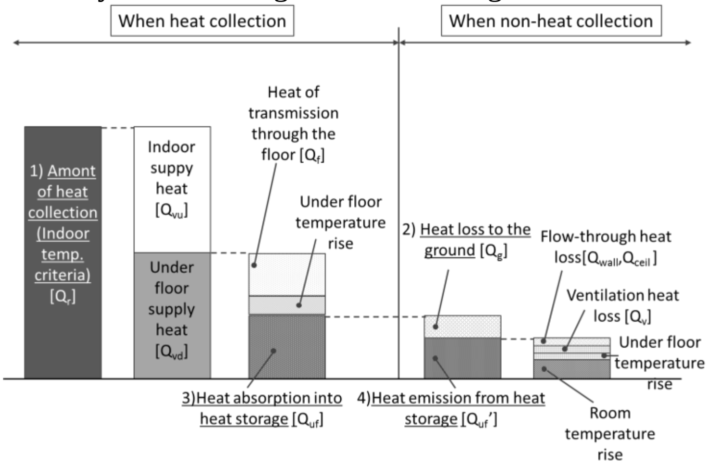


Figure 3: Heat balance of air-based solar system

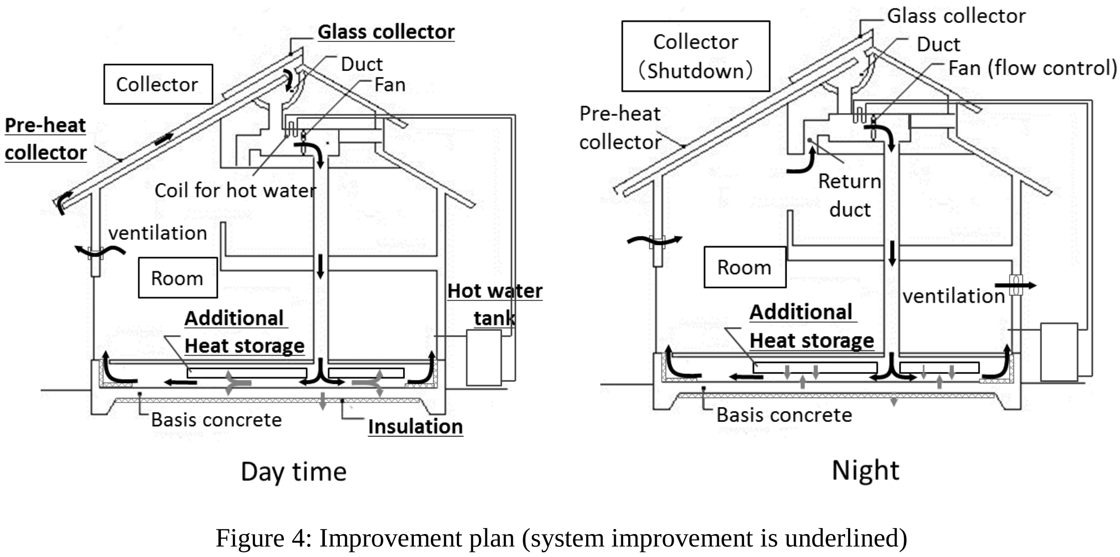


Figure 4: Improvement plan (system improvement is underlined)

3 SPECIFICATION OF HUTS AND MEASUREMENT

We built three huts in Hamamatsu City in Japan, of which size is same. These huts are equipped with well insulation which fills the energy saving standards in Japan. The specification is shown Table 1. Hut no.1 with the system and air conditioner has no insulation under the basis concrete. Hut no.2 with the system and air conditioner has insulation under the basis concrete. Hut no.3 doesn't have the solar heat system and equip only air conditioner. For example, we can compare the previous system with the improved system at the same time.

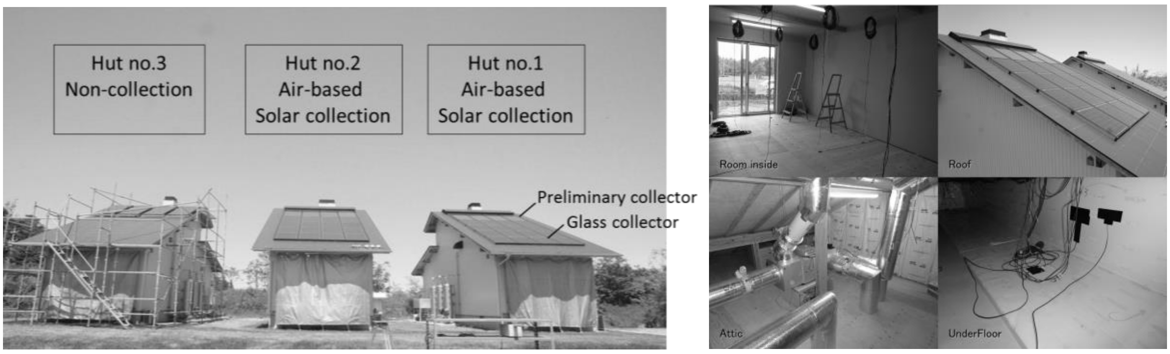
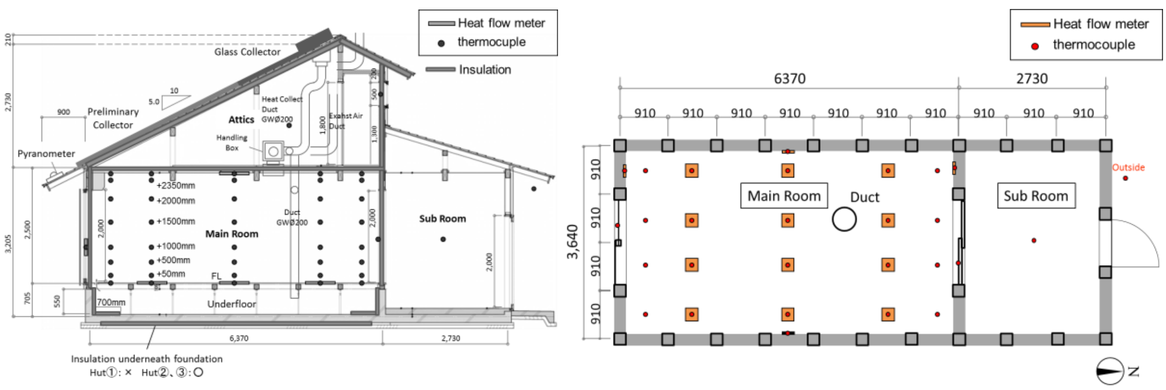


Figure 5: Overview of experiment huts

The measurement was set to analyse the heat balance in the whole huts. Especially, many thermocouples, heat flow sensors, mass flow maters and so on were set in the solar collector, duct space and under floor. The data was recorded every minute. We confirmed the measurement is very precise and grasp the heat balance in the real situation. Fig.6 and Fig.7 shows measurement point of building, collector, and under floor.

Table 1: Specification of experiment huts

Basis concrete	Walls: Width 150 * Height 600mm
	Plane: Thickness180*Width3, 790*Length6, 520mm
Walls (U-value：0.335W/m^2K)	Color steel plate, PB t=9.5mm, vent layer t=36mm, waterproof sheet, structural plywood t=9mm, High performanceGW16K 100mm, PB t=12.5mm
Floor	Structural plywood 28mm (thermal conductivity：0.13W/mK)
Ceiling (U-value：0.361W/m^2K)	Structural plywood t=28mm, phenolic foam t=80mm, Air layer 40mm, PB t=9.5mm
Roof (U-value：0.627W/m^2K)	Resin-based roofing t=0.42mm, Structural plywood t=12mm, GW32K t=50mm
South and North windows	Low-E(insulation)glass(3-A12-3, U-value：1.7W/m^2K)+aluminum multilayer・resin composite frame (Solar radiation through south window is shielded)
Insulation under basis concrete	extrusion method XPS 3-B t=50mm (hut no.2,3 only) (Thermal conductivity：0.03W/mK)
Insulation on basis concrete	Walls: extrusion method XPS 3-B t=50mm
	Outer periphery：extrusion method XPS 3-B t=50mm(Width：700mm)
Partition door between target and store room	Plywood t=3mm, GW32K t=25mm, Plywood t=3mm



(a) Section

(b) Plan

Figure 6: Measurement point in the building

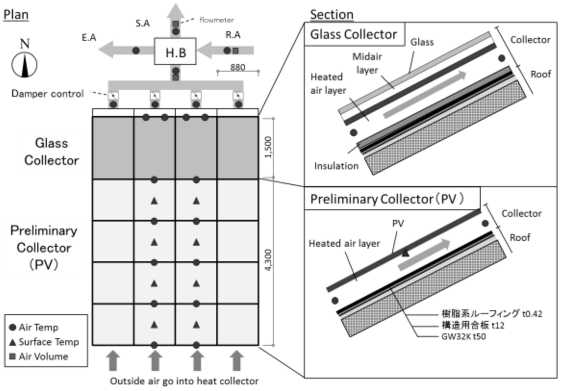


Figure 7: Measurement point of the collector

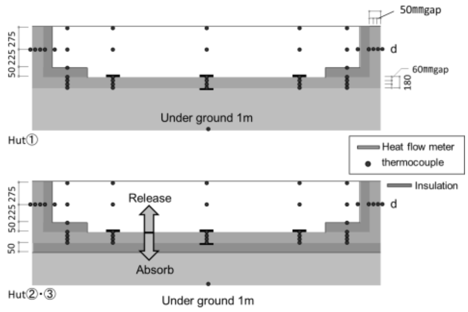


Figure 8: Measurement point of the under floor

4 EXPERIMENT OF HEATING IN WINTER

The purpose of these experiments is to analyse the efficiency of the improve methods. We judged the efficiency by the percentage of the amount of heat emission to the amount of heat absorption. The more the percentage (emission per absorption) is, it is consumed that the heating load will be reduced more.

Experiment 1): Confirmation of the amount of absorption and emission through the basis concrete between hut no.1 and no.2 in case of small solar collector
Experiment 2): Confirmation of the amount of absorption and emission through the basis concrete between hut no.1 and no.2 in case of large solar collector
Experiment 3): Confirmation of the amount of absorption and emission through the basis concrete and the additional heat storage, which was consisted of water in 40 packs (25 Litter per pack), between hut no.1 and no.2 in case of large solar collector
Experiment 4): Confirmation of the amount of absorption and emission through the additional heat storage. In hut no.1, there was water in 2000 PET bottles (0.5Litter per bottle). In hut no.2, there were water in 40 packs same as Experiment 3.

Table 2: Experiment conditions

	Heat Collection Area (Preliminary, Glass) [m^2]	Heat Storage	Experiment Period
Exp 1	7.0, 2.7	Basis concrete	29th Dec. 2012~6th Jan. 2013
Exp 2	14.0, 5.3	Basis concrete	7th Jan. 2013~14th Jan. 2013
Exp 3	14.0, 5.3	Basis concrete +water packs	29th Dec. 2012~6th Jan. 2013 (Hut1,2: Pre Experiment), 26th Dec. 2013~31st Mar. 2014(Hut2)
Exp 4	14.0, 5.3	Basis concrete +PET bottles	26th Dec. 2013~31st Mar. 2014

Insulation installation & Air conditioning

Experiment1, 2, 3:
Hut no.1-None, Hut no.2-possession under basis concrete
Air Conditioning from 6:00 to 23:00 (Set point = 20℃)
Auto-control Solar Collection（Handling Unit was stopped during night-time）

Experiment4:
Hut no.1- possession on basis concrete
Hut no.2- possession under and on basis concrete
Air Conditioning from 6:00 to 23:00 (Set point = 18℃)
Auto-control Solar Collection（Handling Unit was run during night-time）

4.1 Test Results

Experiment 1
If the solar collector doesn't get enough solar heat energy, the amount of absorption into the basis concrete is not sufficient, so the difference of the emission of heat between hut 1 and hut 2 was not appeared.
Clear days continued during the period. Collected Heat in hut 2 was about 10% less than in hut 1, Efficiency is about 15~20% (Figure 9).The reason of this difference may be caused by construction precision, the air-flow balance in the solar collector, and so on.
In hut 1, energy consumption for heating was over 1kW in the morning. It took a few hours to rise indoor temperature to 20 degree C. On the center part of concrete slab, peak value in absorption was about 200W, in emission was about 170W. On the peripheral part of slab, heat emission was quite small (Figure 10).
Electricity consumption in hut2 was less than hut1, in spite of less collected heat. Underneath insulation of hut 2 gave good effect, but heat absorption and emission is small in both hut because of small collectors (Figure 11).

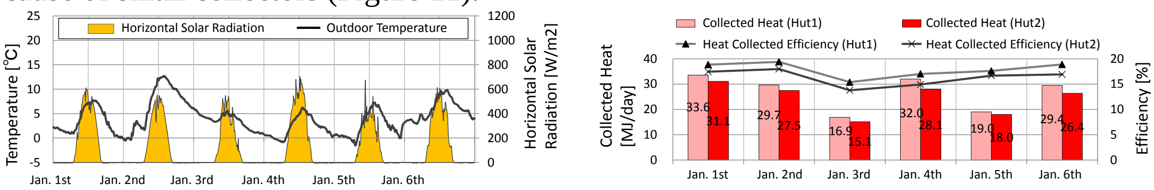


Figure 9: Weather & Collected Heat (Exp1)

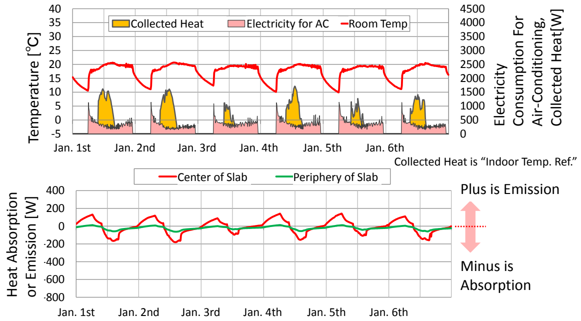


Figure 10: Result in hut 1 (Exp1)

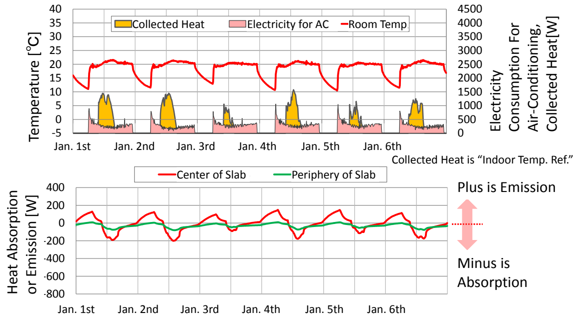


Figure 11: Result in hut 2 (Exp1)

Experiment 2

Collected Heat in hut 2 was about 15% less than in hut 1. Efficiency is about 17% in hut 1 and about 15% in hut 2 (Figure 12).

If the solar collector can get enough solar heat energy, the difference of the heat emission between hut 1 and hut 2 was appeared obviously. Although the amount of heat absorption into the basis concrete of hut 2 was less than hut 1, the amount of the heat emission of hut 2 was bigger than hut 1.

In hut 1, energy consumption for heating was over 1kW in the morning. It took a few hours to rise indoor temperature to 20 degree C. On the center part of concrete slab, peak value in absorption was about 600-750W, in emission was about 100W. On the peripheral part of slab, heat absorption was about 100W and emission was small (Figure 13).

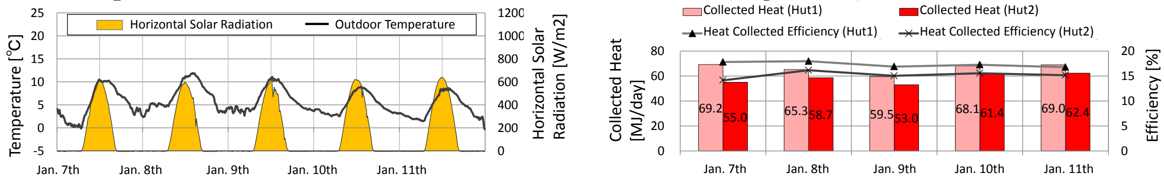


Figure 12: Weather & Collected Heat (Exp2)

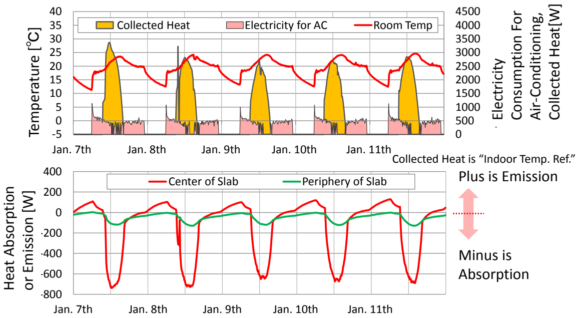


Figure 13: Result in hut 1 (Exp2)

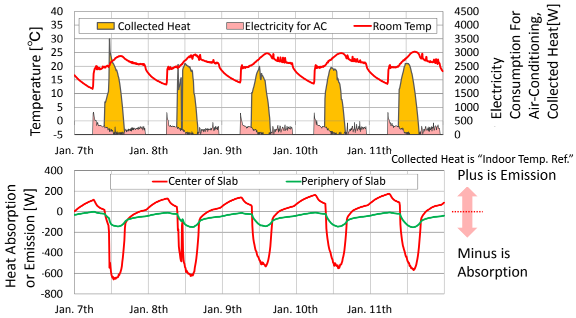


Figure 14: Result in hut 2 (Exp2)

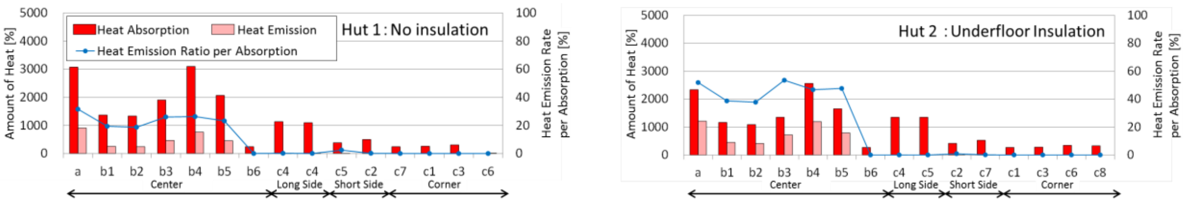


Figure 15: Heat emission per absorption of each heat flus sensors during the measurement period

Electricity consumption in hut2 was less than hut1 and indoor temperature rose in a short time. On the center part of concrete slab, peak value in absorption was about 500-600W, in emission was about 150W. On the peripheral part of slab, heat absorption was about 150W and emission was quite small (Figure 14).

In Hut2, absorption decreased but emission increased in comparison with Hut1. Heat emission ratio per absorption rose to about 50% in center part of slab (Figure 15). The percentage of total amount of emission per absorption was 17.6% in hut 1 and 29.4% in hut 2 during the period.

Experiment 3

Figure 16 shows the composition of water packs. In the case of setting water in packs in the underfloor space, the heat which water packs absorbed into wasn't extinguished as thermal loss into outdoor or soil. The amount of heat emission increased in night time. The percentage (emission per absorption) was 55.7% in hut 2.

Experiment 4

In the case of setting water in PET bottles in the underfloor space (Figure 18), the heat which PET bottle absorbed wasn't extinguished same as Experiment 3. Because the surface area of PET bottle is much larger than one of the water pack, the heat transfer was accelerated and the amount of heat emission highly increased in night time (Figure 18). The percentage (emission per absorption) was 47.0% in hut 1 and 25.7% in hut 2.

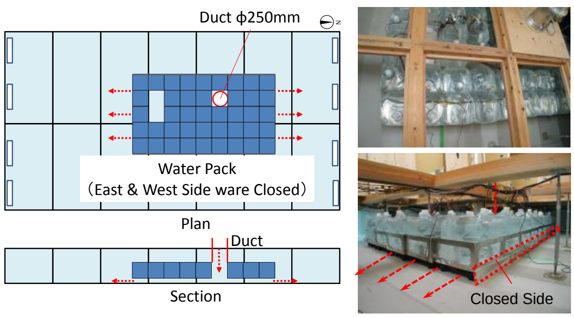


Figure 16: Composition of water packs in hut 2

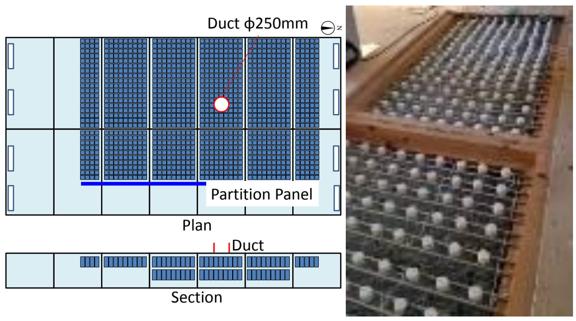


Figure 17: Composition of water PET bottles in hut 1

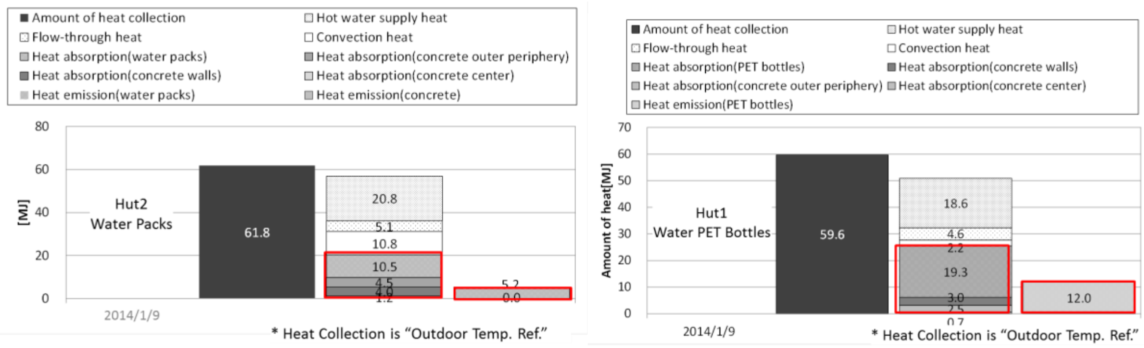


Figure 18: Heat balance in the underfloor in 1st Jan. 2014 (Experiment 3 & 4)

The amount of collected heat (outdoor temperature reference) in hut 1 and 2 were almost same. As shown in figure 19, Temperature of underfloor and floor in hut 2 was relatively higher than the temperature in hut 1, but room temperature of hut 1 was higher than hut2. As a result, the energy consumption for air-conditioning in hut 1 was lower than of hut 2. We confirm the result was caused by the fact that larger heat convection area of heat storage was preserved in hut 1 than hut 2, in spite of same volume of additional thermal storage.

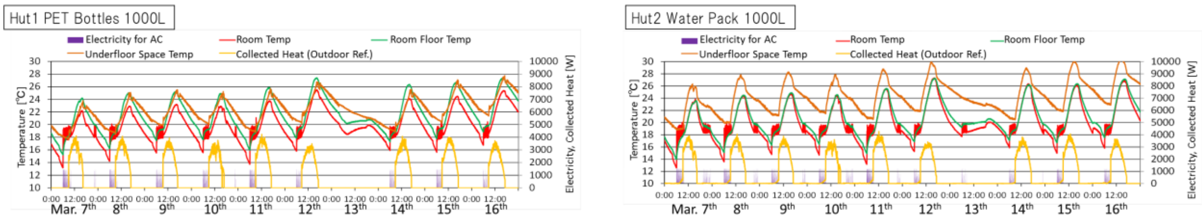


Figure 19: Temperature, collected heat & electricity used by air-conditioning (Experiment 3 & 4)

5 CONCLUSION

We confirmed the improve method of heat storage of the air-based solar system.

Experiment1 & 2:
There must be sufficient solar collector on the roof.
The insulation under the basement slab should be needed for night-time heat emission.
Reinforcement of insulation for the reduction of heat loss from peripheral part of basement should be needed.

Experiment 3 & 4:
Additional thermal mass is effective to increase heat emission in night time.
Especially, water in many PET bottles as thermal mass is effective to improve heat absorption and emission.

But the performance of the improvement system is not sure in the experiment. We'll show the amount of heating and hot-water load reduction by using simulation in the next study

6 ACKNOWLEDGMENT

This study is carried into execution by "NEDO solar energy utilizing type housing technology development grant project, 2011~2013" (OM Solar Inc., The University of Tokyo, Kogakuin University).

7 REFERENCES

Youngjin Choi et al.(2014), Evaluation of the characteristic of heat flow under the floor in the test buildings in winter. Journal of environmental engineering (Transactions of AIJ), 79, 697, 271-280.

Zhejun XIAN et al(2003), An experimental study on the solar floor heating system with water heat-storage floor, Journal of environmental engineering (Transactions of AIJ), 572, 39-45.

Hayashi(1999), Guidelines of Plan of Solar House with Solar Air Collector and Floor Heating System, Architectural Institute of Japan

IMPLEMENTATION OF MULTI-ZONE VENTILATION METHODOLOGY IN THE SPANISH ENERGY PERFORMANCE CERTIFICATION TOOL

Javier García Ramos[1], José Manuel Salmerón Lissén[*1], Laura Romero Rodríguez[1], Servando Álvarez Domínguez[1], José Luis Molina Félix[1], Álvaro Ruiz Pardo[2]

1 Grupo de Termotecnia. Seville University
Camino de los descubrimientos S/N
Escuela Técnica Superior de Ingenieros
Seville 41092, Spain
**Corresponding author: jms@us.es*

2 Cádiz University
C/Chile 11002
Cádiz, Spain

ABSTRACT

The purpose of this paper is to enhance the importance of ventilation regarding energy use and stablishing methods in order to obtain as much data as possible about the behavior patterns of ventilation and infiltration in buildings.
The current Spanish regulation Royal Decree 235/2013 establishes all new-construction, rented or sold buildings must be in possession of the Energy Performance Certificate. There are several examples of dedicated software to calculate it, and the aspiration of this study is to integrate the influence of ventilation in the official Energy Performance Certificate Software developed by the Department of Energy located in the University of Seville. This new feature includes Multi-zone ventilation in pressures and Multi-zone ventilation in air flows. In addition, it allows the user to consider the effect on the demand of energy of the real pressures and air flows given in the different zones compared to the single-zone model for ventilation integrated in the official tool.
The structure of this article is summarized as follows:

1. Introduction.
2. Air flow fundamentals, balance equations, ventilation and infiltration models and CO_2 concentration.
3. Description of the “VENTItool” software to simulate ventilation in dwellings using a multizone model.
4. Validation of results using CONTAM.
5. Results and description of the additional capacity about multi-zone ventilation.
6. Conclusions.

KEYWORDS

Energy performance certificate, multizone ventilation model, ventilation, infiltration, air change rate n50.

1. INTRODUCTION

Ventilation plays a vital role regarding thermal comfort and air quality, meeting the metabolic needs of occupants and diluting and removing pollutants emitted by indoor sources. However, unnecessarily high rates of air change can present an excessive energy burden on a building’s heating or cooling needs. It is estimated that ventilation accounts for at least 30% of space conditioning energy demand. For that reason, there is a conflict between a willingness to minimize ventilation rate, reduce energy demand, and maximize ventilation to ensure optimum indoor air quality [1].

The task of anticipating ventilation needs is an intricate one, since they vary depending on the occupant density, climate and pollutant load. It is decisive to understand its purpose and how it interacts with pollutants and energy performance.

For all these reasons, in recent years new ventilation strategies have been applied and developed, with the aim of diminishing the energy demand and making use of the outside air cooling potential in certain conditions.

The current Spanish regulation Royal Decree 235/2013 establishes all new-construction, rented or sold buildings must be in possession of the Energy Performance Certificate. The Ministry of Industry, Energy and Tourism has created a General Register which contains the official documents recognized for the energy performance certification process.

Among these documents an official software developed by the Department of Energy located in the University of Seville can be found, used by thousands of certifiers to perform the Energy Performance Certificate. This Energy Performance Certificate tool allows the user to define a residential or tertiary building with a high precision making use of specific additional capacities.

There are many software tools that are available for ventilation calculations. One example is CONTAM [2], which is a multizone airflow and contaminant transport analysis software for buildings. Another one is COMIS [3], a FORTRAN-based code which models the air flow and contaminant distributions in buildings. However, despite the availability of many calculation options the aspiration of this study was to integrate the influence of ventilation in the official Energy Performance Certificate Software. As a consequence, it was decided to develop our own tool, in order to achieve an optimal coupling.

Thus, one of the four additional capacities found in the new version of the Energy Performance Certificate Software (apart from "Trombe" Wall, Ventilated Façade and Solar Wall) is Multi-zone Ventilation. The aim of this paper is to describe the latter additional capacity about Multi-zone ventilation in pressures and Multi-zone ventilation in air flows. This feature allows the user to consider the effect on the demand of energy of the real pressures and air flows given in the different zones compared to the single-zone model for ventilation integrated in the official tool.

2. METHODOLOGY AND TECHNICAL BASICS

2.1. Air flow fundamentals

A wind flow produces a velocity and pressure field around a building. The free-flow relationship between velocity and pressure in each point of the field can be derived from the Bernoulli equation:

$$P + \frac{1}{2}\rho \cdot v^2 = cte \tag{1}$$

In order to describe the pressure distribution around the building envelope a dimensionless coefficient called "Pressure coefficient" (C_p) is usually used, which corresponds to the ratio between the dynamic pressure over the surface and the dynamic pressure of the undisturbed flow at the reference height. For a point of the surface k (x, y, z), the expression of the pressure coefficient comes from the following equation:

$$C_p\left(z_{ref}\right) = \frac{\left(p_k - p_0(z)\right)}{\frac{1}{2}\rho_0 \cdot v^2(z_{ref})} \tag{2}$$

Consequently, the strategy used to evaluate the C_p is through numerical methods. The C_p values are tabulated depending on different parameters (surroundings of the building, wind direction) and have been obtained through the interpolation of the results in several trials.

2.2. Balance equations

For each considered zone the incoming and outgoing air flows will be accounted. In addition, the difference should be equal to zero. The balance equation for each zone is the following:

$$\sum_{i=No.façade} q_{walls,i} + \sum_{i=No.façade} q_{windows,i} + \sum_{i=No.façade} q_{outgrilles,i} + q_{ingrilles} = 0 \qquad (3)$$

Where:

- $\sum_{i=No.façade} q_{walls,i}$ is the sum of air flows through walls due to leakages in m^3/h.
- $\sum_{i=No.façade} q_{windows,i}$ is the sum of air flows through windows due to leakages in m^3/h.
- $\sum_{i=No.façade} q_{outgrilles,i}$ is the sum of air flows through outside grilles in m^3/h.
- $q_{ingrilles}$ is the flow which goes from the corridor through the inside grilles.

2.3. Ventilation and infiltration models

2.3.1. Air flow through gaps

To model all the different effects that can occur an exponential law is commonly used:

$$Q = C_s \cdot f(\rho, v, n) \cdot (\Delta p)^n \qquad (4)$$

This exponential law will be different for each type of crack considered, adjusting the parameters experimentally. Finally, the expression remains as follows:

$$Q = C \cdot \Delta p^n \qquad (5)$$

Where “C” and “n” are obtained from experimental results.

2.3.2. Air flow through walls caused by defects

There exists a “wall leakage coefficient” which represents the value of the permeability of walls, which depends on their composition.
Since the pressure difference doesn’t necessarily have to be the same as that of the trial, it is recommendable to make use of the following expression:

$$q_{walls} = C_{walls} \cdot S_{walls} \cdot \left(\frac{|\Delta P_{real}|}{\Delta P_{trial}^{walls}}\right)^n \qquad (6)$$

Where:

- q_{walls} is the flow through the walls in m^3/h.
- C_{walls} is the wall leakage coefficient in $m^3/h{\cdot}m^2$.
- S_{walls} is the walls surface in m^2.
- ΔP_{real} is the pressure difference between the inside and outside of the considered zone in Pa.
- ΔP_{trial}^{walls} is the reference pressures difference used in the trial in Pa.
- n is the leakage exponent.

The UNE-EN 15242:2007 standard [4] recommends an n exponent value of 0.667.

2.3.3. Air flow through windows caused by defects

On the other hand, there is also a “windows leakage coefficient” which indicates their permeability and depends on their quality. The expression used is the subsequent:

$$q_{windows} = C_{windows} \cdot S_{windows} \cdot \left(\frac{|\Delta P_{real}|}{\Delta P_{trial}^{windows}}\right)^{n} \quad (7)$$

The UNE-EN 15242:2007 standard [4] also recommends in this case an n exponent value of 0.667.

2.3.4. Air flow through inside and outside grilles

Our study considers the implementation of three different types of grilles whose modeling is going to be outlined.

2.3.4.1. Conventional grilles

The functioning of this type is governed by the following equation:

$$Q = c \cdot \left(\frac{\Delta p}{20}\right)^{0.5} \quad (8)$$

Where:

- Q is the flow through the grille in m^3/h.
- c is the flow rate in m^3/h when the pressure difference is 20 Pa. Its value determines the size or the cross section of the grille. The bigger the c value, the bigger the grille.
- Δp is the pressure difference between both sides of the grille in Pa.

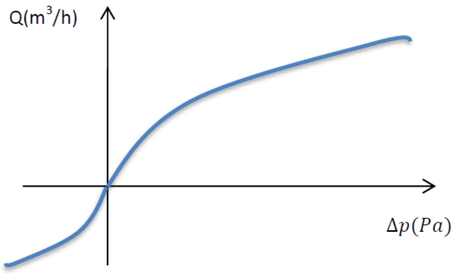


Figure 1: Illustration of the conventional grille’s functioning.

2.3.4.2. Self-regulating grilles

This type includes a component which automatically closes when the outside wind velocity increases, reducing the inlet air and obtaining a more uniform air flow.
The equation used is the following, obtained from the SIREN2000 version 8 manual:

$$Q = \begin{cases} c \cdot \left(\frac{\Delta p}{20}\right)^{0.5} & \Delta p < 20 \\ c + b_1 \cdot (\Delta p - 20) & \Delta p \geq 20 \end{cases} \quad (9)$$

Where:

- b_1 is the coefficient which determines the steepness of the line when $\Delta p \geq 20$ Pa, in $m^3/h/Pa$.

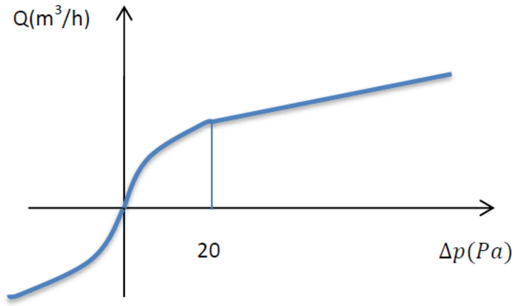


Figure 2: Illustration of the self-regulating grille's functioning.

As is apparent, a linear behavior can be seen when the pressure is over 20 Pa. When smaller, the behavior is similar to that of the conventional grilles.

2.3.4.3. Non-return grilles

They are designed to reduce the air flow going from the inside out.
The equation used is the following, obtained from the SIREN200 version 8 manual:

$$Q = \begin{cases} c \cdot \left(\frac{\Delta p}{20}\right)^{0.5} & 0 < \Delta p < 20 \\ c + b_1 \cdot (\Delta p - 20) & \Delta p \geq 20 \\ -\frac{c}{2} \cdot \left(\frac{\Delta p}{20}\right)^{0.5} & -20 < \Delta p \leq 0 \\ -\frac{c}{2} - b_2 \cdot (\Delta p - 20) & \Delta p \leq -20 \end{cases} \tag{10}$$

Where:

- b_1 is the coefficient which determines the steepness when $\Delta p \geq 20$ Pa in m^3/h/Pa.
- b_2 is the coefficient which determines the steepness when $\Delta p \leq -20$ Pa in m^3/h/Pa.

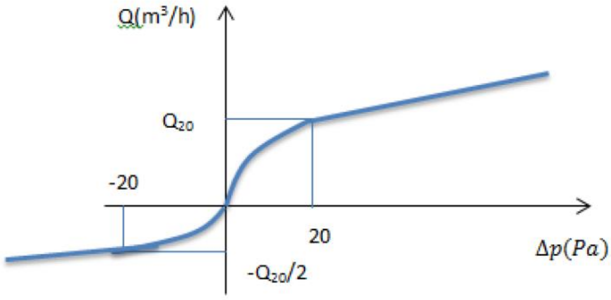


Figure 3: Illustration of the non-return grille's functioning.

As it can be observed, a linear behavior can be seen when the pressure is over 20 Pa. For values between 0 and 20 Pa the behavior is that of the conventional grills. When the pressure difference is negative, the air flow is reduced to a half, and the behavior identical to that of a conventional grille.

2.3.5. Newton-Raphson method

With a view to understanding how our software works, it is necessary to introduce the mathematical fundamentals of the Newton-Raphson method for non-linear equation systems, which is indispensable to solve the equations of the ventilation model.
The recurrence relation of the Newton-Raphson method for non-linear equation systems is:

$$x_{k+1} = x_k - J(x_k)^{-1} f(x_k) \tag{11}$$

Since we are not interested in knowing nor calculating the analytical expression of the Jacobian matrix, this is replaced by its finite difference approximation:

$$\frac{\partial f_j(x)}{\partial x_k} \approx \frac{f(x_k+h_k e_j)-f(x_k)}{h_k} \qquad (12)$$

The following diagram shows the used Newton-Raphson algorithm to solve the equation system:

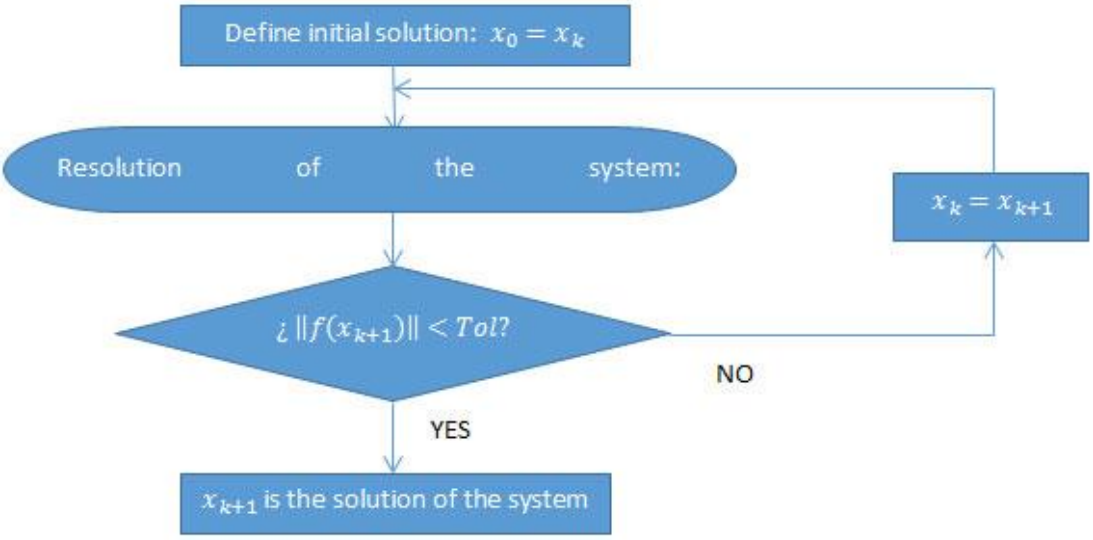


Figure 4: Flowchart of the Newton-Raphson method

2.4. CO_2 concentration

To calculate CO_2 concentration it is necessary to lay out a mass balance in each room. In order to simplify the model the concentration in the zone is supposed to be homogenous.

$$V \cdot \left(C_{int}(t+\Delta t) - C_{int}(t)\right) = \sum_i q_i \cdot C_i(t) \cdot \Delta t + G \cdot \Delta t \qquad (13)$$

Where:

- V is the volume of the zone in m^3.
- C is the CO_2 concentration in mg/m^3.
- Δt is the time increment chosen.
- q is the airflow path that goes in or out the zone in m^3/h.
- G is the CO_2 generated by occupants.

3. SOFTWARE TO SIMULATE VENTILATION IN DWELLINGS USING A MULTIZONE MODEL CALLED VENTITOOL

The aim of this section is to introduce the program we designed to calculate CO_2 concentration and air flows for a given dwelling. Figure 5 shows the flux diagram you need to follow the use the program properly.

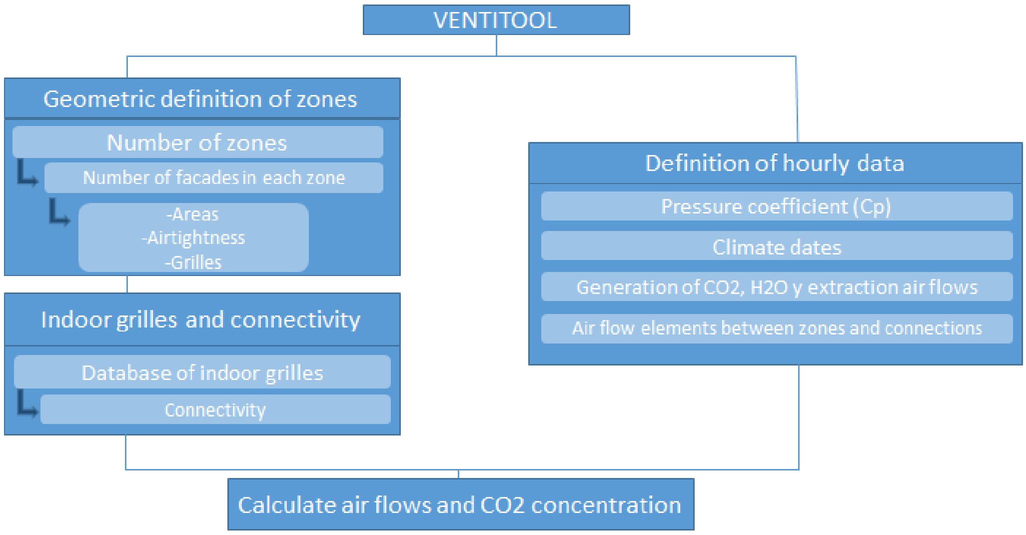


Figure 5: Flux diagram for VENTItool software

There are three steps we must follow before clicking the calculate button:

- Geometric definition of zones: Figure 6-right shows all data needed to define the dwelling, such as number of zones, number of façades in each zone, areas, permeability coefficients for windows and walls and definition of grilles.

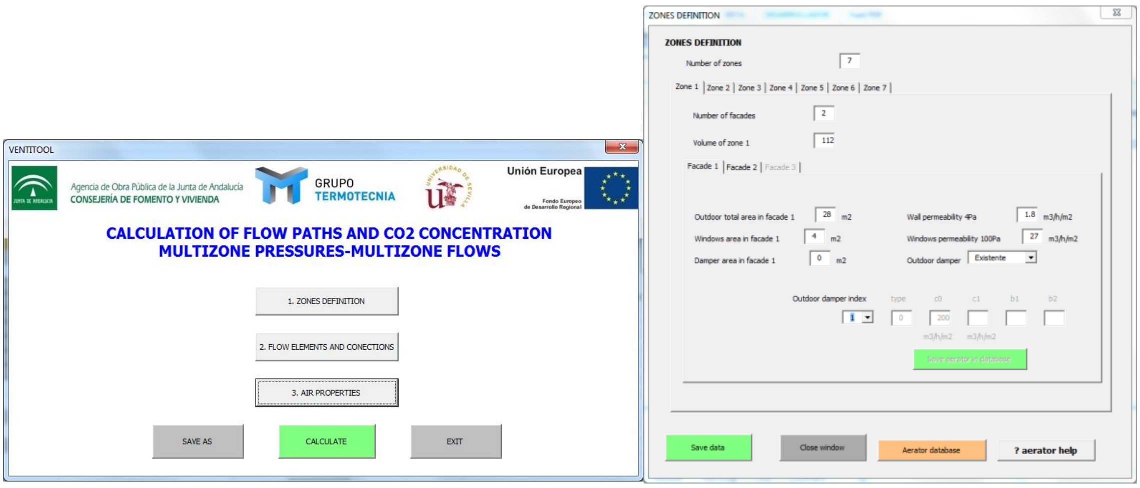


Figure 6: Software Ventitool developed by the Department of Energy in Seville University

- Indoor grilles and connectivity: One must define the kind of connectivity between zones. To do that, firstly you must indicate the type of grilles you will use and what type of grille connects each room. Figure 7-left shows the database of indoor grilles, and figure 7-right shows the connectivity matrix.

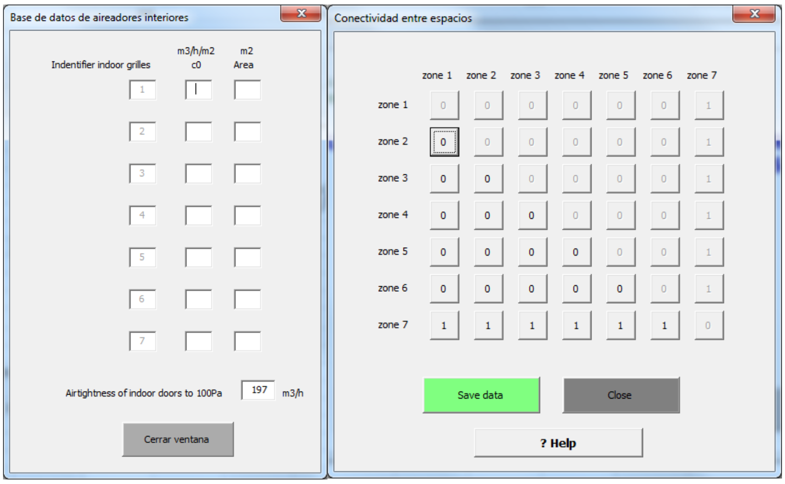
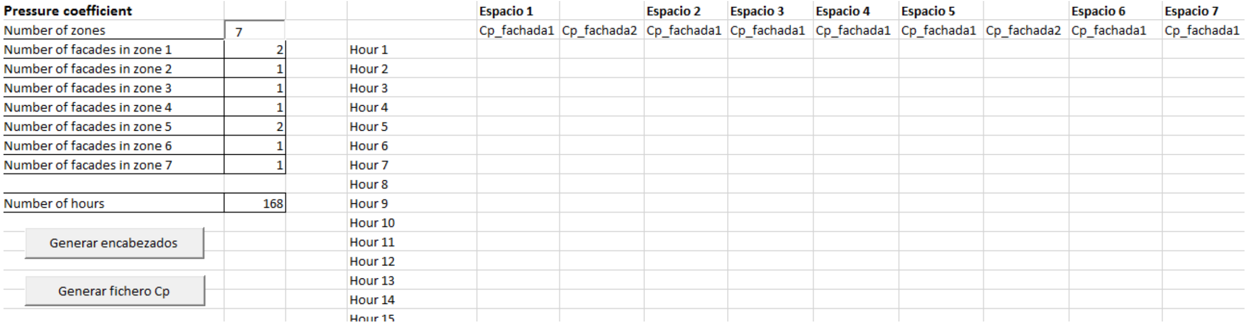


Figure 7: Database of indoor grilles (left) and connectivity matrix

- Definition of hourly data: To be able to do the simulation for a certain number of hours, some coefficients such as pressure coefficient, climate data, generation of CO_2, extraction air flows, air flow element between zones and connections can be defined per hour using text files.



Pressure coefficient			**Espacio 1**		**Espacio 2**	**Espacio 3**	**Espacio 4**	**Espacio 5**		**Espacio 6**	**Espacio 7**
Number of zones	7		Cp_fachada1	Cp_fachada2	Cp_fachada1	Cp_fachada1	Cp_fachada1	Cp_fachada1	Cp_fachada2	Cp_fachada1	Cp_fachada1
Number of facades in zone 1	2	Hour 1									
Number of facades in zone 2	1	Hour 2									
Number of facades in zone 3	1	Hour 3									
Number of facades in zone 4	1	Hour 4									
Number of facades in zone 5	2	Hour 5									
Number of facades in zone 6	1	Hour 6									
Number of facades in zone 7	1	Hour 7									
		Hour 8									
Number of hours	168	Hour 9									
		Hour 10									
Generar encabezados		Hour 11									
		Hour 12									
Generar fichero Cp		Hour 13									
		Hour 14									
		Hour 15									

Figure 8: Excel sheet that creates the hourly text files

Finally this tool obtains as result airflow paths for each hour and the CO2 concentration evaluated each minute. These results will be shown in subsequent sections.

4. VALIDATION OF RESULTS USING CONTAM

In order to check that our program is providing accurate results, we performed the validation of results with CONTAM. Figure 9 and table 1 show the geometric characteristics of the dwelling:

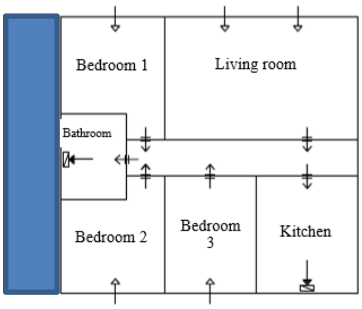


Figure 9: Building used to perform the simulations

The permeability coefficients for windows and walls are supposed to be 27 $m^3/m^2 \cdot h$ and 0 $m^3/m^2 \cdot h$ respectively. The air flow extractions in the kitchen and bathroom are 115 m^3/h and 54 m^3/h respectively.

Table 1: Feature of the building's construction

	Living room	**Bedroom 1**	**Bedroom 2**	**Bedroom 3**	**Kitchen**	**Bathroom**	**Corridor**
Number of facades	2	1	1	1	2	0	0
Volume (m^3)	86.4	43.2	43.2	43.2	43.2	30	34.8
Wall area (m^2)	26.9	9.3	9.3	9.3	20.6	0	0
Window area (m^2)	5.5	1.5	1.5	1.5	0	0	0

Table 2 shows the occupancy schedule used to validate the CO_2 concentration:

Table 2: Occupancy schedule for weekdays (left) and holidays (right)

Time	Livingroom	Bedroom 1	Bedroom 2/3	Kitchen	Bathroom
00:00	0	2	1	0	0
01:00	0	2	1	0	0
02:00	0	2	1	0	0
03:00	0	2	1	0	0
04:00	0	2	1	0	0
05:00	0	2	1	0	0
06:00	0	2	1	0	0
07:00	0	2	1	0	0
08:00	0	0	0	3	1
09:00	0	0	0	0	0
10:00	0	0	0	0	0
11:00	0	0	0	0	0
12:00	0	0	0	0	0
13:00	0	0	0	0	0
14:00	0	0	0	0	0
15:00	0	0	0	0	0
16:00	0	0	0	0	0
17:00	3	0	0	0	0
18:00	3	0	0	0	0
19:00	1	0	1	0	0
20:00	0	0	1	1	1
21:00	3	0	0	0	0
22:00	4	0	0	0	0
23:00	4	0	0	0	0

Time	Livingroom	Bedroom 1	Bedroom 2/3	Kitchen	Bathroom
00:00	0	2	1	1	0
01:00	0	2	1	1	0
02:00	0	2	1	1	0
03:00	0	2	1	1	0
04:00	0	2	1	1	0
05:00	0	2	1	1	0
06:00	0	2	1	1	0
07:00	0	2	1	1	0
08:00	0	0	0	0	3
09:00	4	0	0	0	0
10:00	0	0	0	0	0
11:00	0	0	0	0	0
12:00	4	0	0	0	0
13:00	2	0	0	0	2
14:00	4	0	0	0	0
15:00	4	0	0	0	0
16:00	4	0	0	0	0
17:00	4	0	0	0	0
18:00	4	0	0	0	0
19:00	0	0	0	0	0
20:00	0	0	0	0	0
21:00	1	0	0	0	2
22:00	4	0	0	0	0
23:00	4	0	0	0	0

The difference between intake air flows from the outside calculated with CONTAM and VENTItool are shown in the following table:

Table 3: Difference between intake air flows from the outside calculated with CONTAM and VENTItool

	Intake air flow (m3/h) VENTItool	**Intake air flow (m3/h) CONTAM**	**error**
Living room	85.67	85.69	2.33e-4
Bedroom 1	27.77	27.78	3.59e-4
Bedroom 2	27.77	27.78	3.59e-4
Bedroom 3	27.77	27.78	3.59e-4
Kitchen	0	0	0
Bathroom	0	0	0

The comparison between CO_2 concentration with CONTAM and VENTItool for two scenarios is shown below:

- **Scenario 1:** Figure 10-left shows the wind direction in the scenario 1. Figure 10-right (above) presents the CO2 concentration in the living room using VENTItool, whereas figure 10-right (below) shows the CO2 concentration in the living room using CONTAM.

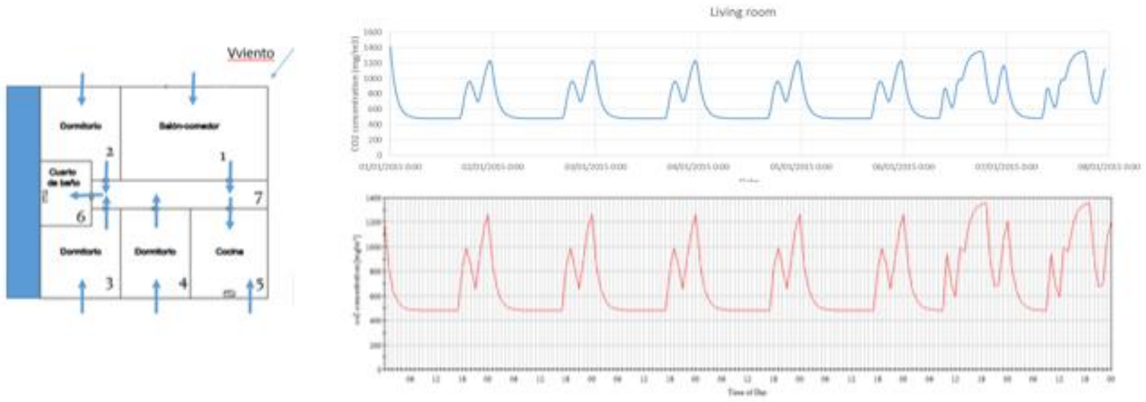


Figure 10: Wind direction in the scenario 1 (left) and CO2 concentration using VENTItool (right-above) and CONTAM (right-below)

Table 4 shows the mean and maximum value in the living room using both programs. The difference is due to the fact that CONTAM presents the hourly values of CO_2 concentration, while VENTItool shows these values each minute.

Table 4: Mean and maximum CO_2 concentration in the living room using CONTAM and VENTItool

CO_2 (mg/m3)	**Mean**	**Maximum**
CONTAM	696.62	1357.49
VENTItool	702.39	1355.31

- **Scenario 2:** Figure 11-left shows the wind direction in the scenario 2. Figure 11-right (above) presents the CO2 concentration in the bedroom1 using VENTItool, whereas Figure 11-right (below) shows the CO2 concentration in the bedroom1 using CONTAM.

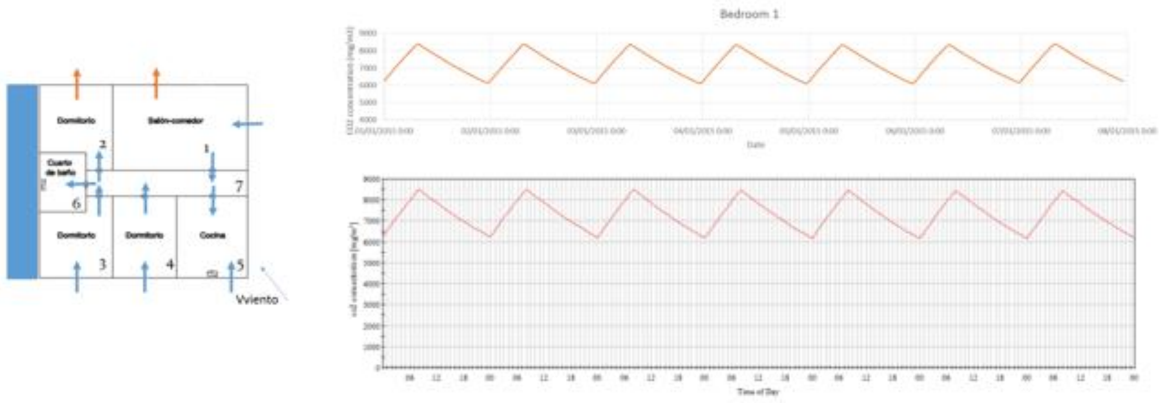


Figure 11: Wind direction in the scenario 2 (left) and CO_2 concentration using VENTItool (right-above) and CONTAM (right-below)

Table 5 shows the mean and maximum value in the living room using both programs.

Table 5: Mean and maximum CO_2 concentration in the bedroom 1 using CONTAM and VENTItool

CO_2 (mg/m3)	Mean	Maximum
CONTAM	7191.84	8354.32
VENTItool	7204.56	8351.27

According to the comparison between VENTItool and CONTAM, we can conclude that VENTItool provides an accurate solution of air flows and CO2 concentrations.

5. RESULTS

VENTItool is not only used as an additional capacity in the Energy Performance Certificate Software to know the real demand of energy for a multizone scenario, but also to analyse the CO_2 concentration, air flows exchanges between the outside and the zones, and even to calculate infiltrations. This is what will be explained in this section.

a. CO_2 concentration

One of the outputs of the program is the CO_2 concentration. For the scenario represented in figure 12-left, where wind speed is equal to zero and an occupancy schedule as in table 2, VENTItool provides the evolution of CO_2 concentration per minutes and per zone and a table with the maximum and average value per zone:

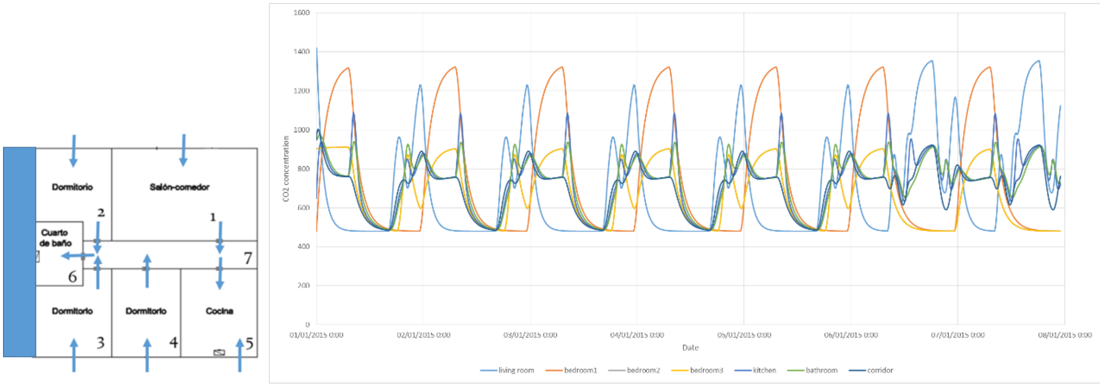


Figure 12: Scenario without wind speed (left) and evolution of CO_2 concentration (right)

Table 6: Mean and maximum CO2 concentration

	Living room	Bedroom1	Bedroom2	Bedroom3	Kitchen	Bathroom	Corridor
CO2 mean	702.39	767.26	669.71	669.71	736.19	733.32	702.75
CO2 maximum	1419.00	1323.21	911.34	911.34	1087.71	974.31	1003.57

As a consequence, it is possible to determine the hour when problems might exist due to a low ventilation rate and high occupancy schedule.

b. Air flows

Another output is the distribution of air flows in each room. For the scenario represented in figure 13-left, where wind speed is equal to 4m/s, VENTItool provides the air flows in any point of time (figure 13-right)

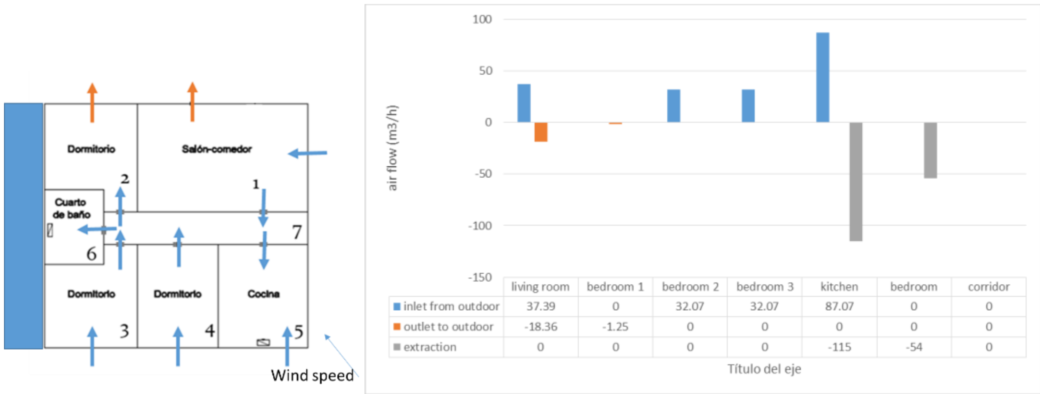


Figure 13: Scenario with wind speed equal to 4m/s (left) and air flows per room (right)

Thus, it is possible to determine at any time which room has problems with air ventilation. In this case note that bedroom 1 has no ventilation flow (a low air flow goes into the room coming from other zones). Figure 14 shows the CO_2 concentration for this scenario, where the concentration in bedroom 1 is too high due to the low air flow.

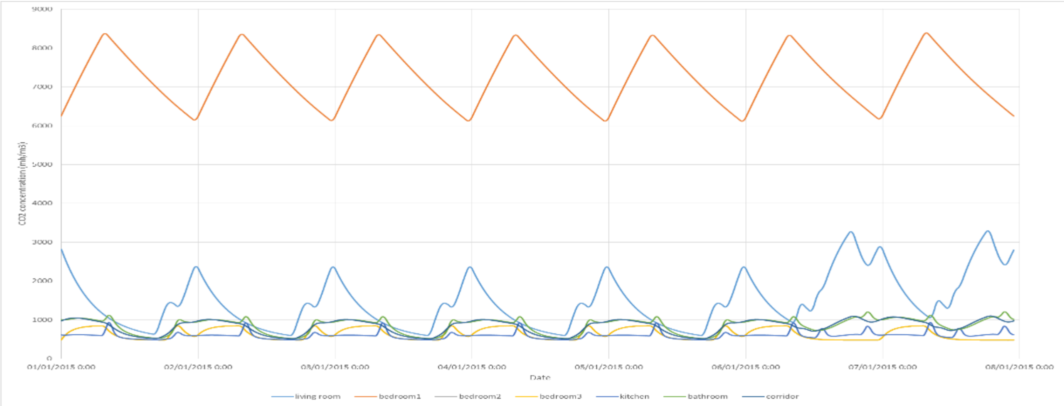


Figure 14: CO2 concentration

c. Relation between mechanical ventilation and infiltrations

VENTItool provides the relation between air change rate due to mechanical ventilation (ACH_{vent}) and the relation between air change rate due to ventilation and infiltration ($ACH_{vent+inf}$). Therefore, it can be determined for any n50 the value of ACH_{vent} required to avoid infiltrations. Figure 15-right shows this kind of graphs for the scenario presented in figure 15-left:

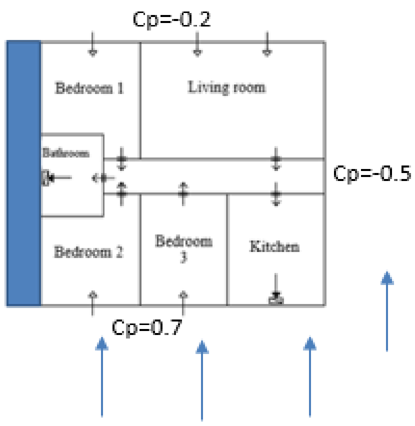


Figure 15: Scenario where 37% of facade is on windward (left) and relation between mechanical ventilation amd infiltration (right)

Note that the program provides the point in which you are working, so you can determine the minimal ACH_{vent} which avoids infiltrations for the airtightness defined by the users.

d. Demand of energy using VENTItool as additional capacity

VENTItool provides the air flows in each room, thus integrating these results into the official Energy Performance Certificate Software we can know the demand of energy for this multizone-scenario.

CONCLUSIONS

In this section the main conclusions are going to be highlighted:

- The Department of Energy at Seville University has developed a software called VENTItool that can estimate the CO_2 concentration for a multizone scenario, thus making it possible to detect for each room if there exists an acceptable IAQ.
- This software can determine the air flow ventilation rate in each room, so it is possible to know if all zones are receiving an acceptable amount of clean air from the exterior.
- VENTItool allows the users to know how many infiltrations there are in the building, and what should be done so as to avoid them.
- Finally, this tool can be integrated as an additional capacity in the Energy Performance Certification Software, therefore it is possible to know the real demand of energy due to ventilation and infiltrations.

REFERENCES

[1] Air Infiltration and Ventilation Centre. A guide to Energy Efficient Ventilation. (1996).
[2] Walton, G.N. and W.S. Dols. CONTAM User Guide and Program Documentation, 7251. National Institute of Standards and Technology: Gaithersburg (2010).
[3] Helmut E. Feustel. COMIS — An International Multizone Air-Flow and Contaminant Transport Model. Environmental Energy Technologies Division Lawrence Berkeley National Laboratory (1998).
[4] UNE EN 15242:2007. Ventilación de edificios. Métodos de cálculo para la determinación de las tasas de los caudales de aire en edificios, incluyendo la infiltración. CEN, 2007.

VENTILATION AND HEALTH – A REVIEW

Susanne Urlaub[*1], Gunnar Grün[2,3], Peter Foldbjerg[4], and Klaus Sedlbauer[2,5]

1 University of Stuttgart,
Chair of Building Physics
Pfaffenwaldring 7
Stuttgart,Germany
**susanne.urlaub@lbp.uni-stuttgart.de*

2 Fraunhofer Institute for Building Physics
Fraunhoferstrasse 10
Valley, Germany

3 Technische Hochschule Nürnberg,
Mechanical and Building Service Engineering
Keßlerplatz 12
Nürnberg, Germany

4 VELUX A/S
Ådalsvej 99
Hørsholm, Denmark

5 Technical University Munich,
Chair of Building Physics
Arcisstrasse 21
Munich, Germany

ABSTRACT

People in industrialised countries spend about 90% of their time indoors. Hence, a good indoor climate is essential for health and well-being. Ventilation of buildings plays an important role concerning health aspects of the occupants and inadequate ventilation may cause health costs that may have been avoidable if ventilation would have been adequate. Additionally, good or bad ventilation has impacts on the quality of the building, e.g. in very tight buildings, the risk of mould and dampness is higher if air change is insufficient. This paper focuses on the influence of different types of ventilation systems and ventilation rate on occupant's health in homes and schools. A literature search has been conducted to review the influence of ventilation on different health outcomes.

Most of the studies were found for the influence of natural vs. any type of mechanical ventilation on different health indicators. But the results of the different studies are not consistent. The reason for this may lie in the sometimes slightly but often considerably different ventilation variables that are evaluated. Studies report e.g. on the one hand that in naturally ventilated day care centres, the prevalence of asthma symptoms and rhinitis decreased whereas in another studies mechanical ventilation systems reduce eye and nasal irritation symptoms as well as tiredness. There are also studies that report no differences between the usage of natural vs. mechanical ventilation systems. One main weak point of many studies is that variables that may bias the results are not indicated, e.g. the state of the filters used. Two studies were designed as intervention study and they investigated the influence of heat recovery ventilators on asthma and respiratory disorders. Wheeze and rhinitis were found considerably lower in homes with heat recovery ventilators.

Low ventilation rates promote the development of allergies and respiratory diseases like wheeze or cough (OR between 1.3 and 2.39). In schools, ventilation rates are often below the recommended values (e.g. about 0.9 to 3 l/s instead of 8 l/s per person) and may therefore ease illness of pupils and teachers.

The following conclusions may be drawn from this work: First, a higher ventilation rate promotes health and attention should be drawn on the achievement of sufficient ventilation rates, especially in schools. Focused research should give a more detailed insight into this relationship. Second, the influence of the type of ventilation system is ambiguous with respect to the outcome. No clear recommendation for natural or mechanical ventilation could be derived from the available studies. More consistent data is needed with better documentation of confounding variables to consider causal relationships.

KEYWORDS

Ventilation rate, ventilation system, health, illness, review

1 INTRODUCTION

Buildings should protect people from harsh climatic conditions outside, e.g. very cold winters or very hot summers or rainfalls. Based on the materials used for construction and furnishing and the way how the rooms are used, a certain indoor climate occurs in the building. A very important part, how good or bad the indoor climate is, is the ventilation of the building. This includes as well the ventilation system itself, i.e. whether natural or mechanical ventilation or a combination of both is used, and additionally the ventilation rate, that is the amount of fresh air that is delivered inside the room. The indoor climate has a fundamental impact on the health of the occupants. It is known that the occurrence of visible mould spots or dampness may promote the development of asthma and allergies especially when exposure begins at a very young age (Quansah, 2012; Mendell, 2011).

While the research about mould in buildings is very popular, the influence of ventilation in all its different aspects (amount of fresh air, ventilation system) has reached much less attention, especially in residential buildings, schools and day care centres. Much work has been done for offices (e.g. Mendell, 1993, Seppänen & Fisk, 2002, Seppänen et al., 2006), here with an additional focus on performance, but the mere health aspects are seldom considered in a systematic way. The objective of this work is therefore to summarize the available knowledge and to identify future research needs.

2 METHOD AND RESULTS

A literature search was conducted to find the relevant scientific studies that focus on the influence of ventilation on health. The focus was set on homes and schools. To do this, databases, conference proceedings and journal articles were searched. Additionally, references from identified studies and reviews (Seppänen et al., 1999, Wargocki et al., 2002, Daisey et al., 2003, Seppänen & Fisk, 2004, Sundell et al., 2011) were searched by hand for further data. In total, 28 studies could be identified and were analysed in the next sections.

Based on the analysed studies, the following differentiation has been done. Some studies focus on the influence of different ventilation systems on different health outcomes whereas other research looks at the influence of the ventilation rate on health independent of the ventilation system. Other studies (e.g. intervention studies) could not be integrated in this scheme, therefore they are analysed separately.

2.1 The influence of the ventilation system on health

In this part, 13 studies were identified. Mainly, they focus on the influence of natural ventilation (NV) vs. mechanical exhaust ventilation (ME) vs. balanced ventilation (BV), i.e. mechanical supply and exhaust systems. The main characteristics of the studies are listed in Table 1. Inclusion and interpretation criteria were that data for ventilation system and health outcomes are analysed against each other and conclusions about the type of ventilation system and health outcome are drawn.

Table 1: Overview about the studies with focus on the ventilation system and health in homes and schools

Study	Ventilation system	Outcome	Results
Bogers et al., 2011 (H)	ME vs. BV	Perception of IAQ, subjective health	Perception of IAQ better with ME, no differences on subjective health
Bornehag et al., 2005a (H)	NV vs. ME vs. BV	Asthma, wheeze, cough, rhinitis, eczema	Eczema and rhinitis sig. higher with ME and BV compared to NV
Clausen et al., 2011 (H)	NV vs. ME, NV vs. BV	Child absenteeism in day care centres	No influence of ventilation system
Ebbehoj et al., 2005 (S)	Mechanical ventilation vs. not	General, mucous membrane and skin symptoms	Male teachers tended to have more throat irritation symptoms in mechanically ventilated schools
Emenius et al., 2004 (H)	NV vs. ME, NV vs. BV	Wheeze	No effect when OR are adjusted with different confounders
Engvall et al., 2003 (H)	NV vs. ME vs. BV vs. mixed system	Irritation symptoms, headache, tiredness	Irritation symptoms lower with ME and BV, headache and tiredness not clear
Engvall et al., 2005 (H)	Constant vs. season-ally adapted mecha-nical ventilation	Irritation symptoms, headache, tiredness	No difference between the two ventilation systems
Norbäck et al., 1995 (H)	NV vs. ME vs. BV	Asthma	No influence of ventilation system
Øie et al., 1999 (H)	NV vs. ME vs. BV	Bronchial obstruction	No influence of ventilation system
Ruotsalainen et al., 1991 (H)	NV vs. ME vs. BV	Irritation symptoms, cough, headache…	Lethargy, weakness, nausea sig. more often reported in NV houses
Smedje et al., 2011 (S)	Mixing vs. displace-ment ventilation	Perceived IAQ, clini-cal health parameters	No differences in perceived IAQ and clinical parameters, more eye symptoms with displacement system
Wålinder et al., 1998 (S)	NV vs. ME vs. BV vs. supply only vs. displacement	Nasal symptoms , biomarkers	Nasal symptoms lower with NV, nasal obstruction sig. higher with BV compared to any other type
Zuraimi et al., 2007	NV vs. air-conditioned vs. air-conditioned +BV vs. hybrid	Asthma, allergies, respiratory symptoms	Lower prevalence for most asthma, allergy and respiratory symptoms in NV day care centres, AC has higher prevalences of phlegm and cough

H=homes; S=schools

As it is noticeable from Table 1, the number of studies that examine this relationship in a clearly controlled and documented way is rather small. Further, the health outcomes are quite different and range from irritation symptoms and more general symptoms like tiredness to severe diseases like asthma or bronchial obstruction.

Unfortunately, the results do not point in the same direction. Five studies report no differences between the different types of ventilation systems, thereunder those with focus on the illnesses asthma and bronchial obstruction. Rhinitis and eczema seem to be less prevalent in completely naturally ventilated buildings, whereas irritation symptoms seem to be lower with a mechanical exhaust or balanced ventilation system. If differences are found, they include only one study. Probably, there are some confounders that may bias the results, like the unknown state of filters of mechanical systems or the ventilation rate that has been delivered in the rooms.

The study of Zuraimi et al. reported that in day care centres with mechanical ventilation or air-conditioning the ventilation rates were very low and lower as in naturally-ventilated centres. Children had a higher risk for respiratory symptom when they attended a mechanically or hybrid ventilated day care centre.

Supposed that in the aforementioned studies, the filters were adequately maintained and were no source for additional pollution, the conclusion might be that not the ventilation system itself is crucial but the amount of fresh air that is supplied in the room, i.e. ventilation or air exchange rate is much more important than the system itself.

2.2 The influence of the ventilation or air exchange rate on health

The research question in this part is whether an increased ventilation rate could contribute to a better health of the building occupants. Measurements have shown that concentrations of pollutants may be increased when ventilation or air exchange rate is too low (Wålinder et al., 1997a).

The literature search identified in total 10 studies with different physical measurements, thereunder the air change rate (ACH), i.e. how often per hour the air of the room is completely exchanged, the ventilation rate (VR), normally indicated by the amount of l/s-person of air supplied in the room or CO_2 concentrations, where a higher level stands for an inadequate ventilation and air exchange. The main characteristics of the studies are displayed in Table 2.

Table 2: Overview about the studies with focus on the ventilation rate and health in homes and schools

Study	Type of physical measurement	Outcome	Result
Bornehag et al., 2005b (H)	Mean ACH of cases and controls	Asthma, rhinitis, eczema	ACH in houses of children with diagnosed disease are lower
Hägerhed-Engman et al., 2009 (H)	VR above or below median	Asthma, rhinitis, eczema	ORs for all three health outcomes at low ventilation rate are higher, but did not reach significance
Mendell et al., 2013 (S)	Stepwise increase of VR about 1 l/s-person	Illness absence rate	Illness absence rate decreased significantly when ventilation rate is elevated
Muscatiello et al., 2014 (S)	CO_2 concentration above or below median	Teacher symptoms: mucosal membrane, lower respiratory, neuro-physiologic	Very broad and insignificant OR for all health outcomes, no differences
Myhrvold et al., 1996 (S)	CO_2 concentrations 0-999, 1000-1499 and above 1500 ppm	Headache, tiredness, irritation of airways	Headache, and tiredness increased sig. with higher CO2 concentration, airway symptoms increased as well, but showed a weaker correlation
Shendell et al., 2004 (S)	dCO_2 concentration (indoor –outdoor difference)	Absence rates	Absence rate increases about 10-20% with every increase of dCO_2 about 1000 ppm
Sun et al., 2011 (H)	ACH above or below median in winter	Wheeze, dry cough, rhinitis, eczema	Wheeze and dry cough sig higher when ACH below median
Turunen et al., 2014 (S)	VR in l/s-person	Illness absence, fatigue, headache, rhinitis, cough	Correlations between VR and health outcomes did not reach significance
Wålinder et al., 1997a, b (S)	High vs. low ACH	Mucosal swelling	Low ACH increased swelling of nasal mucosa and affect the airways.

H=homes; S=schools

As in the previous section, the measured health outcomes are very different. Taking a look on the influence of ventilation on absence rates, two studies showed that they could be influenced by the amount of ventilation that is available in classrooms. Airway infections like wheeze or cough are higher at a low air change rate, but this is significant in only one study.

In two other studies, airway diseases are not significantly influenced by the ventilation or air change rate.

Higher levels of CO_2 seem to promote headache and fatigue of teachers, but this reached significance as well in only one study. Airway symptoms were not influenced by CO_2 concentration and the calculated OR were very broad in one study (Muscatiello et al., 2014, e.g. the confidence interval of the OR for mucosal membrane symptoms and CO_2 concentration above median ranges from 0.20 to 11.87).

2.3 Further studies on ventilation and health

Some studies focus on the influence of ventilation in a broader sense on different health outcomes other have a different methodology and could not be compared with the aforementioned studies. The literature search identified in total 5 studies, the main characteristics are displayed in Table 3.

Table 3: Overview about the studies with focus on the ventilation in general and with different methodology

Study	Ventilation parameter	Outcome	Result
Dong et al., 2008 (H)	Use of ventilation device (exhaust fan, chimney, fume hood)	Cough, phlegm, asthma, wheeze, allergic rhinitis	No significant difference when ventilation device is used for all health outcomes
Kishi et al., 2009 (H)	Mechnical ventilation by duct or fan, existence in all rooms, regular operation	Sick house syndrome symptoms	No significant influence of the ventilation parameters.
Kovesi et al., 2009 (H)	Installation of home heat recovery ventilators (HRV)	Respiratory disorders of Inuit children	Reported rhinitis and wheeze were in part significantly reduced with HRV, hospitalizations did not change
Simons et al., 2010 (S)	Different ventilation problems	Absenteeism	Air intake near garbage storage, fresh air intake blockage, dirt in ductwork, damper malfunction, inadequate outside air or bad IAQ rating increase absence significantly.
Smedje et al., 2000 (S)	Installation of a new ventilation system	Allergies, asthma, asthmatic symptoms	Allergies or asthma incidences did not differ in rooms with or without new ventilation system, asthmatic symptoms were reduced

H=homes; S=schools

Two of these studies (Kovesi et al., 2009, Smedje et al., 2000) are intervention studies that aim at showing potential ameliorations of health status when a ventilation system is installed that increases air change rate. The results of both studies support the benefit of a newly installed ventilation system: at least some of the included health indicators showed reduced incidences. Nevertheless, the population of the first study is a very special one (Inuit children).

The three other studies used different classifications of ventilation parameters, two of them (Dong et al., 2008, Kishi et al., 2009) focus on the influence of partial ventilation like a fan or fume hood in the kitchen, no differences could be shown here. The third study (Simons et al., 2010) was a cross-sectional study that links reported ventilation problems with health outcomes. Health was impaired when any part of the ventilation system does not work properly, i.e. dirt in ductwork or wrong position of air intakes (near garbage storage). The conclusion of the last study is therefore, that health impairments emerge relatively clearly when the ventilation system does not work properly.

3 CONCLUSIONS

This literature search analysed in total 28 studies with different methodologies and measurements. The studies investigated homes and schools. One important conclusion is that more systematic research with detailed documentation about all possible confounding variables is necessary. The type of ventilation system seems to be less important compared to the ventilation rate itself and the amount of fresh air that is supplied in the room. Frequently, no differences are found between naturally ventilated and mechanically ventilated buildings. Sometimes, the mechanical ventilation system seems to promote prevalence of respiratory illnesses, sometimes people feel better in mechanically ventilated buildings. The state of the ventilation system plays an important role here to make conclusions so this must be documented better in future studies.

Low ventilation rates seem to interact with the prevalence of asthma and other respiratory diseases, but do not reach significance in all cases. Additionally, they promote absenteeism from school, as has been shown in two studies.

Nevertheless, all statements made above are derived from only one or two studies because the health outcomes are very different and not really comparable and therefore more studies on the same health outcome with different parameters of ventilation are needed to get a more clearer picture about the influence and to derive a health-optimal and energy efficient ventilation concept.

4 ACKNOWLEDGEMENTS

The authors express their gratitude to the graduate program “People Inside” at the Chair of Building Physics, University Stuttgart, Germany and Velux A/S, Denmark for a fruitful cooperation and support of this study. This work is benefitting from the research project Building I conducted at the Energy Campus Nuremberg, funded by the Bavarian State Government. The authors are responsible for the content of this publication.

5 REFERENCES

Bogers, R., Jongeneel, R., Van Kamp, I., Koudijs, E. (2011). Health and wellbeing in relation to the quality of ventilation systems in newly built dwellings in The Netherlands. Proceedings of Indoor Air 2011, Paper-ID 799.

Bornehag, C.G., Sundell, J., Hagerhed-Engman, L., Sigsgaard, T., Janson, S., Aberg, N., DBH Study Group. (2005a). ‘Dampness’ at home and its association with airway, nose and skin symptoms among 10,851 preschool children in Sweden: a cross-sectional study. *Indoor Air, 15*(Suppl. 10), 48-55.

Bornehag, C.G., Sundell, J., Hägerhed-Engman, L., Sigsgaard, T. (2005b). Association between ventilation rates in 390 Swedish homes and allergic symptoms in children. *Indoor Air, 15*(4), 275-280.

Clausen, G., Gustavsen, S., Buhl, S., Lagegaard, M.B., Callesen, M., Toftum, J. (2011). Indoor environment exposure and absenteeism in 151 Danish day care facilities. Proceedings of Indoor Air 2011, Paper-ID: 754.

Daisey, J.M., Angell, W.J., Apte, M.G. (2003). Indoor air quality, ventilation and health symptoms in schools: an analysis of existing information. *Indoor Air*, *13*(1), 53-64.

Dong, G.-H., Ma, Y.-N., Ding, H.-L., Jin, J., Cao, Y., Zhao, Y.-D., He, Q.-C. (2008). Housing characteristics, home environmental factors and respiratory health in 3945 pre-school children in China. *International Journal of Environmental Health, 18*(4), 267-282.
Ebbehoj, N.E., Meyer, H.W., Würtz, H., Suadicanni, P., Valbjørn, O., Sigsgarrd, T., Gyntelberg, F., Members of a working group und the Danish Mold in Buildings Program (DAMIB). (2005). Molds in floor dust, building-related symptoms and lung function among male and female schoolteachers. *Indoor Air, 15*(Suppl. 10), 7-16.
Emenius, G., Svartengren, M., Korsgaard, J., Nordvall, L., Pershagen, G., Wickman, M. (2004). Building characteristics, indoor air quality and recurrent wheezing in very young children (BAMSE). *Indoor Air, 14*(1), 34-42.
Engvall, K., Norrby, C., Norbäck, D. (2003). Ocular, nasal, dermal and respiratory symptoms in relation to heating, ventilation, energy conservation and reconstruction of older multi-family houses. *Indoor Air, 13*(3), 206-211.
Hägerhed-Engman, L., Sigsgaard, T., Samuelson, I., Sundell, J., Janson, S., Bornehag, C.G. (2009). Low home ventilation rate in combination with moldy odor from the building structure increase the risk for allergic symptoms in children. *Indoor Air, 19*(3), 184-192.
Kishi, R., Saijo, Y., Kanazwaw, A., Tanaka, M., Yoshimura, T., Chikara, H., Takigawa, T., Morimoto, K., Nakayama, K., Shibata, E. (2009). Regional differences in residential environments and the association of dwellings and residential factors with the sick house syndrome: a nationwide cross-sectional questionnaire study in Japan. *Indoor Air, 19*(3), 243-254.
Kovesi, T., Zaloum, C., Stocco, C., Fugler, D., Dales, R.E., Ni, A., Barrowman, N., Gilbert, N.L., Miller, J.D. (2009). Heat recovery ventilators prevent respiratory disorders in Inuit children. *Indoor Air, 19*(6), 489-499.
Mendell, M.J. (1993). Non-specific symptoms in office workers: a review and summary of the epidemiological literature. *Indoor Air, 3*(4), 227-236.
Mendell, M.J., Eliseeva, E.A., Davies, M.M., Spears, M., Lobscheid, A., Fisk, W.J., Apte, M.G. (2013). Association of classroom ventilation with reduced illness absence: a prospective study in California elementary schools. *Indoor Air, 23*(6), 515-528.
Mendell, M.J., Mirer, A.G., Cheung, K., Tong, M., Douwes, J. (2011). Respiratory and allergic health effects of dampness, mold and dampness-related agents: a review of the epidemiologic evidence. *Environmental Health Perspectives, 119*(6), 748-756.
Muscatiello, N., McCarthy, A., Kielb, C., Hsu, W.-H., Hwang, S.-A., Lin, S. (2015). Classroom conditions and CO2 concentrations and teacher health symptom reporting in 10 New York State Schools. *Indoor Air, 25*(2), 157-167.
Myhrvold, A.N., Olsen, E., Lauridsen, Ø. (1996). Indoor environment in schools – pupils health and performance in regard to CO2 concentrations. Proceedings of Indoor Air 1996, 369-374.
Norbäck, D., Björnsson, E., Janson, C., Widström, J., Boman, G. (1995). Asthmatic symptoms and volatile organic compounds, formaldehyde and carbon dioxide in dwellings. *Occupational and Environmental Medicine, 52*(6), 388-395.
Øie, L., Nafstad, P., Botten, G., Magnus, P., Jaakola, J.J.K. (1999). Ventilation in homes and bronchial obstruction in young children. *Epidemiology, 10*(3), 294-299.
Quansah, R., Jaakola, M.S., Hugg, T.T., Heikkinen, S.A.M., Jaakola, J.J.K. (2012). Residential dampness and molds and the risk of developing asthma: a systematic review and meta-analysis. *PLoS One, 7*(11), e47526.
Ruotsalainen, R., Jaakola, J.J.K., Rönnberg, R., Majanen, A., Seppänen, O. (1991). Symptoms and perceived indoor air quality among occupants of houses and apartments with different ventilation systems. *Indoor Air, 1*(4), 428-438.

Seppänen, O., Fisk, W.J. (2002). Association of ventilation system type with SBS symptoms in office workers. *Indoor Air, 12*(2), 98-112.
Seppänen, O.A., Fisk, W.J. (2004). Summary of human responses to ventilation. *Indoor Air, 14*(Suppl. 7), 102-118.
Seppänen, O., Fisk, W.J., Lei, Q.H. (2006). Ventilation and performance in office work. *Indoor Air, 16*(1), 28-36.
Seppänen, O.A., Fisk, W.J., Mendell, M.J. (1999). Association of ventilation rates and CO2 concentrations with health and other responses in commercial and institutional buildings. *Indoor Air, 9*(4), 226-252.
Shendell, D.G., Prill, R., Fisk, W.J., Apte, M.G., Blake, D., Faulkner, D. (2004). Associations between classroom CO2 concentrations and student attendance in Washington and Idaho. *Indoor Air*, *14*(5), 333-341.
Simons, E., Hwang, S.-A., Fitzgerald, E.F., Kielb, C., Lin, S. (2010). The impact of school building conditions on student absenteeism. *American Journal of Public Health, 100*(9), 1679-1686.
Smedje, G., Mattson, M., Wålinder, R. (2011). Comparing mixing and displacement ventilation in classrooms: pupils‘ perception and health. *Indoor Air, 21*(6), 454-461.
Smedje, G., Norbäck, D. (2000). New ventilation systems at selected schools in Sweden – Effects on asthma and exposure. *Archives of Environmental Health, 55*(1), 18-25.
Sun, Y., Zhang, Y., Bao, L., Fan, Z., Sundell, J. (2011). Ventilation and dampness in dorms and their associations with allergy among college students in China: a case-control study. *Indoor Air*, *21*(4), 277-283.
Sundell, J., Levin, H., Nazaroff, W., Cain, W.S., Fisk, W.J., Grimsrud, D.T., Gyntelberg, F., Li, Y., Persily, A.K., Pickering, A.C., Samet, J.M., Spengler, J.D., Taylor, S.T., Weschler, C.J. (2011). Ventilation rates and health: multidisciplinary review of the scientific literature. *Indoor Air, 21*(3), 191-204.
Turunen, M., Toyinbo, O., Putus, T., Nevalainen, A., Shaughnessy, R., Haverinen-Shaughnessy, U. (2014). Indoor environmental quality in school buildings and the health and wellbeing of students. *International Journal of Hygiene and Environmental Health, 217*(7), 733-739.
Wålinder, R., Norbäck, D., Wieslander, G., Smedje, G., Erwall, C. (1997a). Nasal mucosal swelling in relation to low air exchange rate in schools. *Indoor Air, 7*(3), 198-205.
Wålinder, R., Norbäck, D., Wieslander, G., Smedje, G., Erwall, C. (1997b). Nasal congestion in relation to low air exchange rate in schools. *Acta Oto-laryngologica, 117*(5), 724-727.
Wålinder, R., Norbäck, D., Wieslander, G., Smedje, G., Erwall, C., Venge, P. (1998). Nasal patency and biomarkers in nasal lavage – the significance of air exchange rate and type of ventilation in schools. *International Archives of Occupational and Environmental Health, 71*(7), 479-486.
Wargocki, P., Sundell, J., Bischof, W., Brundrett, G., Fanger, P.O., Gyntelberg, F., Hanssen, S.O., Harrison, P., Pickering, A., Seppänen, O., Wouters, P. (2002). Ventilation and health in non-industrial indoor environments: report from a European Multidisciplinary Scientific Consensus Meeting (EUROVEN). *Indoor Air, 12*(2), 113-128.
Zuraimi, M.S., Tham, K.W., Chew, F.T., Ooi, P.L. (2007). The effect of ventilation strategies of child care centers on indoor air quality and respiratory health of children in Singapore. *Indoor Air, 17*(4), 317-327.

EVALUATION OF VENTILATION AND IAQ PARAMETERS MEASURED IN SOCIAL HOUSING IN MADRID

Sergio Rodríguez-Trejo*[1], Sergio Vega[2], Consuelo Acha[2]

1 Universidad Politécnica de Madrid
Avda. Juan de Herrera 4
Madrid, Spain

2 Universidad Politécnica de Madrid
Avda. Juan de Herrera 4
Madrid, Spain

**Corresponding author: sergio.rodriguez@upm.es. Phone number: 0034913363891*

ABSTRACT

Within this paper, an evaluation of Indoor Air Quality in residential buildings, and the experience after a building retrofit is shown. One residential building in a Madrid social housing neighbourhood serves as case study and base for the monitoring.

During the last decades, increasingly in the last years, energy conservation in buildings has become a major concern, as it represents an important share of the global energy use and contributes a great deal the GHG global emissions. This concern leads to promote retrofitting actions in buildings to improve their envelope thermal behaviour and reduce energy demand for HVAC. These retrofits affect the indoor air quality conditions as they change the original configuration of the building, including the ventilation systems. Occupant behaviour and material emissions are main sources of indoor environment pollutants. Atmospheric environment represents also a major challenge in big cities, as the pollutant levels overcome frequently the threshold levels set up in international regulations. Indoor environment quality is affected clearly by all the aforementioned factors, and residential buildings are especially sensitive to all of them as regulations have been less strict for this important building stock. Within this paper, an evaluation of the actual transient indoor air quality conditions depending on all factors influencing is developed. This characterization of a residential unit which is established in a social housing neighbourhood in Madrid (Ciudad de los Angeles), an area which has been object of a deep retrofit process during the last years. Envelope improvement, which is also the most common intervention in this neighbourhood, was executed during the last year. The intervention consisted in wall cavity insulation via injecting glass wool. Indoor environment variations are analyzed and discussed. Monitoring of the indoor comfort and air quality conditions was done before and after the retrofit process. Indoor contaminant sources coming from material and human activity are considered. Actual monitoring is also considered in unoccupied and occupied units. As conclusions, the results from monitoring some aspects of IAQ will be compared to the existing regulations on residential buildings.

KEYWORDS

IAQ, ventilation, residential building retrofit, indoor pollutants

1 INTRODUCTION

Uses and habits in developed societies show that around 90 % of the time is spent indoors [ECA, 2000] and the main share of this time, from 60 % to 70 % is spent in residential indoor spaces [Thatcher and Layton 1995]. In the case study shown in this paper, the share of time spent in houses is even higher, around 80 %, as the most of occupants are around 80 years old. Physical and chemical properties of air in these environments have been thoroughly studied and different regulations have set the parameters to guarantee a healthy and satisfactory indoor environmental quality from subjective comfort and wellness aspects, as the Percentage of Dissatisfied (PD) [ISO 7730:2005] to health and illness correlation pattern issues [WHO,2000]. These aspects, and the progressive worsening of the urban environment due to traffic and heating been mostly based on fossil energy sources in cities as Madrid, make it necessary to re-adjust the way to achieve the objectives of lowering the energy demand, evaluating and including in a more precise way the effect on indoor quality due to ventilation systems. Regulations and standards for keeping indoor environment in housing are less strict than those existing in other building types and in European regulations, as the case of Spain, the impact of ventilation for energy calculations is usually underestimated [Sotorrio et al,2013].

The SIREIN project (Integral Systems for Energy Retrofitting) 2011-2015, aimed at bearing down barriers to reach the objective of a deep and wide spread retrofitting process in the residential stock. The solutions selected and investigated, are aimed at reaching the maximum cost-benefit ratio via making use of the Pareto Principle (20% of the causes account for 80% of consequences). By using this principle, the focus is set on the most common building type, user type climate boundary conditions, and combining the most cost-effective technical solutions, we will obtain the most suitable combined solutions to apply to a large stock of buildings, minimizing the need of public and private investment and maximizing the effectiveness of the actions, both in energy and sustainable savings. Leaded by Saint Gobain Isover and the Technical University of Madrid, the project was funded by the National Ministry of Science and Innovation with grant number IPT-2011-1980-920000.

1.1 Environmental pollution in Madrid metropolitan area

Atmospheric environment is currently a major concern for developed countries, especially in cities. Since the 1990's, urban pollution is monitored in many of european cities (Airqualitynow Project 2010), and the measured values, though in global have diminished from 2005, still are in many occasions above the reference maximum values established by different Organizations. The European Union has developed an extensive body of legislation which establishes health based standards and objectives for a number of pollutants in air. The so-called CAFE (Clean Air for Europe programme) Directive (2008/50/EC) was published on 21st May 2008 and merges earlier directives into a single directive on air quality. The standards and maximum values are taken from this Directive.

As it can be seen in figure 3, target limits would be overcome in daily basis quite frequently, though they are not on yearly basis. The annual report for 2014 "Air Quality in the Madrid city area" [Madrid region government, 2014] shows that the general concentrations are below the target limits, and still the perception of the air quality within the population is not satisfactory. In general all the values have decreased, excepting those from Troposphere Ozone (during summer) since 2005 but the traffic and the use of fossil energy sources produce high levels outdoors and indoors that have to be reduced. Most of the pollutants, but specifically NO2, NOx, CO and benzene concentrations indoors are highly related with the concentration outdoors.

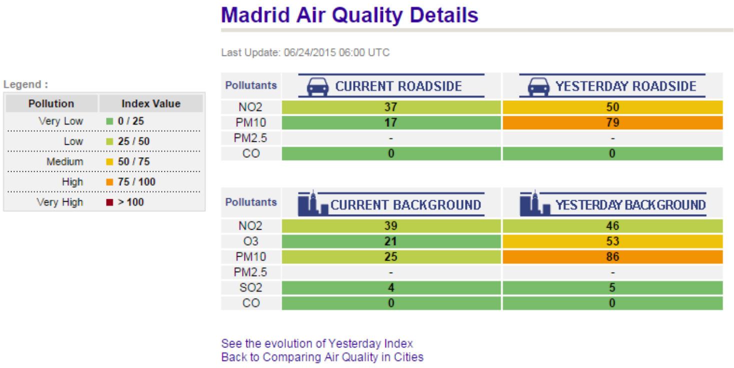


Figure 3: Daily pollutant monitoring extracted from Madrid City Council sources in web http://www.airqualitynow.eu/

For the purposes of this study it can be assumed that the indoor concentration of these atmospheric pollutants will not be higher than outdoors. The usual solution to solve this high concentration of pollutants in these events of high pollution outside requires filtering the intake air.

1.2 Indoor pollution in residential multi-storey buildings

Indoor pollution inside residential multi-storey buildings has been analyzed and measured thoroughly in Europe, Japan, United States (Järnström 2008) but it is not deeply studied in Spain. This lack of reference values to evaluate the indoor conditions in Spanish residencies and especially social housing and other representative building stock needs to be changed. In this study several measurements are done.
Usual pollutants found indoors are oxides of nitrogen (NOx), sulphur (SOx) and carbon (CO, CO2). These compounds originate from combustion processes as heating and traffic (WHO 1989). Formaldehyde, Total Volatile organic compounds(TVOCs), semi volatile organic compounds (SVOCs), Volatile organic compounds(VOCs) are present in indoor environment and are usually related to material and furniture emissions, human activity, and vary with temperature and humidity changes, as well as depend on the air renovation rate (Järnström 2008).

2 METHODOLOGY AND MATERIALS

The objective of this paper is to evaluate aspects related to the Indoor Air Quality (IAQ) of a representative building unit case study, based on the monitoring of different pollutants commonly present indoors, focusing on dynamic variations of concentrations of indoor pollutants and actual ventilation rates and emission rates.
This evaluation considers different aspects:

1. Measurement of the environmental conditions before and after the building retrofit.
2. Consideration of the occupant profile and occupancy schedules
3. Levels of contaminants due to material emissions
4. Ventilation rates and infiltration

The evaluation begins at studying the case study for the retrofit: The building, the occupants, and the retrofit. Analysis will continue with the material emissions, the ventilation and infiltration drivers.

2.1 The case study

Although building retrofits are still not spread as widely as it would be desirable in Spain, there are specific areas as “Ciudad de los Ángeles”, in the metropolitan area of Madrid, Spain, where a specific plan to promote a neighbourhood retrofit was developed, beginning in 2005, and including a wide range of interventions, but mostly focused on envelope improvement [ARI Ciudad de los Angeles, 2005].”Ciudad de los Ángeles”, erected at the end of the fifties, accounts for almost 8.000 dwellings, in 485 buildings, and hosts a population of around 24.250 inhabitants. The socio-economic characteristics of this area are the low income of the neighbours and the advanced age of people. Although new inhabitants have been coming to the neighbourhood in the last years, the economic situation is still modest. Most of the buildings lacked correct groundings, elevators and basic thermal insulation and energy efficiency in the utilities. So the retrofit actions were focused mostly on these interventions. The public investment reached a maximum ratio of 21.000 € per dwelling (around 75% of the total costs). The public entity EMVS (Social Housing Enterprise of Madrid) managed the interventions and approved the actions. The actions were only publicly funded when affecting the whole building.

2.2 The building characteristics

Within the SIREIN research project, a demonstration building was insulated by injecting glass wool inside the wall cavities. The demo is an 8 stories building, with a representative typology within the area and the Spanish residential stock. 16 dwellings, elevator, utility rooms below the first floor and stairs are the main spaces in the building. The housing units surface is around 70 m^2 each, and have cross ventilation due to the three orientations of the facades on each apartment. Fig 1 and 2 show the configuration of the demo-building. The building was built during the sixties, on a concrete matrix structure and a double layer façade. These are the main features:

- Opaque envelope formed by a double brick layer with a 6 cm wide void cavity between them. The finishing to the inside is gypsum tiling and concrete layering painted to the outside.
- Windows are aluminum framed with single glass pane and no thermal bridge break.
- Some of the houses have solar shading elements to south, east and west orientation.
- Heating and hot water is provided by gas boilers excepting one of all the 16 which uses an electric boiler, and other which uses no heating.
- Some of the housing units installed Air conditioning with a divided system.
- Housing units have installed water and electricity, although these utilities are old and should be renovated.

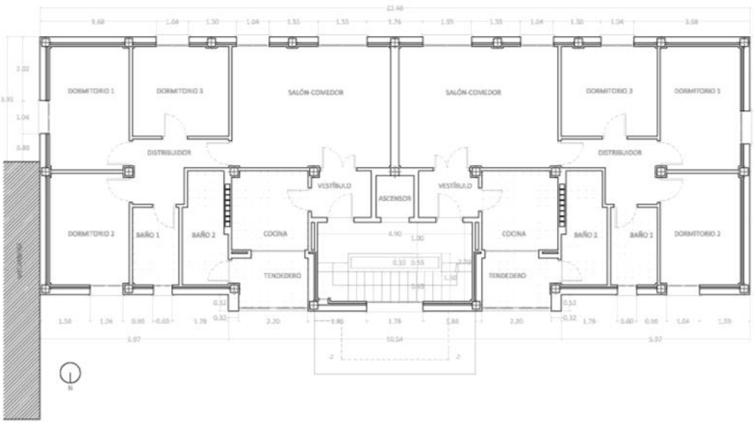


Figure 1: south façade of the demo case

Figure 2: Floor plan of the demo case

2.3 The retrofit

The intervention in the building was intended to improve the façade thermal resistance. Its U value (Thermal transmittance) was measured, helped by flux meter, from 1,50 W/m2K to 0,34 W/m2K in the opaque façade, which is consistent to the theoretical values obtained. Of course, great attention was paid to minimizing the risk of condensation inside the cavity, which was calculated according to the regulation in the building Code. The analysis performed show that internal condensation risk is not relevant, fact that has been confirmed with periodic visits to the building during the last year.

The intervention, not being a deep retrofitting example, has improved the building global performance, showing differences caused in the majority by the difference between users, but accounting for a significant reduction in energy consumption. Previous research show for our case study in the location of Madrid, and a cavity width of 60 mm injection with base of glass wool reduces the annual energy demand in circa 25% (Hernandez et al., 2014).

2.4 Monitoring and measurement

The measurement of actual concentrations and variations has been done helped by Wireless IAQ (Indoor Air Quality) Profile Monitor unit, from PPM technologies Ltd. The IAQ Profile system has been designed to give a visual representation of indoor air quality in buildings, as part of the buildings management standards in relation to conditions such as Sick Building Syndrome. Since a great number of units can be networked, the system can show precise changes in concentration of selected IAQ parameters in various locations over time. In the experiments, only one unit has been connected to the grid, set up in the living room of the different units.

A continuous monitoring system has been set up to observe the effectiveness on the thermal resistance improvement of the opaque envelope (operative Temperature, superficial temperature inside and outside the cavity) but also other higrothermal and indoor quality indexes were monitored; Relative Humidity (RH), CO_2, Formaldehyde, and TVOCs were measured.

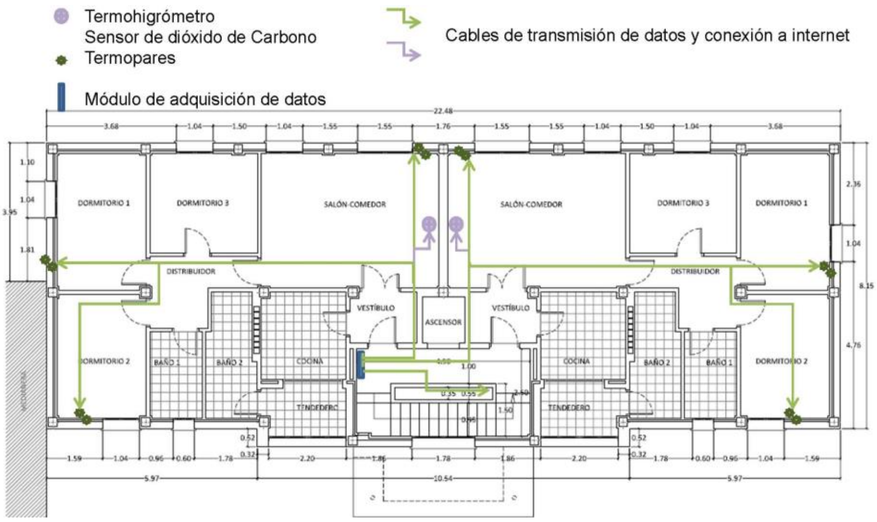


Figure 3: Monitoring set up

2.5 Occupant profiles and behaviour

The residential units are occupied by aged people, around 80 years old. They have a sedentary live style and do not go out often, unless they visit the doctor. There are two persons per apartment. The usual schedule for them is:

Table 2: Occupancy schedule

Room	Monday to friday	% occupancy	Saturdays and sundays	% occupancy
Bed room	0:00 – 7.00	100	0:00 – 9:00	100
Living room	15:00-16:00	50	10:00-12:00	50
	19:00-21:00	100	15:00-17:00	100
	21:00-22:00	50	19:00-21:00	50
	22:00-24:00	100	21:00-22:00	100
			22:00-24:00	100
kitchen	8:00-9:00	100	8:00-9:00	100
	14:00-15:00	50	14:00-15:00	50
	21:00-22:00	50	21:00-22:00	50
Bath room	7:00-8:00	1 person	9:00-10:00	1 person

The trends of this specific type of user allow us to be confident in the behaviour trend to be kept constant along days and to take conclusions from the measurements and the conclusions inferred by them.

3 RESULTS AND DISCUSSION

3.1 Ventilation and infiltration

Regarding ventilation, most of the buildings in the area rely ventilation merely on the windows and openings operated by users, assuming infiltration is a major contribution of the total amount of the air renovated inside of the dwellings. They lack of any intake or exhaust vent. The case study building was built before any regulation regarding thermal insulation or ventilation were applied (in Spain this is in 1979 with the NTE 79), so the ventilation scheme could be assimilated to a multizone indoor space, for which the different orientations and temperatures, wind velocities and wind pressure can be taken to calculate the renovation rate. For the calculations of the air coming inside the building, reference has been taken from the regulation EN 15242.

$$q_{vairing} = 3{,}6 * 500 * A_{ow} * V^{0{,}5} \quad (1)$$

$$V = C_t + C_W \cdot Vmet^2 + C_{st} \cdot H_{window} \cdot abs(\theta_i - \theta_e) \quad (2)$$

Aided by the difference of temperatures in the different facades cavities we can estimate the force induced to produce the cross ventilation. Fig. 4 and 5 show the differences in facades and inside the cavities, before the intervention.

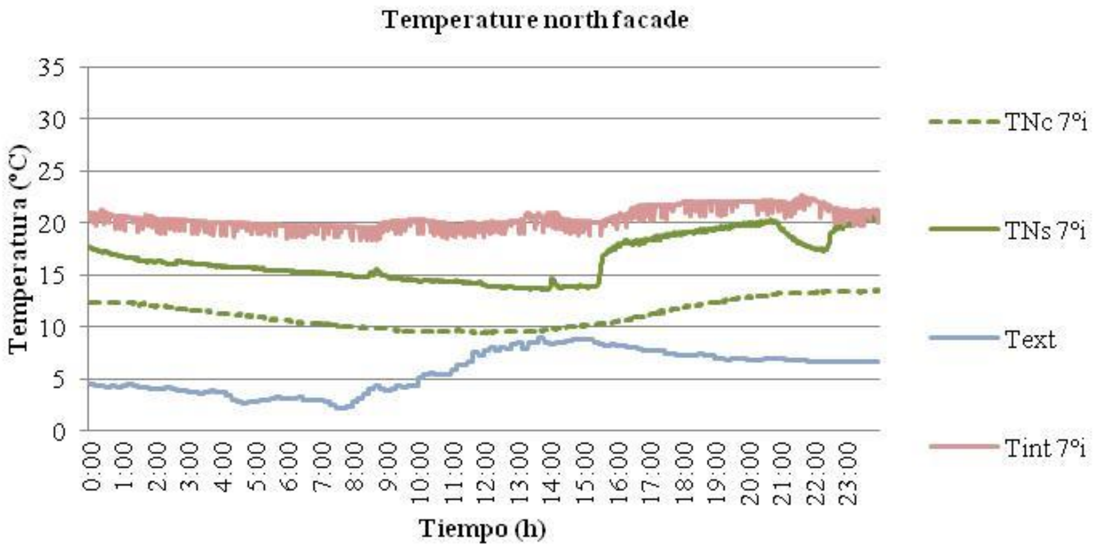


Figure 4: North façade. Temperature

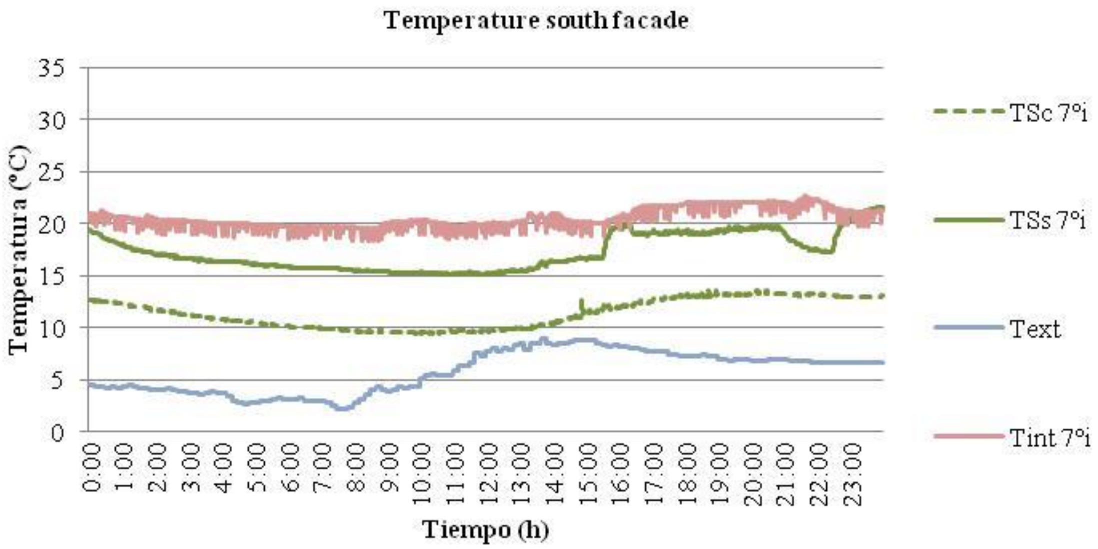


Figure 5: South façade. Temperature

The ventilation produced inside the apartments is not counting on stack effect, as the only duct which could contribute to this aspect is found in the kitchen, with the complementary exhaust duct, reaching the roof. So only infiltration, and operation of windows and pressure and temperature differences between facades and indoor and outdoor temperatures is responsible for providing the adequate renovation air. In many occasions the renovation rate is not enough in this apartments to guarantee a pleasant and healthy atmosphere.

3.2 Indoor pollutants in the case study

Contaminant concentrations and emission sources have been established by reference values taken from literature and actual values taken from measurement During the monitoring one unoccupied apartment has been monitored to extract materials and basal concentration of pollutants. Figures 6 and 7 show basal concentrations in the unoccupied apartment. We have estimated a renovation rate of 0,19 ACH obtained from the CO2 dilution method (Roulet et al., 2008).

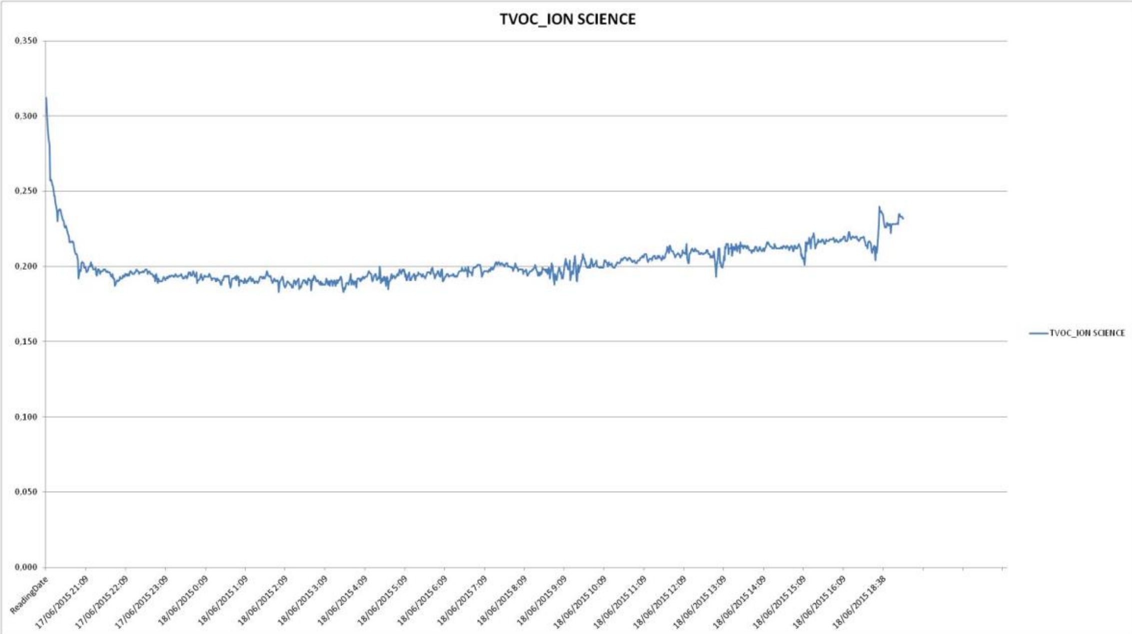


Figure 6: TVOC concentration measurement from the unoccupied apartment

The sum of all individual VOCs, referred to as TVOCs (total volatile organic compounds), measured in mg/m3, are often used as a guide to determine whether chemical levels are elevated in air samples. These levels measured in the unoccupied apartment are nearly constant (fig. 6), and maintains levels similar to LEED's standard levels of 500 µg/m$_3$, established for new houses. In new office buildings, the TVOC concentration at the time of initial occupancy is often 50 to 100 times higher than outdoor air. Occupants almost always complain when TVOC levels are 3000 µg/m$_3$ or higher. IAQ guidelines have been incorporated into the design and construction specifications for new buildings and to a lesser extent homes to reduce VOCs to allowable levels on the basis of current toxicologic information on health, irritation, and odour hazards.

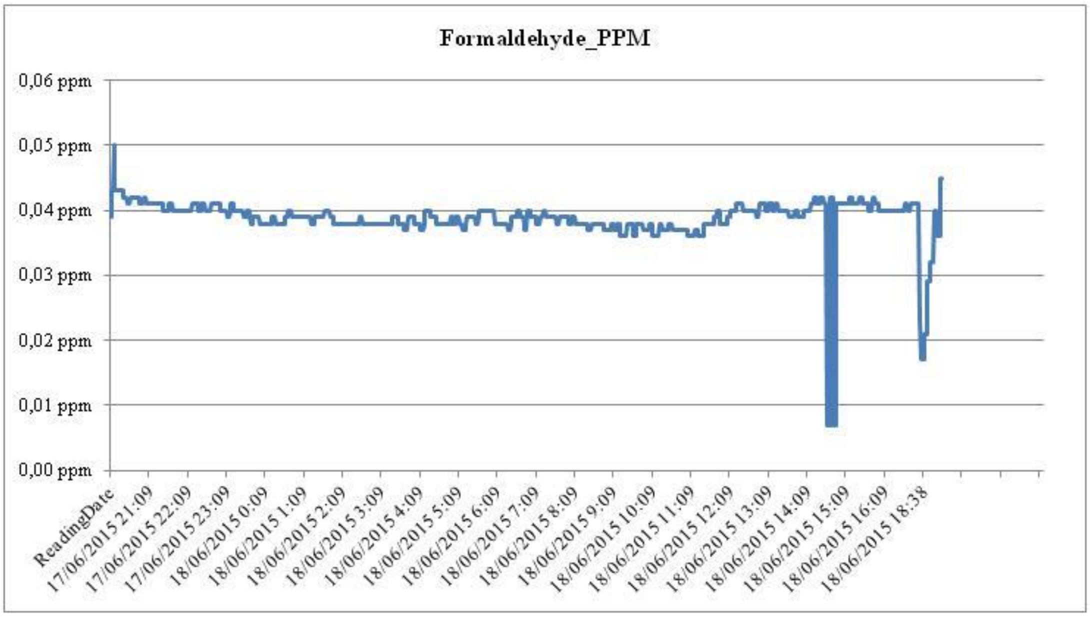


Figure 7: HCHO concentration measurement from the unoccupied apartment

Formaldehyde, among others, is one of the main pollutants in indoor environments, mostly found in offices [Wolkoff, 2013], but not negligible in residences [WHO 2010]. Primary formaldehyde emitters are paints, adhesives, insulations, cabinetry, workstations, ceiling tile, and wallboard. Building occupants and activities are also major sources of these indoor chemicals. The concentration at the unoccupied apartment is kept quite below the standards limits (0,1 mg/m3 is the WHO recommended threshold for 30 min). The major sources of formaldehyde are from indoor construction materials such as particleboard, fiberboard, and plywood. Formaldehyde concentrations are higher in residential buildings compared with office buildings because of the relatively large ratio of pressed wood products to air volume in homes. This concentration is higher in occupied apartments reaching and overpass established levels frequently due to emissions from furniture, materials and occupants as seen in fig. 7.

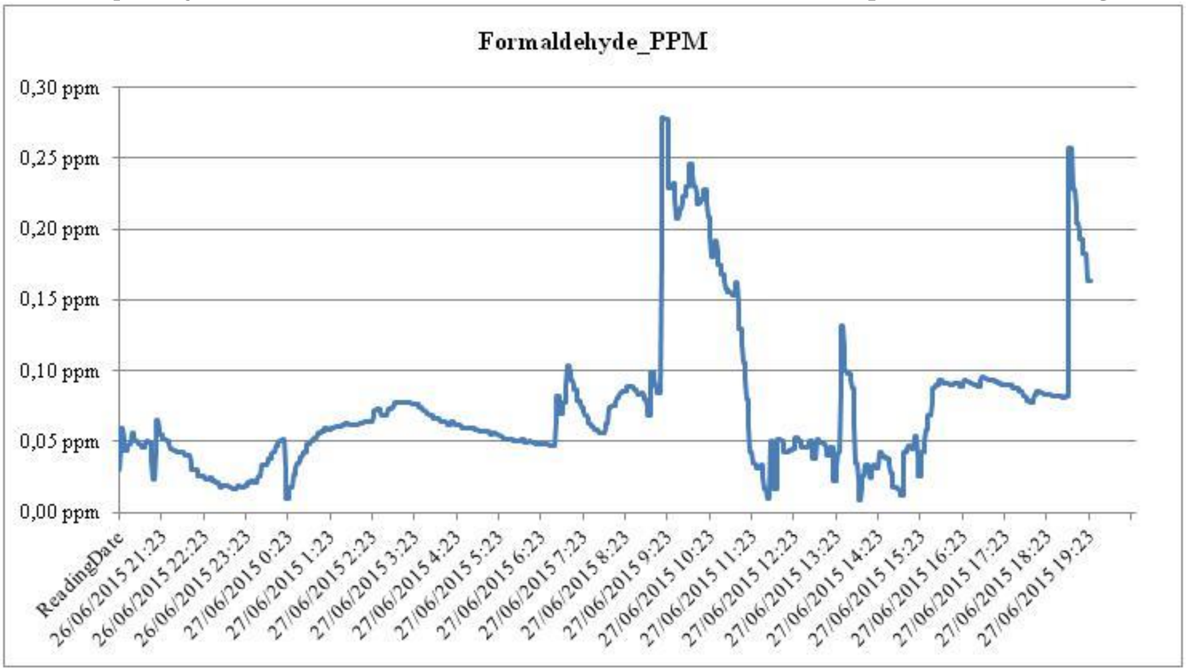


Figure 7: HCHO concentration measurement from the occupied apartment

3.3 Results after the renovation

In the apartments, the retrofit was made during winter. The results in terms of energy demand have been studied for annual consumption, taking as reference the year before and the year after the intervention. The results of the renovation, were significant both in thermal performance of the building and the pollutant levels. There has been a reduction of the energy consumption for heating in around 20%. Other observed phenomenon has been the reduction of the difference between the air temperature in the cavity and the temperature in the surface, as consequence of the reduction in the infiltration through the cavity. Infiltration in the cavity has been stabilized and the consequences of it can be observed in the temperature graphics as can be seen in fig .8 and 9

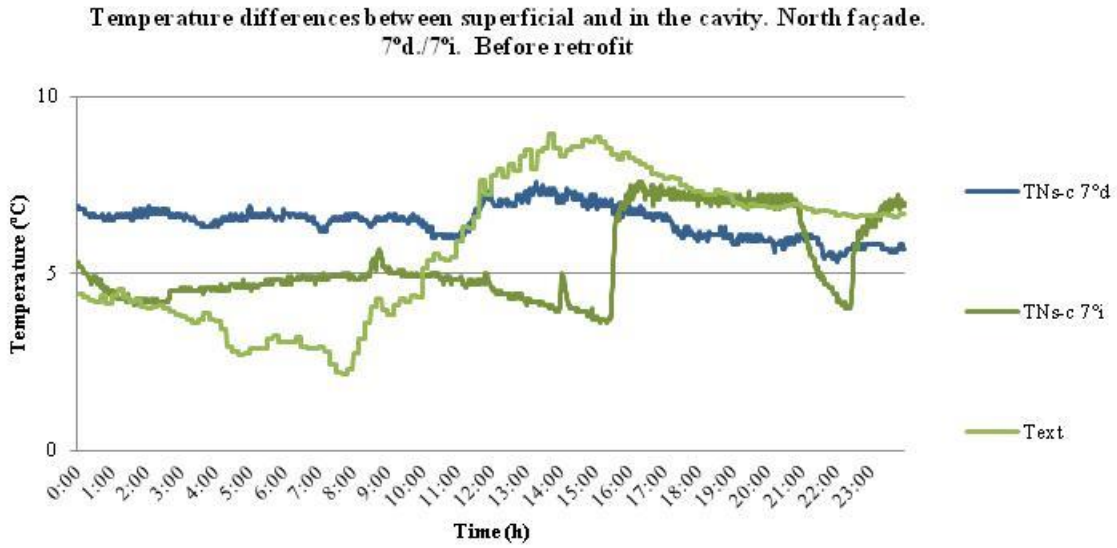


Figure 8: Temperature difference between wall surface and inside the cavity taken from the occupied apartments before the retrofit

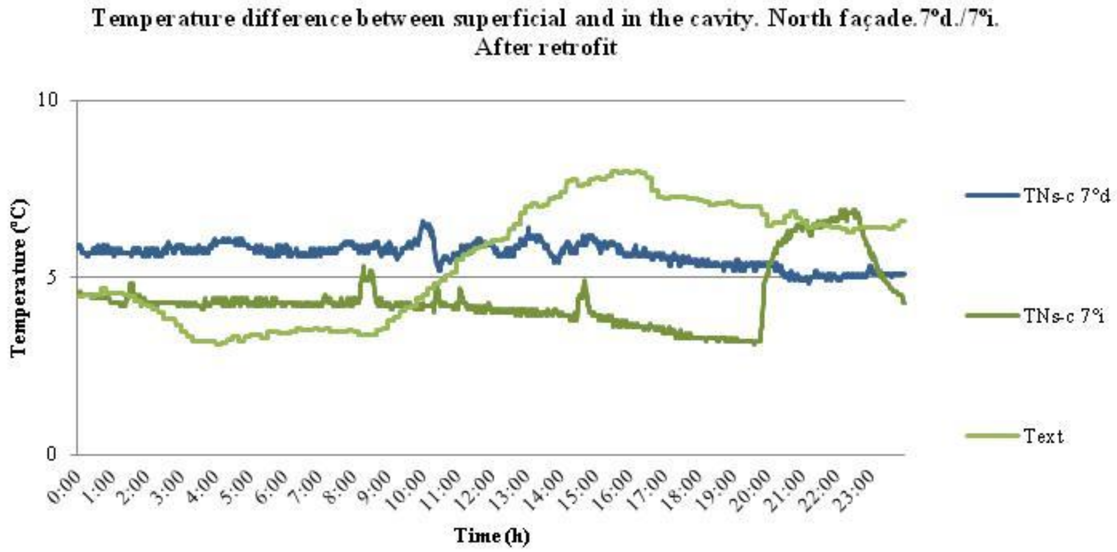


Figure 9: Temperature difference between wall surface and inside the cavity taken from the occupied apartments after the retrofit

We have compared two monitoring days, with similar outdoor and indoor conditions. Before the retrofit process, the concentration of CO2 found inside of the apartments was unusually high in some of the apartments, as only two inhabitants were present in the residence. This can be seen in figure 10. CO2 concentration in apartment 7ºi reached values above 2000 ppm. After retrofitting, we can observe in figure 11 values not only are still above 2000 ppm at some point but the time above this limit is longer. This increased value of CO2 levels after the renovation was a not expected side effect. Searching for an explanation, for these high levels, we consider the existence of fewer ACH (air changes per hour) due to the improved envelope properties and less associated leakiness. The improvement of the wall cavity has some benefit in the infiltration, reducing it, so that the renovation of air through the leaks is less important, showing up in the CO2 levels. We could expect that the users, being able to control the thermal environment, due to the façade improvement, would tend to open the windows more often but they don't seem to notice the higher concentration of CO2, as the windows don't seem to be open to let fresh air inside.

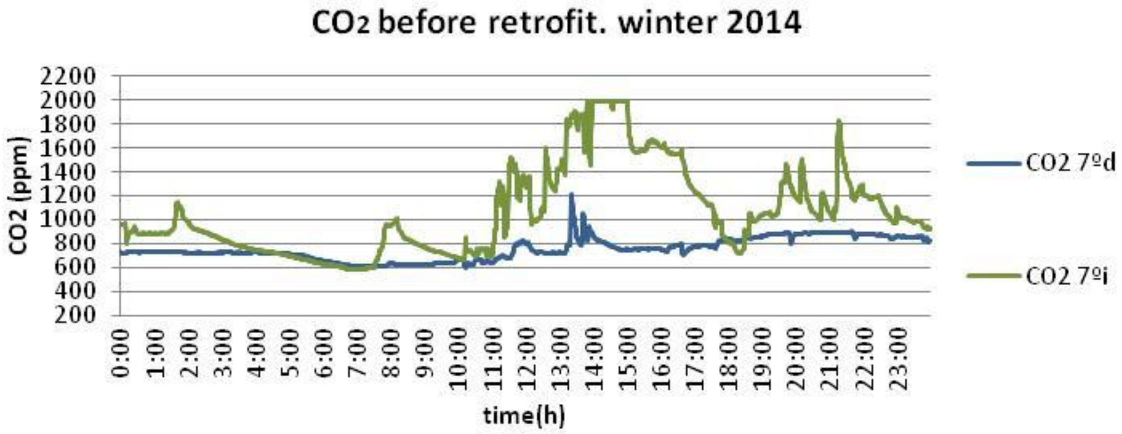


Figure 10: CO_2 concentration measurement from the occupied apartments before the retrofit

Figure 11: CO_2 concentration measurement from the occupied apartments after the retrofit

There seem to be minor inconsistencies in the schedules extracted from the concentration of pollutants and Humidity shown in the monitoring. These corrections to be made, can better establish the user profile to be able to calculate the actual thermal and IEQ behavior of the apartments in which this retrofit was done.

4 CONCLUSIONS

In this paper, an evaluation of different aspects related to IAQ in a residential representative social housing unit is done; before and after the envelope retrofit works. The main conclusions of the paper are referred to the concentration of different pollutants.
As the main conclusion of the paper, we can affirm that building retrofit, even if not directly affecting the envelope as other solutions, as ETHICS (External Thermal Insulation Composite Systems) produce direct variations in indoor environment quality and indexes due to complex interactions that generate a comfortable environment. This could be measured in reduction of both, infiltration, and concentration of contaminants.
In the case of TVOCs, the emission from materials is found to be low for the unoccupied apartments. It shows to be near to LEED standard levels of 500 $\mu g/m^3$. These levels are not

found in occupied apartments, where other factors, as occupants, cleaning products, furniture and equipment contribute to an elevation of these levels.
Similar case occurs referring to Formaldehyde (HCHO). The concentration in the unoccupied apartment was considerably lower than in occupied ones (0,04 ppm to 0,12 ppm). The mean value was stable in the unoccupied piece but suffered significant variations in occupied environments even thought temperature and humidity was kept stable.
The other gas concentration measured was CO_2. The concentrations varied from before to after the retrofit. Peak values of 2000 ppm were found before and after the retrofit but the frequency of this high values was increased after the retrofit was performed.
Another point to remark is that this variation was observed also with a reduction of the variations in the cavity. The theory suggests that leakage in the envelope can reduce effective ventilation and air change, and this is possibly the case. The improvement in the façade hermeticity reduces the infiltration. Changing the windows would have to be studied in detail to guarantee the air change and not to induce moisture and mould growth.
The case study suggests that retrofit has changed the energy behaviour of the building, as well as the environmental conditions, and pollutant concentration levels. Further studies are being conducted on same demo.

5 ACKNOWLEDGEMENTS

This work is part of the Ph.D. Thesis research being conducted by the main author. Special thanks to the Main Researcher of the project and the Project leader, which made this work possible.

6 REFERENCES

Thatcher, T. and Layton, D. 1995. “Deposition, Resuspension, and Penetration of Particles within a Residence.” Atmospheric and Environment. Vol. 29. Pp. 1487-1497

Thatcher, T. and Layton, D. 1995. “Deposition, Resuspension, and Penetration of Particles within a Residence.” Atmospheric and Environment. Vol. 29. Pp. 1487-1497

Ergonomía del ambiente térmico. Determinación analítica e interpretación del bienestar térmico mediante el cálculo de los índices PMV y PPD y los criterios de bienestar térmico local (ISO 7730:2005).

World Health Organization, 2010. “WHO guidelines for IAQ: Selected Pollutants”.

Sotorrío Ortega, G.; Linares Alemparte, M.P.; García Ortega, S., 2013.”Investigación prenormativa para la revisión del documento de calidad del aire interior”.

“SIREIN: Sistema Integral de Rehabilitación Energética” (IPT-2011-1980-920000). Funding Entity: Ministry of Science and Innovation. Partners: SAINT GOBAIN, UPM, TECNALIA, EMVS, R7 CONSULTORES, KOMMERLING, ENERES-Fernandez Molina.

www.airqualitynow.eu: an interactive web service to display and compare air quality across cities.

Direccion General de Sostenibilidad y Planificación de la Movilidad, 2014. “Calidad del aire:2014”.

Järnström, Helena, 2008 “Reference values for building material emissions and indoor air quality in residential buildings”

WHO, World Health Organization. 1989. Indoor Air Quality: Organic Pollutants. EURO Reports and studies No. 111. Copenhagen: World Health Organization.

El A.R.I. de Ciudad de los Ángeles: rehabilitación de viviendas. http://www.espormadrid.es/2009/03/el-ari-de-ciudad-de-los-angeles_22.html

Hernández, Belén, Rodriguez, Sergio; Delgado, Elda; Contreras, Cristóbal; Vega, Sergio; Pallares, Ana;2014. II Congreso de Edificios de Energía Casi Nula. ISBNAE2014.14002976 “Edificios de ECN mediante el sistema combinado de inyección en cámara y solución de puentes térmicos. Demostrador a escala real: edificio de viviendas”

Ventilation and Airflow in Buildings, Methods for Diagnosis and Evaluation , 2008. ISBN-13: 978-1-84407-451-8

P. Wolkoff /Indoor air pollutants in office environments: Assessment of comfort, health, and performance International Journal of Hygiene and Environmental Health 216 (2013) 371–394”.

The European Parliament and The Council Of The European Union. Directive 2008/50/EC 21 May 2008 on ambient air quality and cleaner air for Europe

INFLUENCE OF AIR QUALITY PERFORMANCE REQUIREMENTS ON THE DEMAND OF ENERGY

José Manuel Salmerón Lissén[*1], Javier García Ramos[1], Servando Álvarez Domínguez[1], José Luis Molina Félix[1], Francisco José Sánchez de la Flor[2]

1 Grupo de Termotecnia. Seville University
Camino de los descubrimientos S/N
Escuela Técnica Superior de Ingenieros
Seville 41092, Spain
**Corresponding author: jms@us.es*

2 Cádiz University
C/Chile 11002
Cádiz, Spain

ABSTRACT

The aim of this paper is to show the effects of variable ventilation rates on the demand of energy and air quality in dwellings, and how airtightness and wind affect this relation. It is interesting to estimate the relation between the air ventilation rate and airtightness of dwellings which makes the dwelling to be under-pressure in order to avoid infiltrations.

The main aspects discussed in this paper are:

4. Influence of wind direction and airtightness in infiltration and the effects in IAQ.
5. Influence of air quality performance requirements in the demand of energy and IAQ.

KEYWORDS

IAQ, Equivalent ventilation rate, Airtightness, Infiltration, Ventilation, Air change rate n50

1 INTRODUCTION

The International Energy Agency (IEA) [1] states that buildings represent 32% of the total final energy consumption, which means almost 40% of the primary energy consumption. The need for reduction of CO_2 emission leads to paying attention to the energy demand in buildings. According to the IEA, ventilation and infiltration are responsible for about 33% of total space conditioning energy usage calculated as average of 13 countries joining the tertiary and residential sectors.
On the other hand the Department of Energy in the University of Seville has estimated this percentage for Spanish residential buildings to be between 40-50%. Thus, there exists a potential energy saving in reducing ventilation and infiltration rates.
This paper carries out simulation studies to determine the influence of ventilation and infiltration rates on energy demand and IAQ, highlighting important parameters such as wind direction and airtightness which affect the results significantly.

2 DEFINITIONS

In order to help the reader to properly understand the contents presented in this paper, some definitions are provided:

- Ventilation: Air Infiltration and Ventilation Center [2] defines ventilation as the process by which ‘clean’ air (normally outdoor air) is intentionally provided to a space and stale air is removed. This may be accomplished by either natural or mechanical means.
- Infiltration: It is the process by which the air goes inevitably into a building through adventitious or unintentional gaps and cracks in the envelope.
- Ventilation and infiltration load: It is the energy required to maintain the building in thermal comfort conditions due to air ventilation and infiltration. The amount of air that enters into the building depends on the following factors:
    - Wind speed and direction which affects over-pressure or under-pressure in the façade.
    - Size, location and permeability of elements in the façade.
    - Stack effect.
    - Air extractors which produce under-pressure.
- Air change rate n50: It is a measure of the global airtightness in buildings using a pressure difference of 50 Pascal. This value can be obtained analytically using the following expression:

$$n_{50} = airtight\_wall\_4Pa(m^3/m^2h) \cdot \frac{A_{wall}(m^2)}{Vol(m^3)} \cdot \left(\frac{50}{4}\right)^{0.67} + airtigh\_window\_100Pa(m^3/m^2h) \cdot \frac{A_{window}(m^2)}{Vol(m^3)} \cdot \left(\frac{50}{100}\right)^{0.67} \quad (1)$$

3 VENTILATION MODELS

Among the different ventilation models, this paper focuses on the two well-known models called single zone in pressure/single zone in flows (single-zone model) and multizone in pressure/multizone in flows (multizone model).

3.1 Single-zone model

A single-zone model is one in which internal partitions are not considered. Although these types of buildings are not frequent in practice, we can use this simplified model with buildings where the effect of internal partitions has little influence on the movement of air, such as small homes or buildings with large interior spaces. Any building where the effect of loss associated to the air through doors or air movement in zones is considered negligible can also be treated using a single model.

Internal conditions in the single model can be considered homogeneous, so there exists only one volume with one pressure and temperature value assigned to a single node.

To solve the network nodes, we firstly remove the internal pressure from the system equation. Once calculated, it is possible to evaluate the flows throw the building envelope (walls and roof), grilles and windows using the following equation:

$$q_v = C \cdot (\Delta P)^n \quad (2)$$

Figure 1 and equation 3 show an example of single-zone model with three external nodes corresponding to a windward façade, leeward façade and roof.

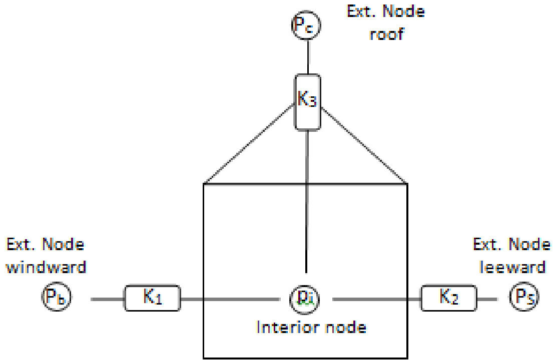


Figure 11: Network nodes in single-zone model

$$tightness\_wall \cdot A_{wall\_w} \cdot \left(\frac{\Delta p_w}{4}\right)^{0.67} + tightness\_window \cdot A_{h\_w} \cdot \left(\frac{\Delta p_w}{100}\right)^{0.67} + C_{grill\_w} \cdot \left(\frac{\Delta p_w}{20}\right)^{0.5} + \\ + tightness\_wall \cdot A_{op\_l} \cdot \left(\frac{\Delta p_l}{4}\right)^{0.67} + tightness\_window \cdot A_{h\_l} \cdot \left(\frac{\Delta p_l}{100}\right)^{0.67} + C_{grill\_l} \cdot \left(\frac{\Delta p_l}{20}\right)^{0.5} + q_{ex} = 0 \quad (3)$$

where,

$\Delta p_w = p_w - p_{\text{int}}$ and $p_w = 0.5 \cdot C_p\big|_w \cdot \rho_0 \cdot v^2$

$\Delta p_l = p_l - p_{\text{int}}$ and $p_l = 0.5 \cdot C_p\big|_l \cdot \rho_0 \cdot v^2$

- $tightness\_wall$ and $tightness\_window$ are the permeability of the walls and windows at 4Pa and 100Pa respectively, in m^3/hm^2,
- C_{grill} is the grille coefficient, in m^3/h,
- q_{ex} is the air flow extraction, in m^3/h,
- C_p is the pressure coefficient,
- ρ_0 is the air density, in kg/m^3,
- v is the air velocity, m/s.

The subscript w means windward and l means leeward. The equation 3 has only one unknown p_{int}, so once calculated, it is possible to know the air flow for each element.

3.2 Multizone model

The multizone model is a more complex and accurate model which solves the equilibrium equation for each zone. The equation to calculate the indoor pressure in each room has the same form as equation (3), but one per room is needed. The following equation shows how to calculate the flow between two zones through a grille:

$$q_{i \to j} = C_{ij} \cdot (p_i - p_j)^{0.5} \quad (4)$$

Where:

- C_{ij} is the air flow that goes through the grille when the pressure difference is 1 Pa.

- P_i is the pressure in the ith zone.
- $Q_{i\rightarrow j}$ is the air flow between two zones through the grille.

Although there are several tools to calculate the infiltration rate and CO_2 distribution such as CONTAM [3] or TRNSYS [4], the Department of Energy in the University of Seville has developed its own tool called VENTItool with the aim of being able to integrate this software as an additional capacity into the official energy performance certificate software.

4 RELATION BETWEEN IAQ-VENTILATION RATE-CONSUMPTION

The qualitative relation between IAQ and ventilation rate is predicted in [2, 5] and shown in figure 2. Note that the higher the ventilation rate, the lower the pollutant concentration.

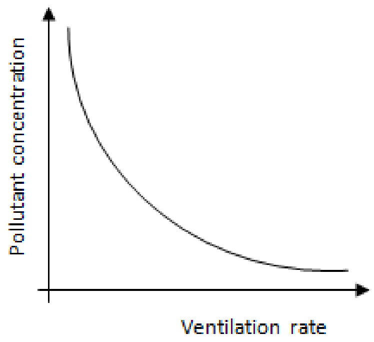


Figure 12: Qualitative relation between IAQ and Ventilation rate

Although this relation seems to be obvious, this curve could have a different behaviour in real buildings where infiltrations can occur. This is what may happen when wind speed rises and n50 has a high value. Figure 3 shows schematically the difference between the ideal situation where the whole building is under pressure and air goes in due to mechanical ventilation (infiltrations are null), and real situation where air goes into the building due to mechanical ventilation and infiltrations, with zones in overpressure where air moves out.

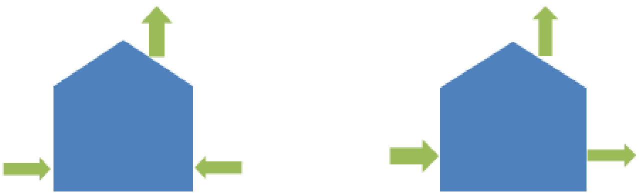


Figure 13: Ideal situation (left) and real situation (right).

The air flow extraction (ventilation flow) which makes the situation ideal can be determined drawing ventilation rate (ACH_{vent}) against ventilation plus infiltration rate ($ACH_{vent+inf}$). Figure 4 shows this relation, where ventilation rates higher than 0.72 (vertical blue line) provokes the ideal situation.
The value of ventilation rate in the ideal situation depends on the airtightness, wind speed, wind direction and the model used to calculate air flows. This graph has been calculated using a single zone model considering half façade windward and half façade leeward for simplicity, though we will discuss in subsequent sections the influence of using different wind directions and a multizone model instead of the simplified one.

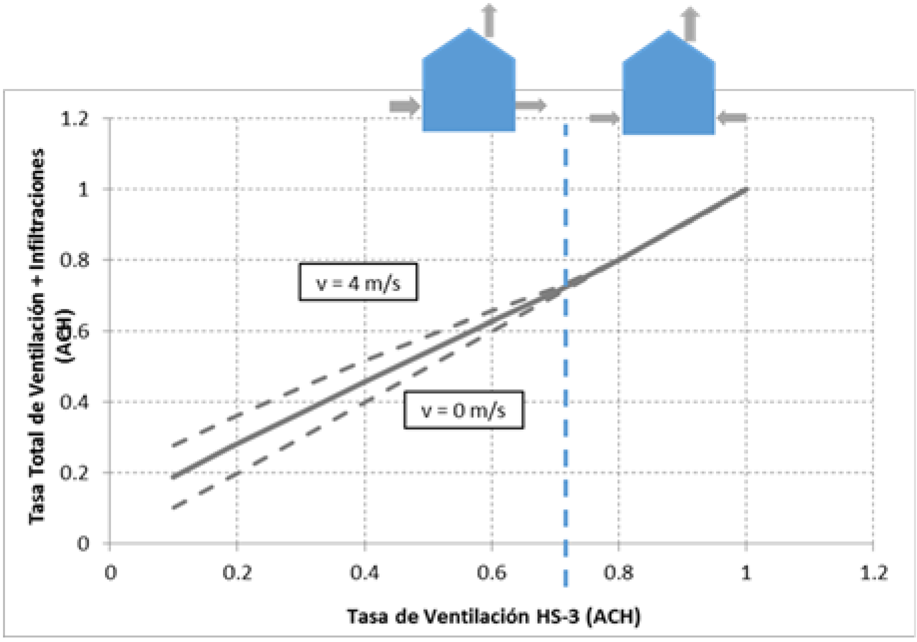


Figure 14: Relation between ventilation + infiltration rate and ventilation rate

Finally the demand of energy rises when ventilation and infiltration rate increases. Ventilation and infiltration loads are directly proportional to the air flow that goes into the building, thus the evolution is linear as shown in figure 5:

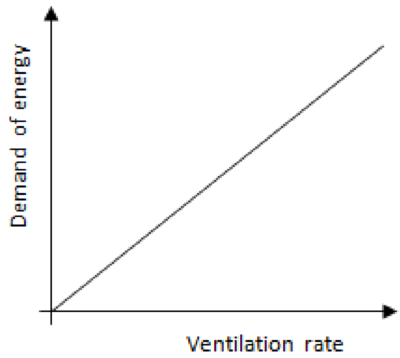


Figure 15: Qualitative relation between demand of energy and ventilation rate

5 DEFINITION OF STUDIED BUILDING

The building used to perform the simulations is the same as that presented by the Spanish Regulation in its document called HS-3. This building consists of 3 bedrooms, a living room, a kitchen and a bathroom. All these spaces are connected by a corridor as shown in figure 6:

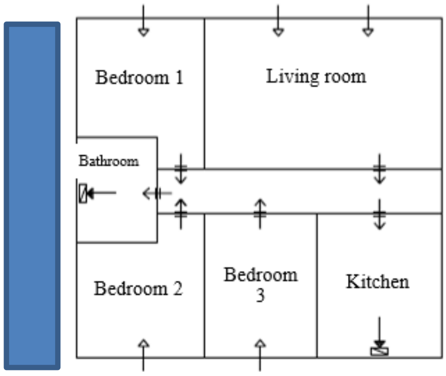


Figure 16: Building used to perform the simulations

The features of the building’s construction are indicated in the table below:

Table 3: Features of the building's construction

	Living room	**Bedroom 1**	**Bedroom 2**	**Bedroom 3**	**Kitchen**	**Bathroom**	**Corridor**
Number of facades	2	1	1	1	2	0	0
Volume (m^3)	86.4	43.2	43.2	43.2	43.2	30	34.8
Wall area (m^2)	26.9	9.3	9.3	9.3	20.6	0	0
Window area (m^2)	5.5	1.5	1.5	1.5	1	0	0

An important feature that has not been mentioned above is the airtightness coefficient. This property will be variable in order to perform the study with different values of n_{50}, as explained in the following section.
The connections between rooms and corridor have been modelled as a closed door, whose permeability coefficient is 195 m^3/h.
Mechanical ventilation will be carried out by extractors located in the kitchen and bathroom. The flow of each extractor will be variable in order to be able to do a parametric analysis, as shown in section 6.

5.1 Occupancy schedule

In order to define the pollutant concentration in each room, it is needed to create an occupancy schedule. There is one schedule for each room, distinguishing between working days and holidays, as shown in table 2. Note that the occupation of bedrooms 2 and 3 is the same, while the occupation of the corridor has been neglected.

Table 2: Occupancy schedule for working days (left) and holidays (right)

Time	Livingroom	Bedroom 1	Bedroom 2/3	Kitchen	Bathroom
00:00	0	2	1	0	0
01:00	0	2	1	0	0
02:00	0	2	1	0	0
03:00	0	2	1	0	0
04:00	0	2	1	0	0
05:00	0	2	1	0	0
06:00	0	2	1	0	0
07:00	0	2	1	0	0
08:00	0	0	0	3	1
09:00	0	0	0	0	0
10:00	0	0	0	0	0
11:00	0	0	0	0	0
12:00	0	0	0	0	0
13:00	0	0	0	0	0
14:00	0	0	0	0	0
15:00	0	0	0	0	0
16:00	0	0	0	0	0
17:00	3	0	0	0	0
18:00	3	0	0	0	0
19:00	1	0	1	0	0
20:00	0	0	1	1	1
21:00	3	0	0	0	0
22:00	4	0	0	0	0
23:00	4	0	0	0	0

Time	Livingroom	Bedroom 1	Bedroom 2/3	Kitchen	Bathroom
00:00	0	2	1	1	0
01:00	0	2	1	1	0
02:00	0	2	1	1	0
03:00	0	2	1	1	0
04:00	0	2	1	1	0
05:00	0	2	1	1	0
06:00	0	2	1	1	0
07:00	0	2	1	1	0
08:00	0	0	0	0	3
09:00	4	0	0	0	0
10:00	0	0	0	0	0
11:00	0	0	0	0	0
12:00	4	0	0	0	0
13:00	2	0	0	0	2
14:00	4	0	0	0	0
15:00	4	0	0	0	0
16:00	4	0	0	0	0
17:00	4	0	0	0	0
18:00	4	0	0	0	0
19:00	0	0	0	0	0
20:00	0	0	0	0	0
21:00	1	0	0	0	2
22:00	4	0	0	0	0
23:00	4	0	0	0	0

5.2 Wind direction

In order to know how wind direction affects the results, we have simulated the building with two different wind directions. The first of them (figure 7-left) has a 37% of façade on windward and 63% on leeward. The second one (figure 7-right) has a 63% of façade on windward and 37% on leeward.
The value chosen for wind velocity is 4 m/s. This is the average velocity for buildings located in urban areas in Seville [5]

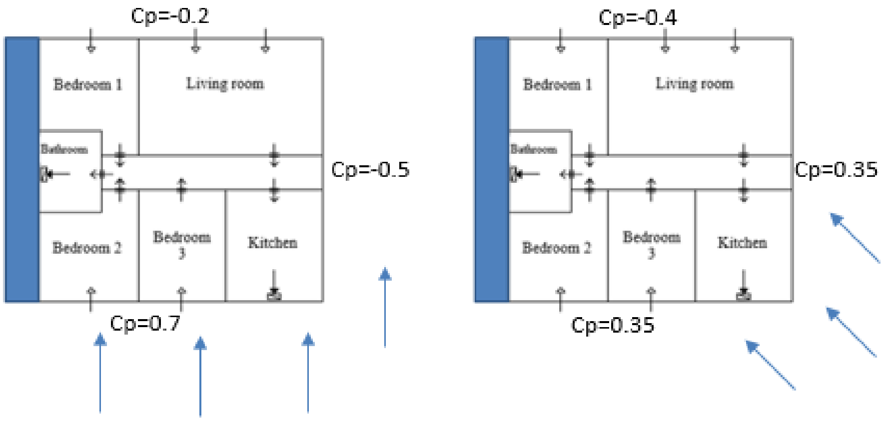


Figure 17: Building with 37% of facade on windward (left) and 63% on windward (right)

The pressure coefficients for each façade are indicated above, and are used to calculate the outdoor pressure as follows:

$$P_{outdoor} = \frac{1}{2}\rho \cdot c_p \cdot v^2$$

Where ρ is the density of air, c_p is the pressure coefficient and v is the velocity.

6 RESULTS

This section is divided into three parts. The first one presents the relation between mechanical ventilation and infiltrations, and how this relation varies with airtightness, speed direction and the ventilation model. In the second part the evolution of CO_2 concentration when mechanical ventilation rate varies is shown for both ventilation models explained in section 3. Finally it is shown how the mechanical ventilation affects the demand of energy in the studied building.

6.1 Relation between mechanical ventilation and infiltrations

The Spanish regulation establishes that it is required to use mechanical ventilation to maintain an acceptable IAQ, but when wind is blowing infiltrations can appear. Avoiding infiltrations is essential to guarantee that in a multizone scenario all rooms have a good air quality.
The next two graphs show the relation between air changes per hour due to mechanical ventilation (ACH_{vent}) and air changes per hour due to mechanical ventilation and infiltration ($ACH_{vent+inf}$) for both scenarios presented in section 4.2 using the single-zone model:

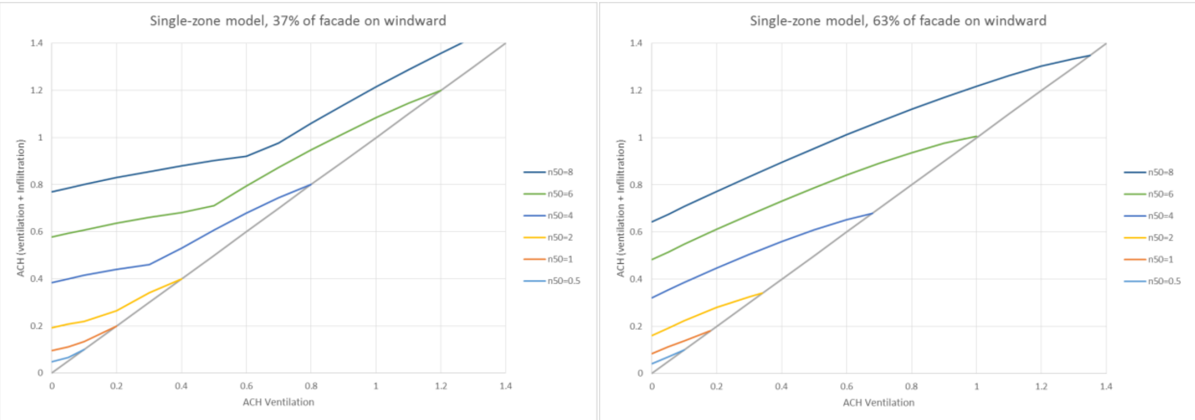


Figure 18: Variation of ACHvent+inf with ACHvent and airtightness using the single-zone model with 37% of façade on windward (left) and 63% of façade on windward (right)

The following observations can be derived from the graphs:

- The higher the air change rate n50, the more infiltrations there are. Thus, the quality of the façade plays an important role in reducing infiltrations.
- The curves presented in figure 8-left change their curvatures for a certain value of ACH_{vent} due to the change in the direction of air flow in a façade. Figure 9 presents what is happening to air flow for an air change rate n50 equal to 6 h^{-1}.
- The 45º straight line in grey represents the situation when wind velocity is considered zero, thus the ACH_{vent} is the same as $ACH_{vent+inf}$ (room is under-pressure). Therefore, there exist infiltrations before the curve cuts the straight line. However, after that the behaviour is similar to the 45º straight line, where there are no infiltrations.
- Curves presented in figure 9-right have the same curvature because there is no change in the direction of flow before reaching the 45º straight line. After cutting it the room will be under-pressure and there will be no infiltrations.

- For any n50, the higher the percentage of windward façade, the lower the ACH_{vent} in which there are not infiltrations (the cut with 45º straight line). This behaviour can be guaranteed for a percentage of windward façade between 30%-70%.

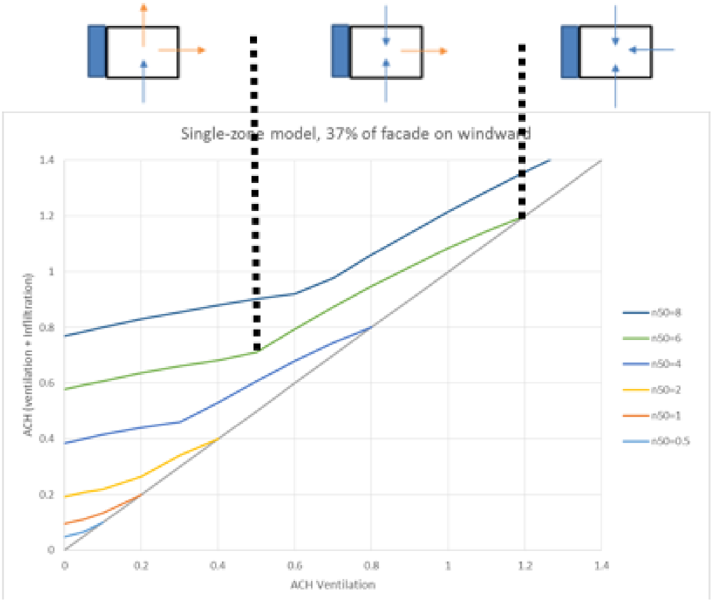


Figure 19: Variation of air flow direction in each facade with ACHvent for n50 equal to 6

The following graphs show the relation between air changes per hour due to mechanical ventilation (ACH_{vent}) and air changes per hour due to mechanical ventilation and infiltration ($ACH_{vent+inf}$) using the Multizone model:

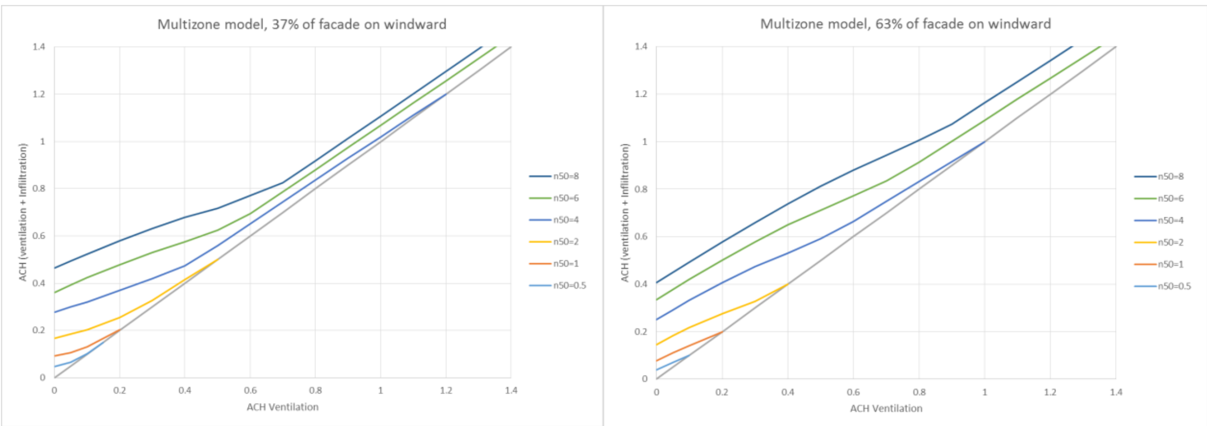


Figure 20: Variation of ACHvent+inf with ACHvent and airtightness using the multizone model with 37% of façade on windward (left) and 63% of façade on windward (right)

The observations derived from the graphs in the multizone scenario are similar to those in the single-zone model, and they are summarized as follows:

- The higher the air change rate n50, the more infiltrations there are.
- The curves presented in figure 10 change their curvatures for certain values of ACH_{vent} due to the change in the direction of air flow in a façade or connection. Figure 11 presents what is happening to air flow for an air change rate n50 equal to 4 h^{-1}.

- The 45º straight line in grey represents the situation when wind velocity is considered zero, thus the ACH_{vent} is the same as $ACH_{vent+inf}$ (all rooms are under-pressure). Thus there exist infiltrations before the curve cuts the straight line, however after that the behaviour is similar to the 45º straight line where there are no infiltrations.
- For any n50, the higher the percentage of windward façade, the lower the ACH_{vent} in which there are not infiltrations (the cut with 45º straight line). This behaviour can be guaranteed for a percentage of windward façade between 30%-70%.
- For n50 higher than 5, note how the curve has an asymptotic behavior. This occurs because there is a room with two façades with a different pressure coefficient when the connection door is closed with a low n50 coefficient, so the air flow goes in and out. The following figure shows what is happening in the living room.

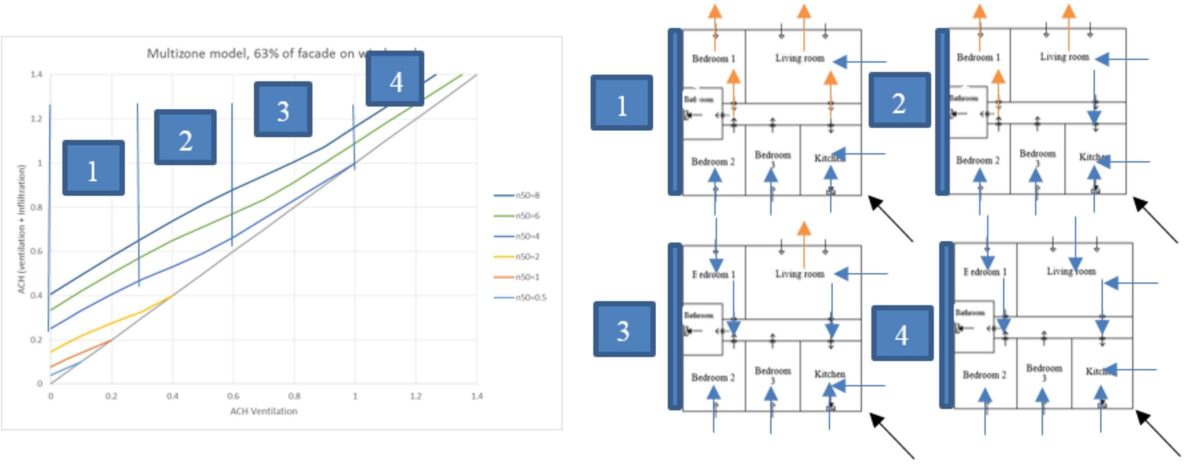


Figure 21: Variation of air flow direction in each facade with ACHvent for n50 equal to 4

As said in the previous section, the single-zone model might differ from the multizone model. Figure 12 illustrates both models superposed for both scenarios presented before:

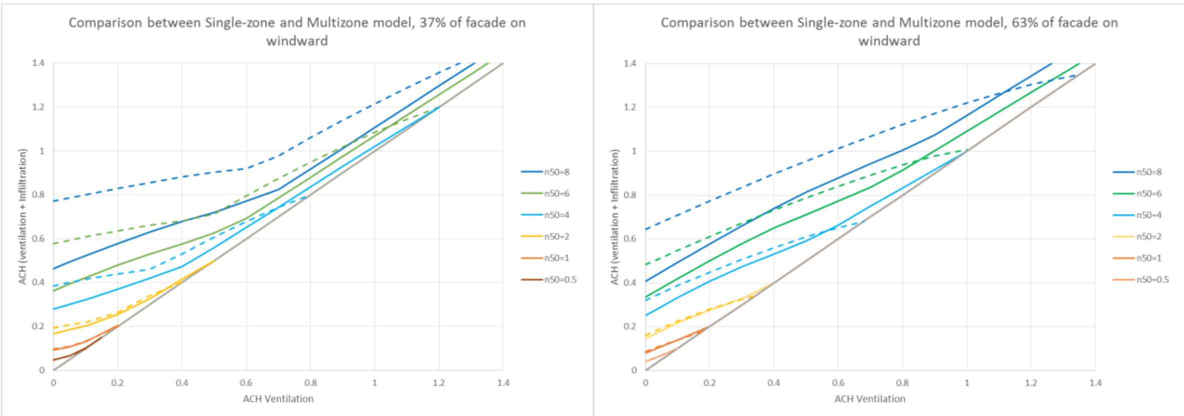


Figure 22: Comparison between single and multizone model for 37% of facade on windward (left) and 63% of façade on windward (right)

For any n50 the curves in single-zone model cuts the 45º straight line (no infiltrations) for an ACH_{vent} value lower than that when using a multizone model. So if we select an n50 using a

single zone model in order to avoid infiltrations and guarantee an acceptable IAQ, the single scenario might not be on the safe side.
For a certain value of n50, there is a value of ACH_{vent} where $ACH_{vent+inf}$ coincide for the single and multizone model. So energy-wise, the demand of energy estimated using the single model will be on the safe side for an ACH_{vent} lower than the cut point between both models.
Finally when n50 is sufficiently low (less than 2), both models estimate nearly the same infiltrations. Table 3 shows the maximum error made using single-zone model instead of the multizone one for each n50. This error is reached when mechanical ventilation is zero:

Table 4: Maximum error made using a single model instead of a multizone one for 37% and 63% of façade on windward

n50 (h^{-1})	**Error 37% windward (%)**	**Error 63% windward (%)**
0.5	0.00	0.00
1	4.34	3.52
2	15.57	11.03
4	38.49	27.89
6	59.67	44.31
8	66.16	58.13

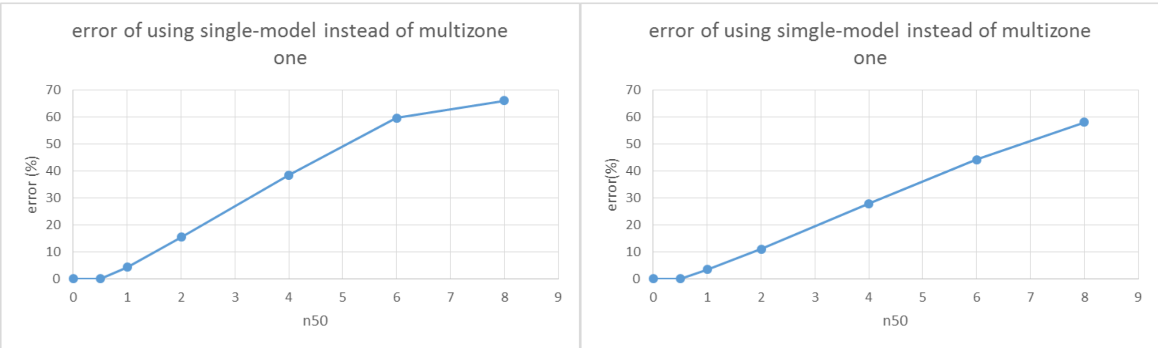


Figure 23: Evolution of maximum error made using a single model instead of a multizone one for 37% (left) and 63% (right) of façade on windward

6.2 Relation between mechanical ventilation and IAQ

The aim of this section is to show the problems due to infiltrations to maintain acceptable levels of pollutant in all rooms when using a multizone model. Values of ACH_{vent} near a change of the curvature in curves such as figure 11 might be critical points where the CO_2 concentration is high (there might not be air intake in certain room). This is what is known as an asymptotical behaviour of the pollutant when an occupied room is not being ventilated. This asymptotical behaviour cannot be perceived using a single model, as shown in figure 14-blue for n50=4 and 63% of façade on windward. However using a multizone model it is easy to detect which rooms have problems with IAQ, and the value of ACH_{vent} for which this problem arises. Figure 14-red shows the average CO_2 concentration using a multizone model for a building with n50=4 and 63% of façade on windward.

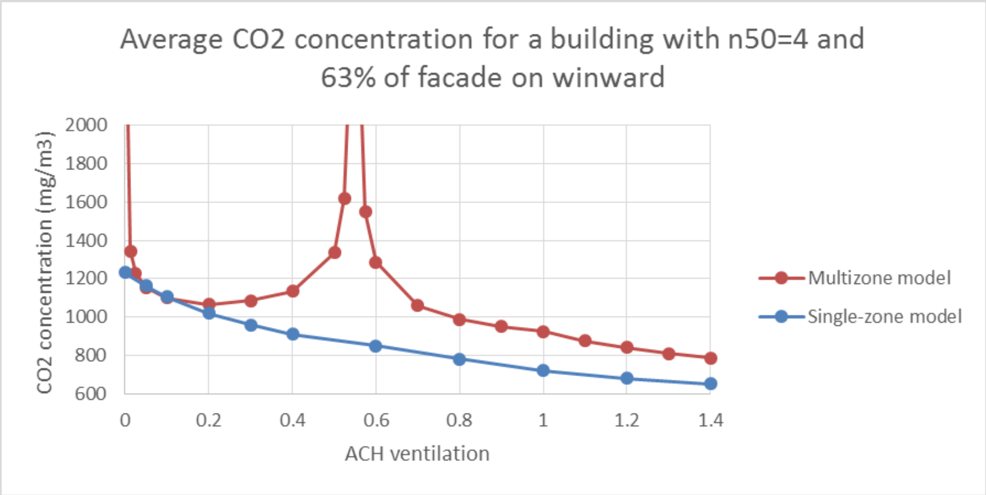


Figure 24: Average CO2 concentration for a building with n50=4 and 63% of facade on windward

The CO_2 concentration using a single model is lower than that when using the multizone one due to the higher infiltration estimated by a single model. This CO_2 difference between models decreases when ACH_{vent} increases, in particular when infiltrations start being zero. According to figure 14-red, there are two critical points where the concentration grows rapidly. First of them occurs when there is no mechanical ventilation in the bathroom because there are no connections between the bathroom and outside. Figure 15-left shows the evolution of average and maximum CO_2 concentration in the bathroom. The second critical point is reached when ACH_{vent} is nearly 0.55, and this is due to the low air intake in bedroom1 as shown in figure 11 from step 2 to 3. Figure 15-right shows the evolution of average and maximum CO_2 concentration in bedroom 1.

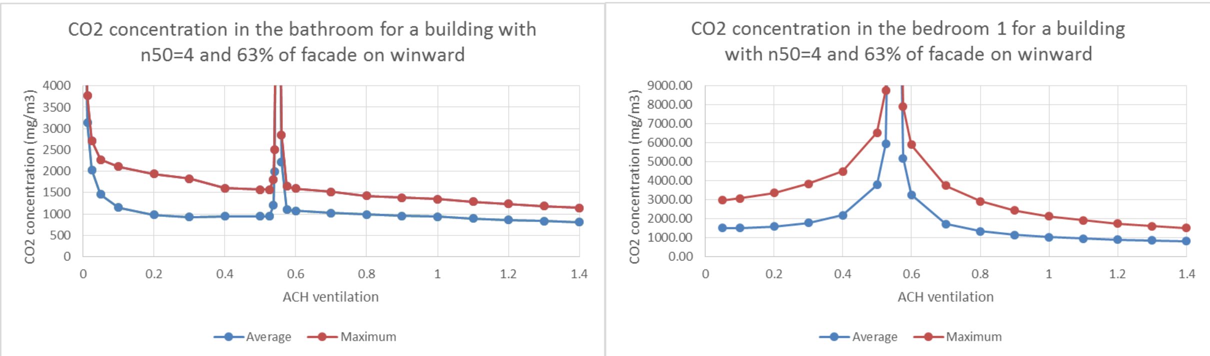


Figure 25: Average (blue) and maximum (red) CO2 concentration in the bathroom (left) and bedroom1 (right) for a building with n50=4 and 63% of façade on windward

For this particular case, there are no problems in IAQ when infiltrations start being nil (ACH_{vent}=1), but this point must be considered as critical because there might be scenarios where the CO_2 concentration could be high due to low air intake.

6.3 Relation between mechanical ventilation and demand of energy

The demand reduction when moving from the single-zone to the multizone model depends on the overall permeability of the building, its geometry and the surface exposed to the wind. The examples developed show that below the cutting point the demand reduction when moving from the single-zone to the multizone model is 37.5 % when n50=8ACH and 30 %

when n50=4ACH, both cases without mechanical extraction. As the mechanical ventilation increases, those values are reduced. For example when there is an extraction of 0.4 ACH, the results become 22 % (n50=8 ACH) and 18 % (n50=4ACH).
As we have seen, the values when n50<=2 ACH are not affected.
Likewise, above the cutting point the calculated demands with the single-zone model may not be conservative enough with respect to those of the multizone model. That cutting point is located on a variable range and thus it is necessary to evaluate it in each individual case. It has been made clear that the mechanical extraction flows implemented in countries of the European Community may be at around that range [6, 7].

7 CONCLUSIONS

The main conclusions drawn from the paper are as follows:

- Single-zone and multizone models provide different values for infiltrations. Moreover, the estimated value of ACH_{vent} that avoids infiltrations in the single-zone model is lower than that of the multizone model.
- Wind direction and airtightness affects considerably the infiltrations, so defining these variables accurately is important in order to obtain the real behavior of the dwelling. However, it is quite difficult to find a reliable database with wind direction and velocity and to measure airtightness precisely.
- Having infiltrations in a dwelling leads to an asymptotic behavior of CO_2 in zones with leeward façade. Using the multizone model, one can estimate the value of ACH_{vent} that causes this problem and identify the problematic zone. This cannot be predicted using a single-model model.
- n50 lower than two allows us to use the single-zone model without having significant errors. However, this airtightness is quite far from that in real dwellings in Spain.
- When the n50 is higher than 6, the value of ACH_{vent} that avoids infiltrations is increased (much higher than 1.4 ACH).
- Calculating the energy demand based on the resulting flows of the single-zone model is more conservative than doing so through the multizone model. This applies on a general basis when the permeability is high (typically n50>2 ACH) and the extraction flows are below 0.8 ACH. However, those values depend on the geometry of the building and its exposed surface, just as discussed in section 6.1.

8 REFERENCES

[1] International Energy Agency (IEA).
[2] Liddament, M.W., AIVC, Guide to energy efficiency ventilation, Conventry, ISBN 0 946075 85 9, 1996
[3] George N. Watson. National Institute of Standard and Technology. *CONTAM User Guide and Program Documentation.*
[4] The University of Wisconsin (http://sel.me.wisc.edu/trnsys/trnlib/library15.htm)
[5] Allard F. et al, Natural ventilation in buildings, James & James, 1998, London
[6] N. Brelih, O. Sepänen, "Ventilation rates and IAQ in European Standards and national regulations", 32nd AIVC Conference and 1st Tightvent Conference, Brussels, October 2011.
[7] Salmerón Lissén JM, Sánchez de la Flor FJ, Álvarez S, and Molina JL "Analysis and Repercussions of the Updating of the Spanish Regulation Regarding Ventilation and Infiltration", International Journal of Ventilation, Vol. 11 No.4, March 2013.

RADON CONCENTRATION CONTROL BY VENTILATION, AND ENERGY EFFICIENCY IMPROVEMENT

Borja Frutos[*1], Manuel Olaya[1], Fernando Martín-Consuegra[1], Carmen Alonso[1]

1 Institute for Building Science Eduardo Torroja – CSIC
Serrano Galvache 4
Madrid, Spain
**Corresponding author: Borjafv@ietcc.csic.es*
Presenting author: Borja Frutos

ABSTRACT

Radon gas is a pathological agent confirmed by World Health Organization in terms of increasing the risk of lung cancer generation when it is inhaled by human in high concentration. This gas comes from soils with uranium content (i.e. granite terrain) and penetrates through the building envelope, such, as floors or basement walls. Its accumulation in indoor spaces increases the radon concentration level, constituting a health problem for occupants. This can be handled by rehabilitation actions in buildings that reduce indoor concentration to acceptable levels.

Ventilation technique can contribute for the same purpose. The gas dilution, by exchanging indoor and outdoor air is an effective mechanism. A priori, this easy to implement and low cost alternative can be recommended when other architectonic options are not feasible. However, the aspects that can compromise energy efficiency in ventilated spaces should also be evaluated. In order to reduce radon concentration from 900 Bq/m3 to 200 Bq/m3, an additional 30000 kWh/year can be needed for achieving indoor thermal comfort. This is showed in some cases where the air change rate required to reduce this concentration is 5ac/h,

This paper presents an approach for understanding radon mitigation by ventilation, the energy lost in the process, and how to reduce such loss by implementing energy efficiency measures.

KEYWORDS

Radon mitigation; Ventilation; Heat recovery, Energy efficiency.

1 INTRODUCTION

Radon gas (hereinafter referred to isotope of radon, Rn-222) is a radioactive element that is generated in areas with high radio content in soils (granitic terrain for example).
Its high degree of mobility allows it to penetrate from the ground into buildings through the enclosure (porosity of materials, fissures, cracks and joints) and to accumulate inside, where it can be inhaled in high concentrations. Figure 1 shows a schematic of the usual ways of radon entry into buildings.

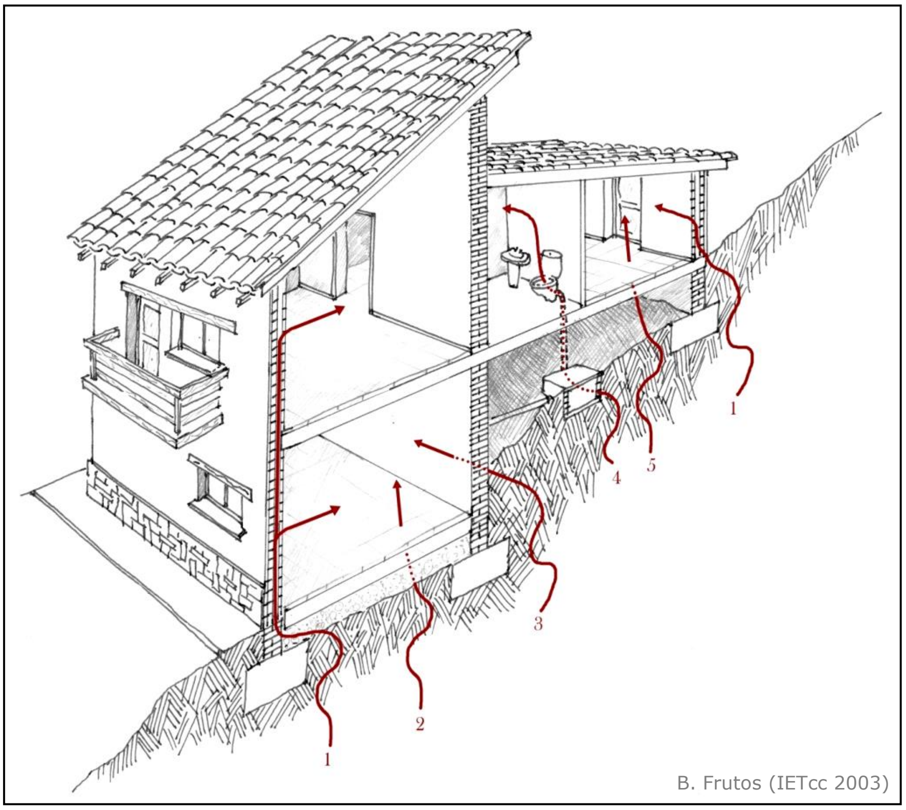


Figure 1:Common ways of radon entry into buildings. (1-Cavity walls, 2 - Through the floor slab, 3 - Through the basement walls, 4 - Through sanitation pipes, 5 - Through the floor slab)

In Europe, maximum concentration levels permitted in dwellings (*2013/59/EURATOM)* are expressed both for existing and new buildings:

300 Bq/m^3 for existing Dwelling. Level of action.
200 Bq/m^3 for new dwelling. Design Level.

With the aim of reducing the concentration levels to this reference benchmark, some architectural remediation options can be proposed. The intervention design depends on the initial concentration, the effectiveness required to reduce levels below recommended limits, and building configuration. They can be classified in two basic strategies (figure 2):

a) Barrier systems: Those that base their effectiveness in providing greater tightness of the building envelope in contact with the ground.

b) Extraction systems: Those that evacuate the gas from the ground, reducing its entry into the building.

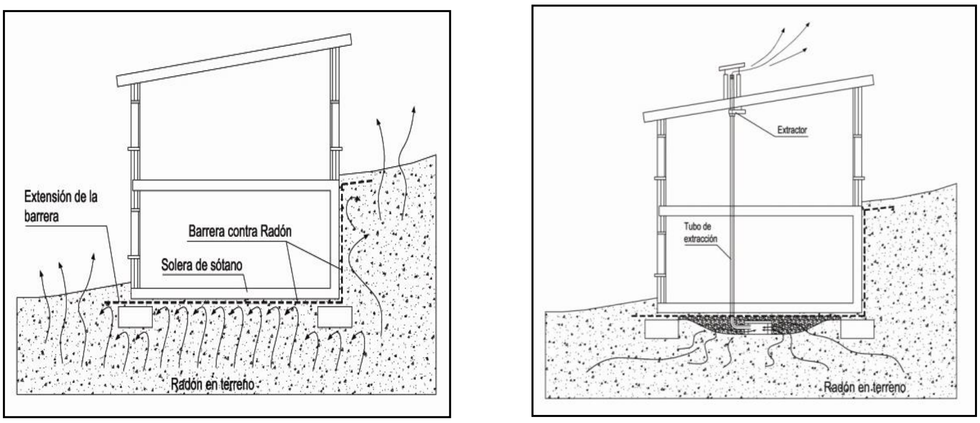


Figure 2: Examples of barrier systems (a) and extraction system (b)

Some of them have been tested in a housing prototype building placed in a high radon area in Spain, with a good effectiveness classification (Frutos, 2011).

Another type of action can be taken into account:

c) Natural or forced ventilation of indoor living spaces for diluting radon concentration. These are attractive measures as they do not involve excessive cost of execution and cause low constructive impact. This strategy has to be deeply studied for its optimization.

This paper will show relevant aspects around this ventilation protection strategy and its relation to energy efficiency.

2 VENTILATION AS A RADON CONTROL TECHNIQUE.

Dilution of the gas exchanging indoor and outdoor air (not usually exceed 20 Bq/m^3) will reduce radon concentration inside. Depending on the ventilation rate needed, this can be achieved naturally or forced with a fan (figure 3).

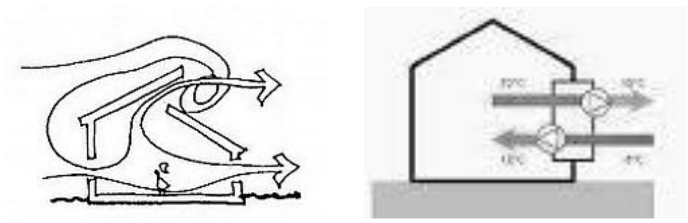


Figure 3:Ventilation strategies: natural and forced.

On the other hand, this action will produce a change in the state of internal pressures, that can modify the entry gas flow rates from the soil. When air is supplied from the outside, internal pressure usually increases, and the flow rate of radon entry gets lower than when air exchange is produced by extraction (figure 4).

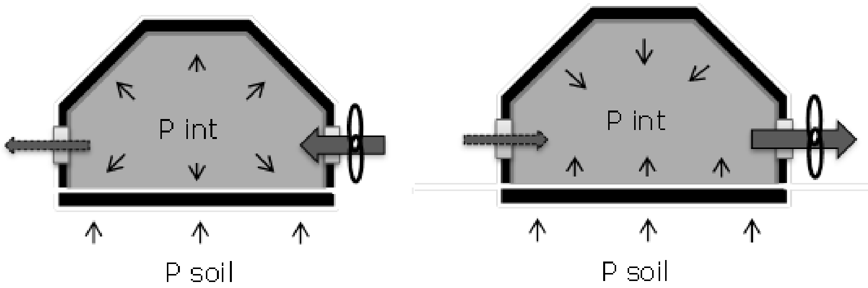


Figure 4: Supply or extract air.

However, there are some factors that should be considered before undertaking rehabilitation against radon by using this strategy, such as:

- Infiltration rates.
- Air change rate needed for reducing radon concentration
- Impact on energy efficiency.

This communication addresses, from a technical point of view, the different variants of this type of action, and the study of associated energy cost.

3 INFILTRATION RATES OF THE BUILDING.

The airtightness of a building, understood as the permeability of the envelope, can beobtained by the blow door test (EN 13829:2002).It plays an important role in outside air exchange. It is estimated that homes before 1950 may have a permeability rate of 1.5 h^{-1} (1.5 changes of the whole air volume per hour are produced). In modern homes, the airtightness is estimated at 0.25 h^{-1}, and lower, according to energy efficiency European directives(DIRECTIVE 2012/27/EU). This value depends primarily on windows permeability, building envelopes, fissures and cracks.

As shown in some studies (Lembrechts 2001) the tightness levels of current building have increased in part due to the improvements in joinery windows and doors, and sealed with waterproof sheets. This also involves an increase in indoor radon levels. See Figure 5.

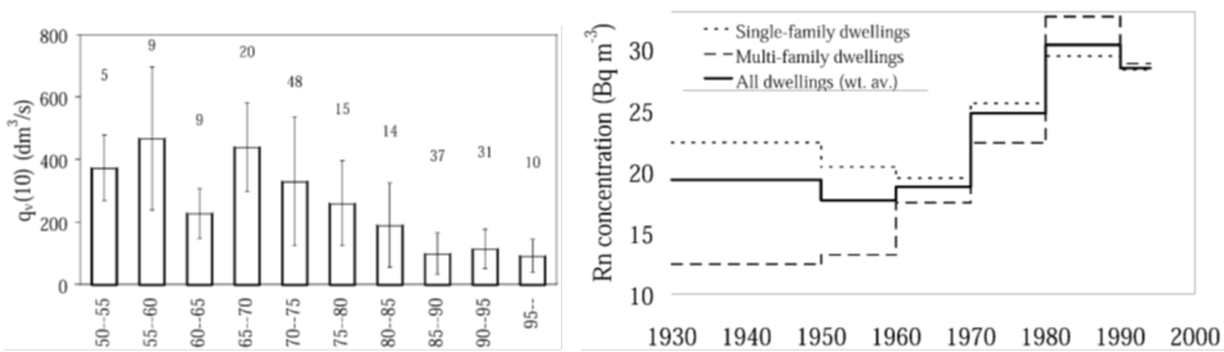


Figure 5:Evolution of infiltration rates and their impact on indoor radon levels. (Lembrechts 2001)

To determine the degree of affectation that permeability of a house has in the final radon concentration, next expression can be used for simulating.

$$C=R/V.\lambda t \quad (1)$$

where:
R: Exhalation rate Bq/h
λt *(h^{-1}) (λd decay radon constant (0,00756)+ λh air tightness)*
V Volume (m^3)

This estimation is useful for knowing the impact of energy rehabilitation measures reducing airtightness of a building, in indoor radon concentration. Next figure 6 shows the variation in indoor radon concentration due to variations in the permeability of the envelope, for the same starting condition and same radon entry rate into the building. Indoor radon concentration goes from 200 $Bq.m^{-3}$ in an old building with a $1.5h^{-1}$permeability rate to 1000 $Bq.m^{-3}$ in a new one with a 0.25 h^{-1}permeability rate.

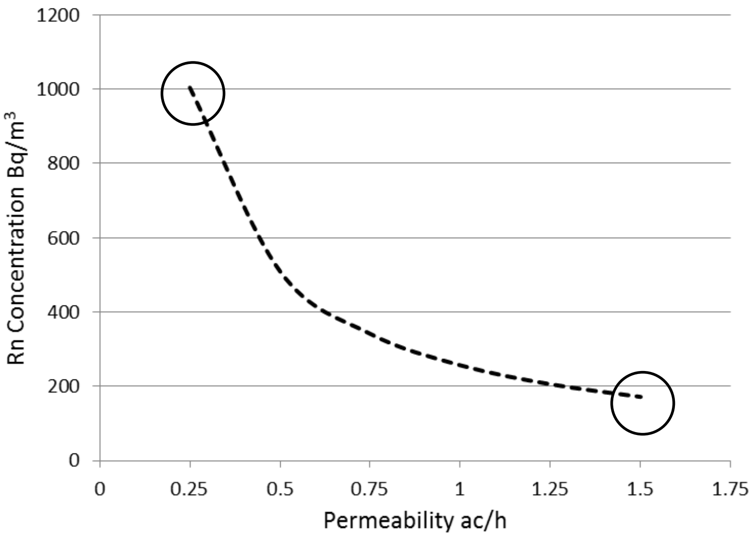


Figure 6: Permeability / Rn concentration

4 AIR CHANGE RATE NEEDED FOR REDUCING RADON CONCENTRATION. A CASE STUDY

An example of a building prototype has been studied on the capacity of the ventilation technique in reducing radon. Its main characteristics are:

- Building: 3 floors, 690 m^3 air volume, heavy construction with thermal transmittance $U=0.35\ W/m^2K$

Air change rates for reducing radon concentration, can be achieved by exchanging outdoor air with a lower concentration. It can be simulated using next expression:

$$C=R/V.\lambda_{total} \quad (2)$$

where:
λ_{total} *(h^{-1}): (λ_d disintegration const. + λ_h initial air tightness + λ_r air change rate /h)*
The following studies have been accomplished: Study 1 with four different cases of initial radon concentration, while maintaining the rate of initial airtightness (1 ac//h), and Study 2

where airtightness is varying while maintaining the initial radon concentration (900 Bq/m^3).Results are shown in figure 7.

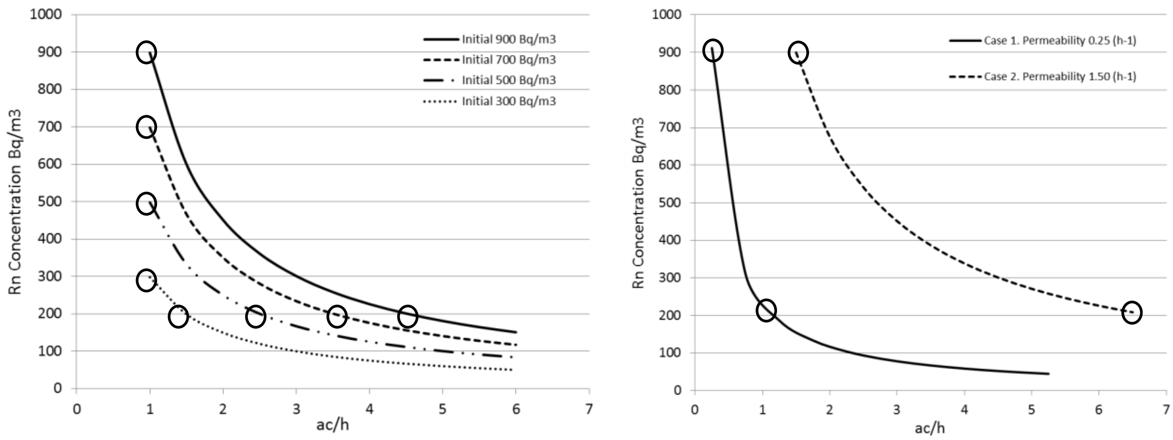


Figure 7: Study 1 (Different initial permeability) and Study 2 (Different initial concentration)

Taking into account 200 Bq/m^3 as the goal of effectiveness, next aspects can be discussed:

In study 1 the air change rate needed increases from 1.5ac/h to 4.5ac/h according to the initial radon concentration from 300 Bq/m^3 to 900 Bq/m^3.

In study 2 the air change rate needed for reducing radon concentration depends on initial air tightness. It can be seen that buildings with less initial permeability need less exchange air for reducing radon concentration than those which more permeability. Note that for the reaching the same initial radon concentration, with more permeability condition (from 0.25 to 1.5), more radon entry rate is considered.

Finally, for defining the air exchange rate as a method for reducing radon, next study is carried out in a standard house with these initial conditions:

- Building: 3 floors, 690 m^3 air volume, heavy construction (U=0.35 W/m^{2}K)
- Radon concentration: The hypothetical radon concentration in ground floor has been set in 900 Bq/m^3
- Ground: the exhalation radon rate from soil into the building: 175Bq/s.
- Airtightness rate of the building: 1 h-^{1}

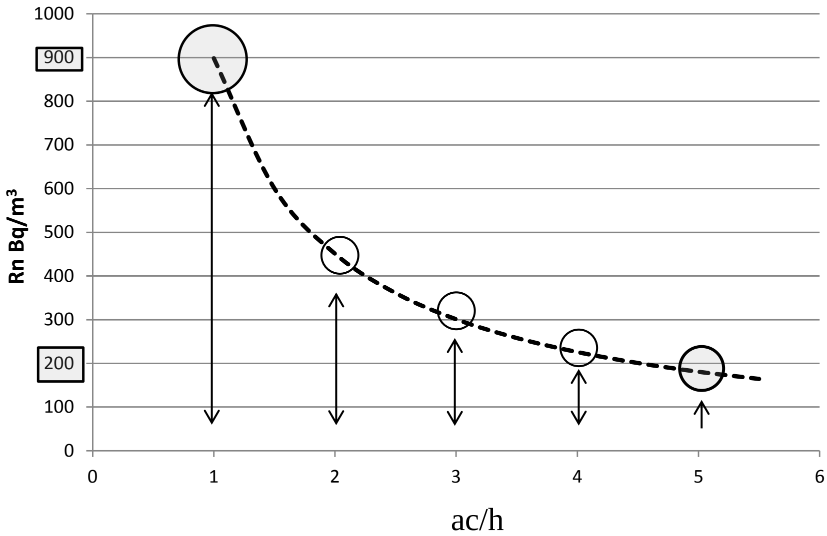


Figure 8: Air change rate vs. radon concentration.

As shown in figure 8, five air changes per hour are needed to reach 200 Bq/m^3. Notice that just dilution is considered and not other effects as pressure state changes.

This high rate can only be reached by forced ventilation in most cases (Cavallo, 1996), and so:
- Forced ventilation implies energy consumption in fans
- High air change rate implies losses of thermal comfort in many types of climates that has to be supported with heating or cooling systems.

5 IMPACT ON ENERGY EFFICIENCY.

Next energy efficiency analysis is made with the prototype house showed in the previous section, taking into account the energy needed for fans, heating and cooling when changing from 1 to 5 air change rate.

The study has been carried out using thermal performance software (Design Builder, V.4.2.0.034). The model is located in Madrid, Spain, as the reference climate. (range of temperatures: -4ºC; 36ºC).

Next figure 9 shows the energy consumption, in the whole year period, when five air changes per hour is implemented. Note that in colder months, heating is needed for maintaining temperatures in a comfort standard of 20 ºC, and in hotter months, cooling is necessary for 25 ºC standard.

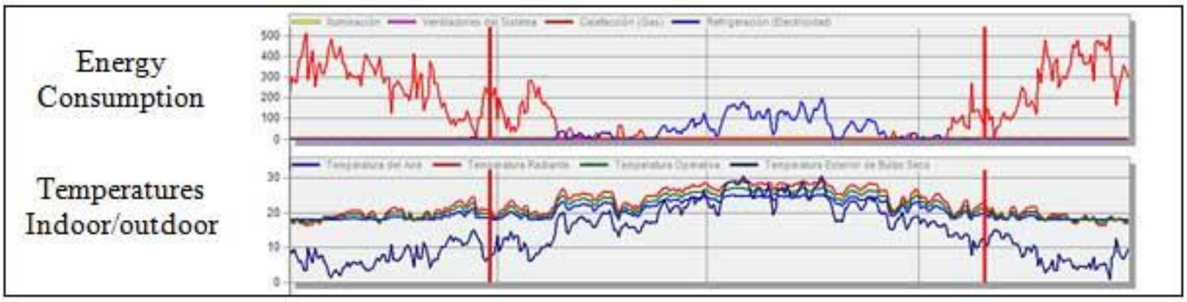


Figure 9: Daily energy consumption for maintaining comfort condition in one year period.

The HVAC annual balance simulation, changing from 1 to 5 air changes, increases the heating loads and slightly lowers cooling loads as shown in figure 10:

Figure 10: Energy consumption (kWh/year) simulated for one year in Madrid with 1 and 5 air changes per hour . (Red colour: energy for heating; Blue colour: energy for cooling)

To ensure 5 air changes per hour, as continuous ventilation for reducing radon levels, it is necessary to implement a forced ventilation system. For the study of energy consumption associated with such measures it has also been considered electricity consumption of the fans and the ability to install or not a heat recovery. The results are shown in the table below:

Table 1: Summary of energy and radon concentrations in the three case studies.

Case study	**Permeability/ Ventilation (ac/h)**	**Rn concentration (Bq/m^3)**	**Heat recovery**	**Electricity consumption FANS (kWh/year)**	**Energy consumption HEATING (kWh/year)**	**Energy consumption COOLING (kWh/year)**
A. Dwelling without forced ventilation.	1 ac/h	900 Bq/m^3	NO	No fans. Ventilation by natural convection	4879	17994
B. Dwelling with forced ventilation	5 ac/h	200Bq/m^3	NO	3267	38305	14688
C. Dwelling with forced ventilation	5 ac/h	200 Bq/m^3	SI	9636	9461	11398

Taking case A as the base case study, it can be observed the different energy consumption with strategies, B and C, depending on whether or not using of a heat recovery system. In B less electricity for fans is used, but the HVAC system needs more energy, while in C, with more electricity consumption in fans, the HVAC energy need is reduced.

For options B and C, the increase of the total energy from initial case A, is:

Initial. Case A: Total 22873 kWh/year
Remediation. Case B: Δ 33387kWh/year
Remediation. Case C: Δ 7622kWh/year

6 LIMITS OF THE STUDY

- The study has been carried out considering ventilation techniques, acting just by dilution without modifying indoor pressures state. This consideration is important because no modification of gas flow penetration rate from soil has been taken into account.
- Madrid climate has low and high extreme temperatures. In other climates, energy consumption has to be analyzed.
- Heating and cooling systems are relevant in final energy consumption. In this case, heating pump and gas boiler has been selected.
- A 0,37 kW fan, has been selected for moving 5 air changes per hour, without heat recovery.
- Two 0,55kW fans have been implemented in the heat recovery, in case C, for treating 5 air changes per hour.

7 CONCLUSIONS

The paper has shown the relation between permeability, ventilation, energy use, and indoor radon levels in a building. Some aspects show up:

- Airtightness retrofitting, according to energy efficiency standards, can have a relevant impact on indoor radon concentration.
- Ventilation technique can be considered as a solution for reducing radon concentration by dilution, but attention must be paid when high radon levels are measured because of the high air change rate required.
- Energy impact, especially in colder climates, can be excessive in order to maintain indoor comfort temperatures when ventilation strategies are implemented
- This impact can be reduced with a heat recovery system.

8 ACKNOWLEDGEMENTS

The authors wish to express their gratitude to Carlos Sainz from Cantabria University for its contribution on the knowledge of the radon entry mechanism.

9 REFERENCES

Cavallo, A. (1996). *Comparison of natural and forced ventilation for radon mitigation in houses*.Environment International, Vol. 22, Suppl. 1, pp. SlO73-S1078.

COUNCIL DIRECTIVE 2013/59/EURATOM *of 5 December 2013 laying down basic safety standards for protection against the dangers arising from exposure to ionizing radiation.*

DIRECTIVE 2012/27/EU *OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 25 October 2012 on energy efficiency*

EN 13829:2002.*Thermal performance of buildings - Determination of air permeability of buildings - Fan pressurization method.*(ISO 9972:1996, modified)

Frutos, B; Olaya, M; Quindós, L. S.; Sainz, C.; Fuente, I.(2011).*Experimental study of effectiveness of four radon mitigation solutions, based on underground depressurization, tested in prototype housing built in a high radon area in Spain.* Journal of Environmental Radioactivity 102, 378-385.

Lembrechts, J; Janssen, M; Stoop, P. (2001)..*Ventilation and radon transport in Dutch dwellings: computer modelling and field measurements.* The Science of the Total Environment 272, 73-78.

WHO (2009).*Handbook on Indoor Radon.A public health perspective*.World Health Organization.

HOW THE FILTRATION OF THE INCOMING AIR DECREASES THE PARTICLE CONCENTRATION WITHIN A SCHOOL EQUIPPED WITH A BALANCED VENTILATION SYSTEM

Alain Ginestet[*1], Dominique Pugnet[1], and Mirela Robitu[1]

1 CETIAT
Centre Technique des Industries Aérauliques et Thermiques
Domaine scientifique de la Doua, 25 avenue des Arts
69100 Villeurbanne, FRANCE
**Corresponding author: alain.ginestet@cetiat.fr*

ABSTRACT

To study the impact of the filtration efficiency level on the particle concentration in a rural school equipped with a balanced ventilation system with heat recovery, measurements of indoor and outdoor particle concentrations have been carried out by using three different efficiency filters. The tested filters are respectively classed G4, F7 and F9 according to NF EN 779 (2012). Air has been sampled alternatively and continuously inside the fresh air duct at the inlet of the air handling unit, inside the delivered air duct at the outlet of the air handling unit and inside the exhaust air duct. These samples have been analyzed by an optical particle counter for particle concentration measurements in the 0.3 – 10 µm particle size range. Particle concentration values have been used for the filtration efficiency calculation as well as for the calculation of the I/O ratio of the indoors to the outdoors particle concentration. The results show that the efficiency of the G4 filters on the finest particles being in size lower than 1 µm is very low and close to zero. But with the use of the F7 filters and even more with the F9 filters the filtration efficiency is much better, respectively 47 to 55 % with the F7 filters and 82 to 87 % with the F9 filters at 0.4 µm (depending on the day of measurements). When the school is unoccupied, the I/O ratio decreases as the filtration efficiency increases. When the school is occupied, the I/O ratio is the same as when the school is unoccupied considering particles smaller than 0.5 µm. But for larger particles, the I/O ratio is higher and increases when the particle size considered increases and when the filter class decreases. The I/O ratio exceeds 1 depending on the class of the filter and the particle size considered.
In summary, the ventilation system used in the studied school is able to protect people as the I/O ratio is lower than 1 for the most harmful (diameter < 1 µm) and most abundant (99 % in number) particles recorded outdoors when fine filters (F7 or even better F9) are installed within the air handling unit.

KEYWORDS

Ventilation, Air filtration, school, particle concentration, IAQ

1 INTRODUCTION

The European Commission claims that buildings consumed 41 % of the final energy in Europe in 2010 (European Commission, 2013). Moreover, the average energy consumption of the building sector (220 kWh/m^2 in 2009) has increased by around 1 % per year since 1990, with residential buildings (around 200 kWh/m^2 on average) representing a 0.6 % per year

increase compared to 1.5 % per year for non-residential buildings (around 200 kWh/m^2 on average). As a consequence of the energy and environmental issues, it is necessary to reduce the energy consumption of buildings. So, the air tightness of building envelopes is being improved and the air change rate due to infiltration is decreasing. It is then even more important than in the past that the buildings are equipped with well designed and working ventilation systems in order that the air renewal within buildings is ensured.
Balanced ventilation with heat recovery is one of the systems whose use is supposed to allow the buildings to reduce their energy consumption. These systems include air filters used to protect the heat exchanger (both on the fresh air and exhaust air sides) and to enhance the quality of supplied air.
In schools, it has been shown by various studies (Turunen et al. (2014), Mendel et al. (2013), Wargocki and Wyon (2013), Bako-Biro et al. (2012), De Gids (2007)) that the children are very sensitive to the amount of fresh air provided to them as there is a negative impact on their absenteeism, performances, well-being and perception of air quality when the air flow rate decreases. But the fresh air provided to the school by the mechanical systems comes from outdoors and it may be polluted by a mixture of various pollutants amongst them particles. Then the filtration of the incoming air can help to reduce the amount of pollutants injected into the schools and to enhance the indoor air quality. But how efficient the filters have to be in order to contribute to good indoor air quality?
The objective of our study was to determine the influence of the filtration efficiency of the particulate filters used for the filtration of the air provided to a nursery school by a balanced ventilation system with heat recovery on the particle concentrations within the school. There is according to our knowledge no data available in the literature on this topic.

2 METHOD AND MEASUREMENTS

Measurements with three different filtration levels were performed in a recent nursery school equipped with a balanced ventilation system with heat recovery. The school is located in a rural environment in France. Four identical filters have been installed in parallel within the air handling unit. The three types of filters are respectively classed G4, F7 and F9 according to NF EN 779 (2012).

2.1 The school and its ventilation system

2.1.1 The school

The nursery school is located in a rural environment (a town of 1200 inhabitants) at about 50 km from Lyon in France, away from busy roads and without specific outdoor pollution sources (see Figure 1). The school has been built in 2010 and about 100 children attend it in 4 different classrooms (see Figure 2) from Monday to Friday except on Wednesday afternoon. There are also corridors, a sports room, a sleeping room, toilets (with its own mechanical ventilation system) and offices for the teachers.

Figure 1 : Outside of the school


Figure 2 : Inside of a classroom

2.1.2 The ventilation system

The school is mechanically ventilated (100 % fresh air) with a balanced ventilation system with heat recovery for the mechanical driving of both the incoming and the exhaust air. The balanced ventilation system uses an air handling unit (see Figure 3) which in its normal use is equipped with F7 class filters (NF EN 779 (2012)) for the filtration of the incoming air while G4 class filters are used for the filtration of the exhaust air. On both sides, 4 filters are used in parallel.

The regulation of the ventilation system ensures that it is well balanced, that means the incoming air flow equals the exhaust air flow. The air flow varies between 500 and 2700 m³/h as it is modulated and depends on the number of children present within the classrooms.
The fresh air is diffused within the classrooms and the corridor via ceiling diffusers and is extracted via ceiling grilles.


Figure 3 : The air handling unit of the ventilation system

2.1.3 The filters

For the needs of our study it has been decided to use pleated G4 filters (Figure 4a), mini-pleated F7 filters (Figure 4b) and mini-pleated F9 filters (Figure 4c).

Figure 4a : The G4 filter

Figure 4b : The F7 filter

Figure 4c : The F9 filter

2.2 Measurements

During the measurements (3 consecutive days per week for 3 consecutive weeks in September and October 2014), the air flows have been set and periodically controlled (measurements of the air velocity on straight area of the ducts) to values always between 2400 and 2600 m^3/h (air exchange rate: 1.3 to 1.4 vol/h). The regulation of the ventilation system has allowed to balance the incoming and the exhaust air flows.

Air has been sampled alternatively and continuously inside the fresh air duct at the inlet of the air handling unit (which is supposed to be representative of the outdoor air), at the outlet of the air handling unit inside the delivered air duct and inside the exhaust air duct (which is supposed to be representative of the indoor air). These samples have been analyzed by an optical particle counter for particle concentration measurements in the 0.3 – 10 µm particle size range. Particle concentration values have been used for the in-situ filtration efficiency (E) calculation (comparison of the particles concentrations measured in the fresh air duct and in the delivered air duct) as function of particle size as well as for the calculation of the I/O ratio of the indoors to the outdoors particle concentration (comparison of the particles concentrations measured in the fresh air duct and in the exhaust air duct).

For each testing condition corresponding to one type of filters being studied (see 2.1.3.), 1 day measurements has been carried out per week (one day different every week) which means that 3 days measurements have been carried out per type of filters. The filters are installed at the end of the afternoon the day prior to the measurements.

Also, the performances (pressure drop and efficiency by particle size on DEHS particles) of the filters have been measured in laboratory prior being installed in-situ. The test air flow rate in laboratory is 650 m^3/h which corresponds approximately to the value met in-situ taking into account 2400 to 2600 m^3/h for 4 filters used in parallel.

Finally, during the measurements the doors and the windows of the school have been kept closed.

3 RESULTS AND DISCUSSION

3.1 Filters efficiency

The filter efficiency has been measured both in laboratory and in-situ. Both types of values can be compared because the air flow rate per filter is the same in laboratory and in-situ.
Each day of measurements, the efficiency has been calculated 10 to 15 times because the particles concentration has been alternatively and successively measured at the three measuring points. So for the in-situ values (see Figure 5) the curve represents an average taking into account about 35 to 40 results.

The results show that the in-situ filtration efficiency of the air handling unit is very close to that of the filters measured in laboratory (see Figure 5) which means that the filters are installed within the air handling unit with no significant leaks.

Not surprisingly the efficiency of the G4 filters on the finest particles being in size lower than 1 µm is very low and close to zero.
But with the use of the F7 filters and even more with the F9 filters the filtration efficiency is much better, respectively 52 % with the F7 filters and 82 % with the F9 filters at 0.4 µm.

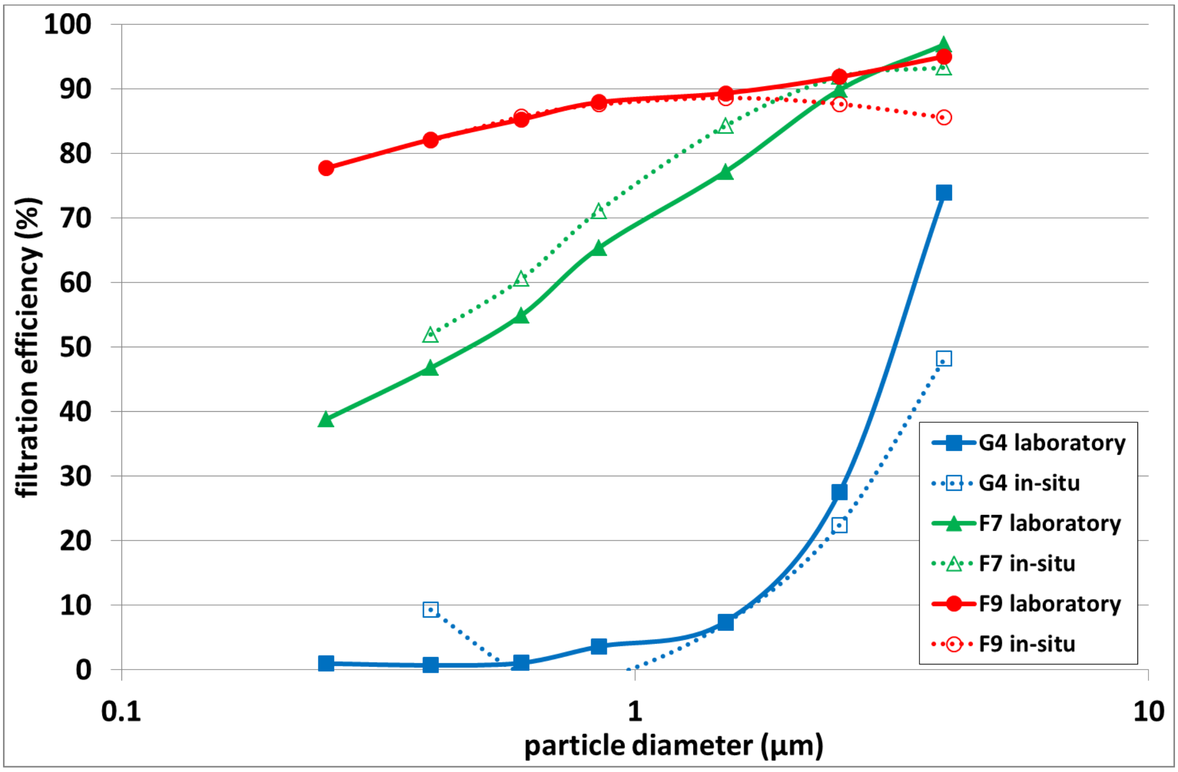


Figure 5 : Filter efficiency determined both in laboratory and in-situ

3.2 Indoor/outdoor particle concentration ratio

The indoor particle concentration depends on many parameters amongst them the outdoor particle concentration. For this reason the influence of the type of filters on the indoor particle concentration is expressed on the indoor/outdoor particle concentration ratio.
Figure 6 gives an example of typical indoor/outdoor particle concentration ratio as function of time, particle size and children attendance during a full school day.

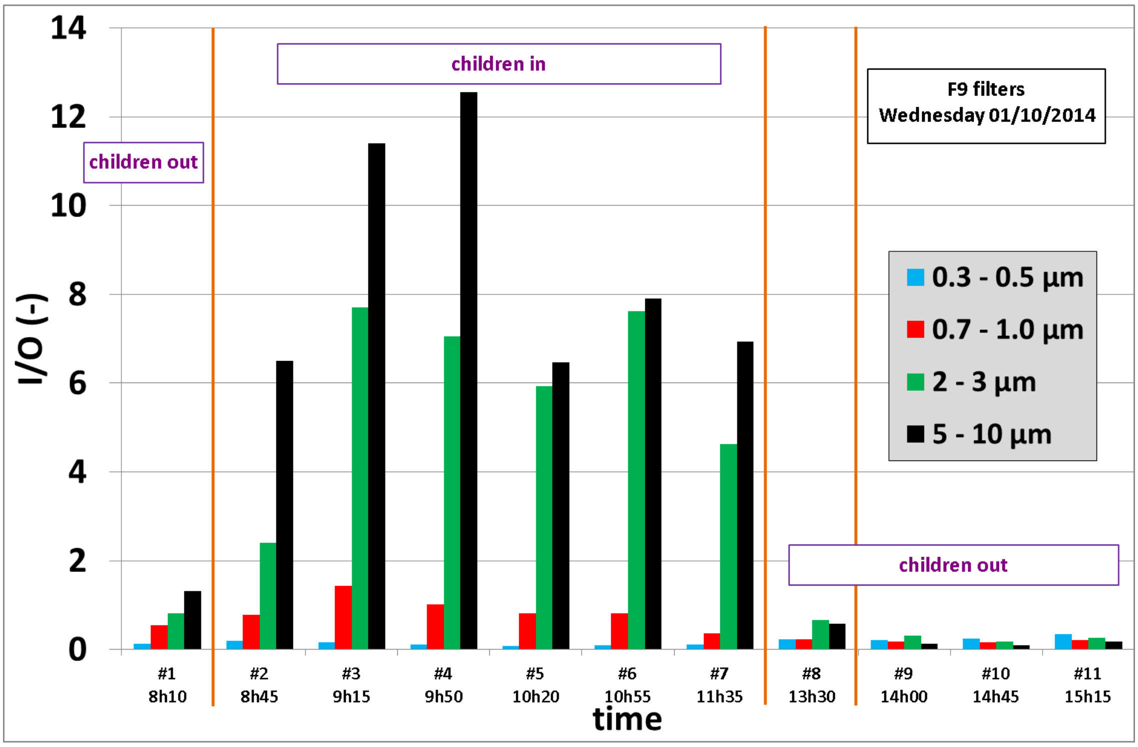


Figure 6 : Indoor/outdoor particle concentration ratio as function of time, particle size and children attendance

Before the children arrive to the school the I/O ratio is generally low, lower than 1. In the example presented in Figure 6, the I/O ratio is higher than 1 for the 5-10 µm particles which can be explained by the presence within the school of teachers. The I/O ratio increases immediately after the children arrive (at 8h30), see #2 to #7 in Figure 6, and this trend is more pronounced when the particle size increases with I/O ratio values which can largely exceed 1 ; the I/O ratio appears not influenced by the presence of the children if we consider the smallest measured particles (0.3 - 0.5 µm).

After the children and the teachers have leaved the school for lunch break (at 12h30), see #8 to #11 in Figure 6, the I/O ratio value returns back to much smaller values which are well below 1 for all the particle sizes considered (children do not go to school on Wednesday afternoon).

The summary of the 9 full school measurement days is reported in Table 1 and the indoor/outdoor particle concentration ratio is expressed as function of time, particle size and children attendance.

When the school is unoccupied, the I/O ratio decreases if the filtration efficiency increases: 0.92 to 0.95 with the G4 filters (the efficiency of these filters is close to zero), 0.41 to 0.52 with the F7 filters and 0.30 to 0.38 with the F9 filters for particles with diameter lower than 3 µm. For larger particles there is no clear relationship (the amount of particles is low both indoors and outdoors) but the I/O ratio is still lower than 1.

When the school is occupied by children and teachers, the I/O ratio is the same as when the school is unoccupied considering particles smaller than 0.5 µm; this is explained by the fact that these small particles mainly come from outdoors and are not generated indoors. But for larger particles, the I/O ratio is higher and increases when the particle size increases and when the filter class decreases. The I/O ratio exceeds 1 (largely in some situations) depending on the class of the filter and the particle size considered.

Table 1: Indoor/outdoor particle concentration ratio with no children within the school (in blue) and with the children at school (in red in brackets)

Filters	G4			F7			F9		
Particle diameter	Day 1	Day 2	Day3	Day 1	Day 2	Day3	Day 1	Day 2	Day3
0.3 - 0.5 (µm)	0.96	0.89	-	0.70	0.34	0.51	0.43	0.33	0.27
	0.93 (0.81)			0.52 (0.58)			0.34 (0.27)		
0.7 - 1.0 (µm)	0.84	1.06	-	0.31	0.48	0.43	0.41	0.32	0.18
	0.95 (1.43)			0.41 (0.79)			0.30 (0.75)		
2 - 3 (µm)	1.29	0.55	-	0.15	0.75	0.51	0.56	0.34	0.25
	0.92 (5.09)			0.47 (2.43)			0.38 (3.15)		
5 - 10 (µm)	0.52	0.41	-	0.15	1.19	0.63	0.73	0.29	0.14
	0.46 (6.86)			0.66 (6.39)			0.39 (5.24)		

4 CONCLUSIONS

In a recent (2010) nursery school located in a rural environment in France and equipped with a balanced ventilation system with heat recovery, the impact of the filtration efficiency level of the incoming air has been studied on the particle concentration within the school represented by the indoor/outdoor particle concentration ratio (I/O). There are 4 classrooms and an average of 100 children. Doors and windows were closed during the measurements. The air exchange rate was 1.3 to 1.4 vol/h.

When the children and the teachers are present, the I/O ratio value is primarily controlled by the filtration for the smallest particles (diameter < 0.7 µm); the I/O ratio is then close to 1 with G4 filters and decreases as the class of the filters increases. For the biggest particles which are generated indoors there is less influence of the filtration and the I/O ratio may largely exceed 1.
The filtration of the incoming air provided to the studied nursery school can reduce the exposition of the children to fine particles whose sources are outdoors. The influence of the class of the filters on the I/O ratio in a school has not been studied according to our knowledge and only one study carried out in a commercial has been identified (Rendek et al. (2011)) whose conclusions are similar to those of this study.

In order to reduce the amount of the biggest particles within the school, it has been decided to complete our study and to carry out the same kind of measurement with one portable air cleaner in operation within each of the 4 classrooms.

5 ACKNOWLEDGEMENTS

This study has been financially supported by the ″Air Filtration″ CETIAT technical committee. The authors are grateful to the city of Lent which has welcomed the study ; many thanks to the Mayor of the city and to the teachers of the school who have facilitated the work of the people in charge of the measurements. The authors are also grateful to the manufacturers (Camfil) who have provided the filters used for the study.

6 REFERENCES

European Commission (2013). *Report from the Commission to the European Parliament and the Council - Financial support for energy efficiency in buildings – Commission staff working document accompanying the document*. Document SWD (2013) 143 final, April 2013.

Turunen, M., Toyinbo, O., Putus, T., Nevalainen, A., Shaughnessy, R. and Haverinen-Shaughnessy, U. (2014). *Indoor environmental quality in school buildings and the health and wellbeing of students*. International Journal of Hygiene and Environmental Health, Vol. 217, No. 7, p. 733 – 739, September 2014.

Mendell, M. J., Eliseeva, E. A., Davies, M. M., Spears, M., Lobscheid, A., Fisk, W. J. and Apte, M. G. (2013). *Association of classroom ventilation with reduced illness absence: a prospective study in California elementary school*. Indoor Air, Vol. 23, No. 6, p. 515 – 528, December 2013.

Wargocki, P. and Wyon, D. (2013). *Providing better thermal and air quality conditions in school classrooms would be cost-effective*. Building and Environment, Vol. 59, p. 581 – 589, 2013.

Bako-Biro, Z., Clements-Croome, D. J., Kochhar, N., Awbi, H. B. and Williams, M. J. (2013). *Ventilation rates in schools and pupils' performance*. Building and Environment, Vol. 48, p. 215 – 223, February 2012.

De Gids, W. (2007). *Measurement of relation between ventilation and the learning skills of pupils in classrooms*. 28nd AIVC Conference, Crete Island, Greece, September 2007.

NF EN 779 (2002). *Particulate air filters for general ventilation – Determination of the filtration performance*.

Rendek, E., Ginestet, A., and Pugnet, D. (2011). *Impact of the filtration system on the indoor-outdoor particles concentration relationbship in an air conditioned office building*. 32nd AIVC Conference and 1st TightVent Conference, Brussels, Belgium, 12-13 October 2011.

ENERGY PERFORMANCE OF ACTIVE POLARIZATION FILTERS VS. CONVENTIONAL FILTERS IN HVAC SYSTEMS

Eduardo Theirs[1], Fernando Varela*[2], Cristina González[2], Prado Díaz de Mera[3] , Fernando Feldman[4] and Tomás Higuero[4]

1 JG Ingenieros
Paseo de la Habana 200
Madrid, Spain

2 ETS Ingenieros Industriales UNED
c/Juan del Rosal, 12
Madrid, Spain
**Corresponding author: fvarela@ind.uned.es*

***3** URJC (Universidad Rey Juan Carlos)*
c/ Tulipán, s/n
Móstoles, Spain

4 Aire Limpio
c/ Velázquez, 100, 4º izda.
Madrid, Spain

ABSTRACT

Pressure drop due to filters embedded in HVAC systems is one of the energy loss causes in building air conditioning systems, which can become important in large all-air systems with highly demanding filtering needs, such as hospitals, clean rooms, laboratories or pharmaceutical environments. This pressure drop increases with filter use, since grid spacing diminishes with filtering effect, due to accumulation of particles in the grid.

This energy loss can be partially avoided with the use of special filters, with the same filtering effect but that, based in a different filtering technology, lead to a lesser head loss. In this work we consider active polarization filters, which use an active electrostatic field to polarize both the fibers of a media pad and the particles to be removed. The polarized particles are drawn to the polarized media fibers and to each other, form bigger clusters that are more easily captured in the grid. This way, with a coarse filter (and consequently a lesser pressure drop) an equivalent filtering effect is achieved.

The aim of this work is to assess the effect in final energy use of a building of this kind of filters compared to conventional ones, during a filter life period between two substitution/cleaning of the filters.

A 7,500 m^2 non-residential building (office building) with a ventilation system has been chosen to perform a simulation with each one of the mentioned types of filters, which allows evaluating the energy consumption difference and the corresponding economic savings. This case has been considered as an average case, being the influence of the filters over the total energy consumed restricted, since filtering level is not very demanding and HVAC system considered is not an all-air system (VRV systems are considered for HVAC), where air flow levels are usually higher and so is consumed fan energy. However, an annual saving of more than a 7% and a cumulated saving in a 5 year life period of a 6% over the final energy consumption is achieved considering unused filters for case former mentioned.

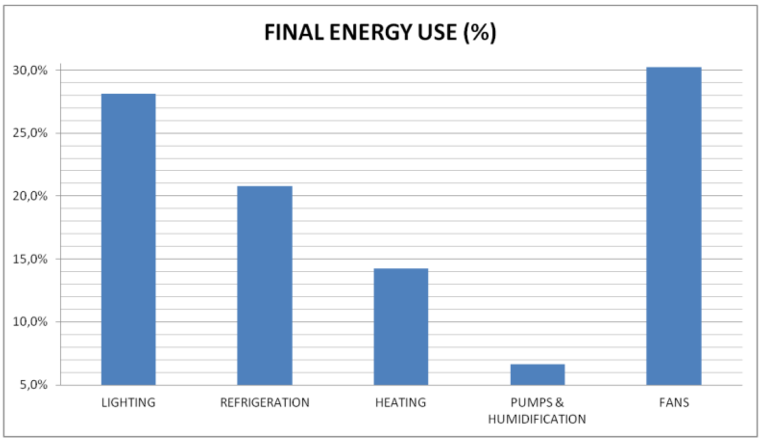
KEYWORDS

Energy efficiency, polarized filters, building energy savings.

1 INTRODUCTION

Nowadays, air transport is one of the most important sources of energy consumption in building's energy performance, thus a reduction in energy consumption in ventilation leads to an important reduction in building final energy consumption. In figure 1, a standard consumption distribution corresponding to a typical office building fulfilling Spanish energy Standards is shown.

Figure 1: Building final energy consumption in an office fulfilling Spanish Standards.



Many studies, among others, those performed by Brothers (Brothers et al., 1986), Wang (Wang et al., 1999), Webster (Webster et al., 2002), and Yang (Yang et al.,2011) have been made about fan energy consumption in air systems, mainly regarded to the importance of control over the flow amount in variable air volume systems (VAV).

Fan energy demand depends not only on the demanded air flow, but also in the pressure drop that must be overcome along the ductwork. Within this ductwork, were are going to focus in an important element, the filters, for two reasons: in one hand, their use is compulsory, and though they must be present in every single HVAC air system, and on the other hand, their pressure drop is significant in all cases, above all in those where sanitary facilities are involved.

The use of air filters which allow reducing the pressure drop inside the air handling units lowers the energy demanded by air handling units' fans improving the building energy performance and reducing consumptions of final energy, primary energy and CO_2 emissions.

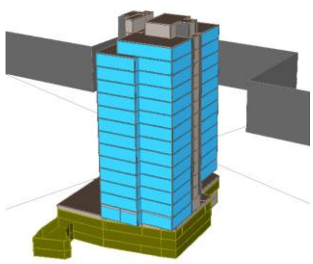
In this context, polarized filters use electricity in order to attract dust allowing the use of coarse grids that ease air pass reducing pressure drop and consequently reducing air handling units' fan power.

In this work, energy performance of polarized filters and conventional filters is assessed in a ventilation system of a standard office building in Madrid.

2 BASELINE DATA

A thirteen floor height office of 7,500 square meters located in Madrid with three underground floors (see figure 2) has been modelled to perform this study.

Figure 2: Building simulation model



Fan-coils and air handling units were considered for building's HVAC equipment, and ventilation system is driven by two CAV AHUs of 22.050m^3/h each. Airflow moved by the building air handling units was dimensioned according to Spanish regulations.

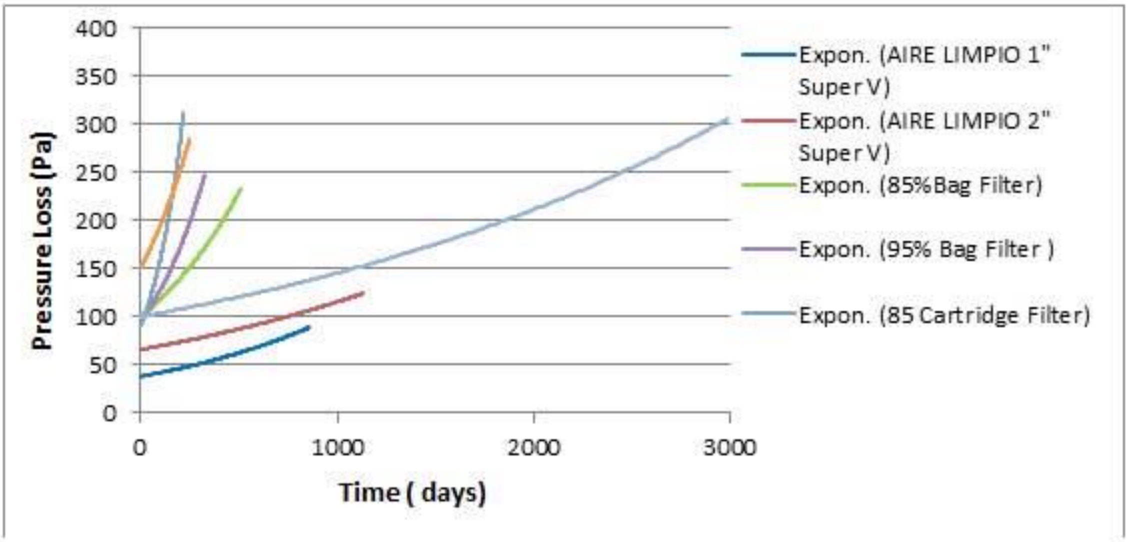
This case of study has been considered as an average case, being the influence of the filters over the total energy consumed restricted, since filtering level is not very demanding and HVAC system considered is not an all-air system (VRV systems are considered for HVAC), where air flow levels are usually higher and so is consumed fan energy.

3 METHODOLOGY

Simulations have been performed for a period of five years (usual polarization filtering substitution lapse), considering the increases of pressure drop caused by the accumulation of dust in the filters. Compliying with Spanidh regulation, in the supply side of the AHU, two filters in series are considered: F7 y F9 type. In the return side, a F7 filter is considered. For the comparison, two cases have been studied: Bag and Polarized filters.

The pressure drop due to the filters has been measured experimentally. These measures indicate that the pressure drop has an exponential evolution (figure 3).

Figure 3: Pressure drop evolution for different kind of filters (data courtesy of Aire Limpio)



Since energy consumption is proportional to pressure drop for a constant flow, in order to have a unique value representative of pressure drop along all the year, the following process has been made:

- Area under the exponential has been calculated integrating the exponential function along all measured days.
- Equivalent pressure drop has been calculated so that the area contained in the rectangular surface formed by this equivalent pressure and the measured days is the same as the calculated in the first step.

3.1 Pressure Drop due to the Use of Bag Filters

Measured pressure drop in a F7 bag filter and a F9 bag filter along the first year of use is shown in the next table:

Table 1: F7 bag filter and F9 bag filter pressure drop

Days	F7 Bag Filter	F9 Bag Filter
0	105,82 Pa	108,31 Pa
365	192,20 Pa	275,98 Pa

Next table indicates the equivalent pressure drop for these two filters:

Table 2: F7 bag filter and F9 bag filter pressure drop

Days	F7 Bag Filter	F9 Bag Filter
0 - 365	144,74 Pa	179,26 Pa

3.2 Pressure Drop due to the Use of Polarized Filters

Next figure shows the pressure drop in a F7 polarized filter and a F9 polarized filter:

Table 3: F7 bag filter and F9 bag filter pressure drop

Days	F7 Polarized Filter	F9 Polarized Filter
0	38,59 Pa	104,58 Pa
365	55,32 Pa	119,55 Pa
730	79,31 Pa	136,66 Pa
1,095	---	156,22 Pa
1,460	---	178,58 Pa
1,825	---	204,14 Pa

Equivalent pressure drop for these two filters is shown in the next figure:

Table 4: F7 bag filter and F9 bag filter pressure drop

Days	F7 Polarized Filter	F9 Polarized Filter
0 - 365	46,45 Pa	111,90 Pa
365 - 730	66,60 Pa	127,91 Pa
730 - 1,095	---	146,22 Pa
1,095 - 1,460	---	167,15 Pa
1,460 - 1,825	---	191,08 Pa

3.3 Simulation Cases

In order to study building energy performance the following combinations of filters has been simulated:

Table 5: Bag filter and polarized filter combinations simulated

Days	F7 Bag Filter	F9 Bag Filter	F7 Polarized Filter	F9 Polarized Filter
0 - 365	144,74 Pa	179,26 Pa	46,45 Pa	111,90 Pa
365 - 730	144,74 Pa	179,26 Pa	66,60 Pa	127,91 Pa
730 - 1,095	144,74 Pa	179,26 Pa	46,45 Pa	146,22 Pa
1,095 - 1,460	144,74 Pa	179,26 Pa	66,60 Pa	167,15 Pa
1,460 - 1,825	144,74 Pa	179,26 Pa	46,45 Pa	191,08 Pa

These combinations of pressure drops were introduced in an air handling unit commercial selection program where the fan supply and the fan return power was calculated.

Calculated fan power is shown in the next table:

Table 6: Bag filter and polarized filter combinations simulated

Days	Bag Filter		Polarized Filter	
	Supply Fan	Return Fan	Supply Fan	Return Fan
0 - 365	8,65 kW	6,68 kW	7,36 kW	6,04 kW
365 - 730	8,65 kW	6,68 kW	7,66 kW	6,18 kW
730 - 1,095	8,65 kW	6,68 kW	7,63 kW	6,04 kW
1,095 - 1,460	8,65 kW	6,68 kW	7,92 kW	6,18 kW
1,460 - 1,825	8,65 kW	6,68 kW	8,00 kW	6,04 kW

4 RESULTS

4.1 Fans Energy Performance evolution using Bag Filters

Simulations have been made to calculate fans' final energy consumption taking into account fan powers using bag filters mentioned in table 6. Values shown in table 7 include terminal units' fans:

Table 7: Fan final energy consumption using bag filters

Days	Fans' Final Energy Consumption
0 - 365	219,880 kWh
365 - 730	219,880 kWh
730 - 1,095	219,880 kWh
1,095 - 1,460	219,880 kWh
1,460 - 1,825	219,880 kWh

4.2 Fans Energy Performance evolution using Polarized Filters

Simulations have been made to calculate fans' final energy consumption taking into account fan powers using polarized filters mentioned in table 6:

Table 8: Fan final energy consumption using polarized filters

Days	Fans' Final Energy Consumption	Polarization Final Energy Consumption	Total Final Energy Consumption
0 - 365	203,104 kWh	54 kWh	203,158 kWh
365 - 730	206,930 kWh	54 kWh	206,984 kWh
730 - 1,095	205,451 kWh	54 kWh	205,505 kWh
1,095 - 1,460	209,188 kWh	54 kWh	209,242 kWh
1,460 - 1,825	208,667 kWh	54 kWh	208,721 kWh

Polarization process has an electric power consumption of 12 W during an estimate operation time of 4.500 hours per year.

4.3 5-year performance comparison

Table 9 shows fans' final energy consumption comparison:

Table 9: Fan final energy consumption comparison

Days	Fans' Final Energy Consumption (Bag Filters)	Fans' Final Energy Consumption (Polarization Filters)
0 - 365	219,880 kWh	203,158 kWh
365 - 730	219,880 kWh	206,984 kWh
730 - 1,095	219,880 kWh	205,505 kWh
1,095 - 1,460	219,880 kWh	209,242 kWh
1,460 - 1,825	219,880 kWh	208,721 kWh
TOTAL	**1,099,400 kWh**	**1,033,610 kWh**

The percentages of final energy savings are shown in table 10:

Table 10: Percentages of final energy savings using polarization filters

Days	Final Energy Savings using Polarization Filters (%)
0 - 365	7,6%
365 - 730	5,9%
730 - 1,095	6,5%

1,095 - 1,460	4,8%
1,460 - 1,825	5,1%
TOTAL	**6,0%**

As can be seen in the former table, the use of polarization filters represents a saving of approximately a 6% in electric energy consumption along a period of life of 5 years.

It is important to remember that this case study has been considered as an average case, being the influence of the filters over the total energy consumed restricted, since filtering level is not very demanding and HVAC system considered is not an all-air system (VRV systems are considered for HVAC), where airflow levels are usually higher and so is consumed fan energy.

5 CONCLUSIONS

As shown before, a study on the impact of differences in building energy performance using bag filters or using polarized filters has been performed, being the main conclusions:

- Fan use derived from HVAC systems is one of the most important consumptions in non-residential building energy performance.
- Fan energy use can be reduced diminishing pressure drop throughout the ductwork.
- Air handling units' fan pressure drop must be carefully studied in order to improve building energy performance.
- Polarization filter usage reduces significantly the pressure drop inside air handling units.
- Savings of a 6% of fans' final energy consumption can be achieved in average office buildings using polarized filters instead of traditional bag filters.

6 ACKNOWLEDGEMENTS

This work was supported by the School of Industrial Engineering (UNED) under Grants 2014-IFC01 and 2014-IEN16.

7 REFERENCES

Brothers, P.W.;Warren, M.L.(1986). *Fan energy use in variable air volume systems*. ASHRAE Trans. 92(2B)

Wang, S. (1999). *Dynamic simulation of building VAV air-conditioning system and evaluation of EMCS on-line control strategies*. Building and Environment, 34(6), 681-705.

Webster, T.; Bauman, F.; Ring, E. (2002). *Supply fan energy use in pressurized underfloor air distribution systems*. CBE Summary Report.

Yang, X;Jin, X;Du, Z;Fan, B; Chai, X (2011) *Evaluation of four control strategies for building VAV air-conditioning systems*. Energy and Buildings, 43(2-3), 414-422.

THE EFFECT OF CO_2 ON THE NOCTURNAL RESTLESSNESS OF AN ALZHEIMER PATIENT

Bart Cremers

Zehnder Group Nederland
Lingenstraat 2
8028 PM Zwolle, The Netherlands
bart.cremers@zehndergroup.com

ABSTRACT

At typical indoor CO_2 levels there is no scientific evidence that CO_2 is harmful to comfort and health of healthy persons, though there is a potential for negative effects on some aspects of performance. Research also indicates that insufficient bedroom ventilation may negatively affect the quality of sleep.

This document reports practical observations of the effect of CO_2 on the nocturnal restlessness of an Alzheimer patient. During a month, the CO_2 values in the bedroom of the patient were recorded. Typical values of the CO_2 level in the bedroom are evaluated, and related to the occupancy of the bedroom and the position of an exterior window and an interior door to the hallway.

The nocturnal behavior of the Alzheimer patient was observed by the partner of the patient occasionally. The result of the study on this particular Alzheimer patient was that the nocturnal restlessness was absent when the bedroom CO_2 level did not exceed 750 – 800 ppm. Above this value, the restlessness emerged in symptoms like humming, teeth grinding, apnoea and sometimes leading to panic. Intervention of the partner by opening the window was followed by a gradual decrease in CO_2 and consequently a more restful sleep.

Although this study is the practical observation of one Alzheimer patient only, it shows that the effect of typical indoor CO_2 values may be larger for people suffering from Alzheimer's disease than for healthy persons. More research on a larger number of patients is being carried out at the moment.

KEYWORDS

Indoor air quality, sleep quality, Alzheimer, nocturnal restlessness

1 INTRODUCTION

The relation between the indoor air quality in buildings and the health of occupants gets more and more attention nowadays. However, it is difficult to point to health effects that have a direct relation to poor indoor air quality. In a risk assessment study Logue (2011) indicated that the risk of indoor air quality on shortened life expectancy is in between the risk of car accidents and the risk of heart diseases.

There may be a stronger relation between the indoor air quality in bedrooms and the sleep quality of inhabitants. Strøm-Tejsen et al. (2014) showed that a higher ventilation rate (open window) had a significant positive effect on sleep latency and the ability to fall asleep.

This research reports the measurement of bedroom CO_2 levels and nocturnal restless behavior of an Alzheimer patient as observed by the husband of the patient.

2 METHOD

A couple (husband and wife) is living in a naturally ventilated house. Fresh air is provided in a natural way via the envelope of the house, possibly increased by the opening of one or more windows.

Since 2007, the wife is suffering from Alzheimer's disease, with gradual increasing symptoms of dementia. During days, she occasionally suffers from restless behaviour, indicated by periodic introvert behaviour and humming sounds. During the nights, the restless behaviour occasionally shows, indicated by snoring, apnoea, restless sleep and panic when she wakes in the night.

During nights and in the afternoon, the wife, and sometimes the husband, take a sleep in the bedroom (fig. 1). The window in the bedroom is a top-hinged turnable window which is occasionally opened according to the wish of the husband. The interior door from the bedroom to the hallway may be opened or closed, as desired by the husband.

Figure 1: The bedroom with a view on the window.

An indicator for the indoor air quality in the bedroom is the CO_2 level, which has been recorded in a 15 minute interval with a CO_2 logger (Wöhler type CDL 210, see fig. 2a). This logger simultaneously records temperature and relative humidity as well. The logger was placed on a small table next to the bed on the side where the husband sleeps, as far as possible from the direct breathing area.

In order to avoid psychological influence by the indicated CO_2 level, it was proposed to cover the screen with tape, but this was refused by the husband so he could see whether an intervention like opening a window had an effect on the CO_2 level.

The nocturnal behaviour of the wife was described during the night and the day by the husband in a notebook. Incidents of restless behaviour were noted with the specific time. Also, changes in the window position and interior door position were noted. Lastly, the occupation of the bedroom was noted. For an example of one page of the notebook (excluding Dutch description of the behaviour), see figure 2b.

The indoor air quality and the observed behaviour was recorded during 5 weeks from the 5th of February 2014 until the 13th of March 2014.

a) 

b) 

Figure 2 a: The CO_2 logger, b: example page of the notebook.

3 INFLUENCE OF WINDOW AND DOOR POSITION ON CO_2 LEVEL

In order to give an idea how the bedroom CO_2 level was dependent on occupation and on window/door position, the average CO_2 level was evaluated in categories. Figure 3 shows columns in the front row for an unoccupied bedroom, the middle row columns for one person and columns in the back row when two persons are in the bedroom. Furthermore, the horizontal axis shows the positions of window and interior door (closed or open).

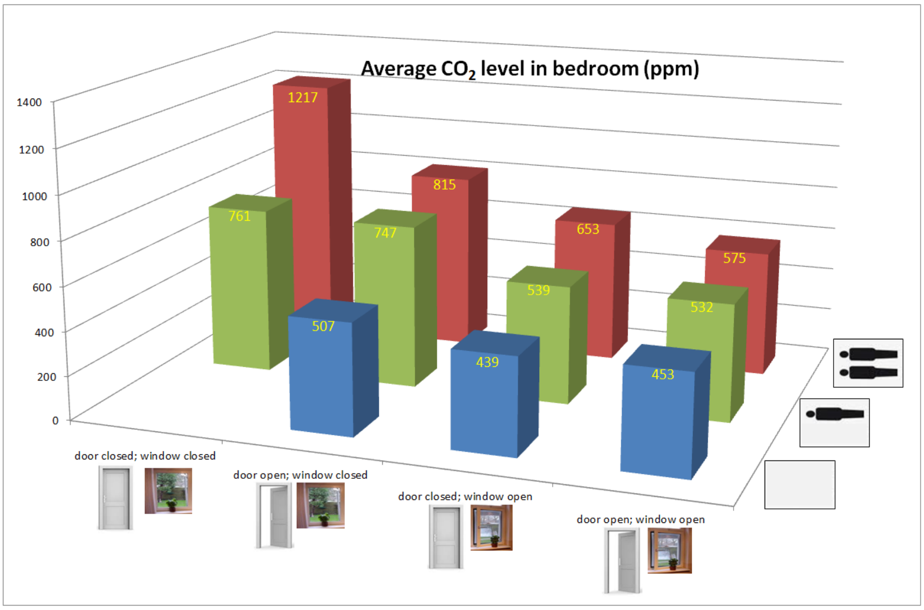


Figure 3: The effect of occupation, window and door position on the average CO_2 level in the bedroom. Left to right: door & window closed, door open & window closed, door closed & window open, door & window open.

As expected, the CO_2 level rises with the number of people in the room. The results also show that the largest reduction of CO_2 can be achieved by opening the window, and leaving the door open to the hallway also helps to reduce the CO_2 level. Obviously, opening the window increases mixing of indoor air with fresh outdoor air, while opening the door increases mixing with indoor air from the hallway. There is one exception to this last effect: when the bedroom is unoccupied, an open door to the hallway leads to a slightly higher CO_2 value, because CO_2 level from the hallway can be larger than CO_2 level from the bedroom.

The average values of CO_2 level in the bedroom are for a 5 week period, so they cannot be regarded as statistical. Moreover, the number of recordings for each situation are not comparable. There were a lot of recordings with window and door both open, while only a few short periods with window and door both closed. In spite of this, the effect of occupation and window and door position on the CO_2 level is logical and gives guidance to keep the indoor air quality to a certain level.

4 INFLUENCE OF CO_2 LEVEL ON RESTLESSNESS OF ALZHEIMER PATIENT

Figure 4 shows the first 5 days of the recording period, where the husband experimented a lot with opening and closing of windows and doors in the bedroom. The observed restless behavior of his wife as described in the notebook coincides with peaks in the bedroom CO_2 level. Whenever this behavior was apparent, the window and/or door was opened in order to lower the CO_2 level again. According to the observations, approximately 30 to 45 minutes after this intervention (also indicated in the graph), his wife was breathing and sleeping normally again.

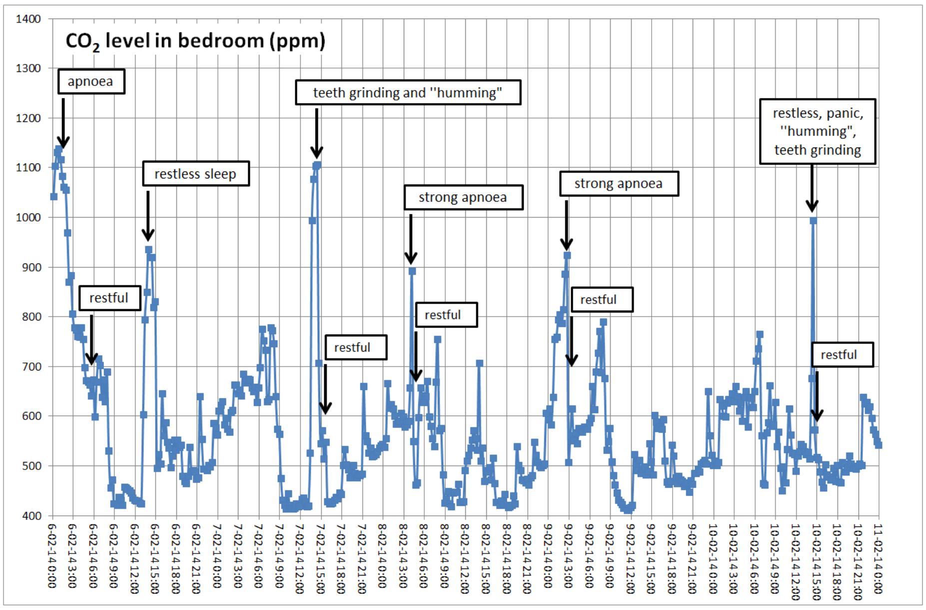


Figure 4: The bedroom CO_2 level and the observed behavior of the Alzheimer patient during a five day experimental period.

After a couple of nights of experimenting, the husband of the Alzheimer patient decided to keep an eye on the CO_2 level and intervene with window and door opening in a more anticipating way. Figure 5 shows that the restless behavior during sleeping has decreased in occurrences, except for some nights when CO_2 levels were quite high where window or door was left closed.

From the recorded period and the noted observations, the data seems to indicate that the restless sleep behaviour is not observed when CO_2 levels are kept below approximately 800 ppm.

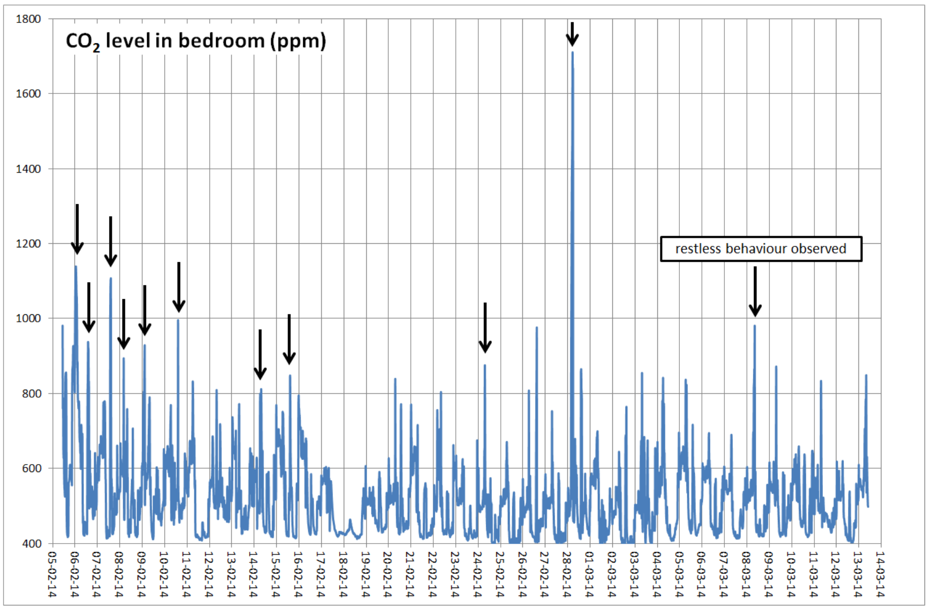


Figure 5: The CO_2 level and the occurrences of restless behavior (arrows) during the 5 week recording period.

5 DISCUSSION

The restlessness during the sleep of the Alzheimer patient in this research seems to be coincident with high levels of CO_2 in the bedroom. According to the observations of her husband during the recording period, but also afterwards, the restless behavior has not been observed anymore when the CO_2 levels are kept below 800 ppm. The temperature and relative humidity recordings were also available, but they had no clear deviating pattern coinciding with the restless behavior of the Alzheimer patient.

From the average CO_2 levels recorded in the bedroom, the advice is given to at least open the window of the bedroom in order to keep the CO_2 level below 800 ppm when the bedroom is occupied with one or two persons.

Following this strategy, the husband decided to place two CO_2 loggers permanently, one for the bedroom and one for the living room. Also daytime restless behavior seems to have been decreased since the time that living room CO_2 levels are kept below 800 ppm by opening windows and doors occasionally. Before this knowledge, it was concluded that the restless behavior during for instance birthday parties was originating from the larger number of people in the living room, bringing stress signals to the Alzheimer patient. But this research and the observations may indicate that the elevated CO_2 level itself is responsible for the restless behavior. Nowadays, people are welcome to the house again, without restless behavior, as long as open windows and door keep the CO_2 level low.

The recordings and observations in this research are for one patient only. But following this research, a couple of initiatives have been started to monitor the indoor air quality and the nocturnal restlessness of Alzheimer patients in care facilities. Only after this larger research these early observations can be substantially backed up by more scientific evidence.

The question arises whether the relation between CO_2 level and restless behavior can be broadened to a larger group like people suffering from dementia, or even healthy people. The author of this article postulates that healthy people have an adaptive response to higher CO_2 levels, meaning they can cope with elevated levels, both physically and mentally, without suffering restless behavior. But people suffering from Alzheimer, or another form of dementia, may have lost this adaptive response to high CO_2 levels and can react physically more intense and already at relatively low CO_2 levels of about 800 ppm. Extensive observational and medical studies have to be carried out to prove if this is the case.

6 CONCLUSIONS

This research shows that the restless behavior of an Alzheimer patient coincides with peaks of the bedroom CO_2 level above 800 ppm. The observed restless sleeping behavior like snoring, teeth-grinding, apnoea and panic were not observed anymore when CO_2 levels are kept below 800 ppm. After a peak of CO_2 level and coincident restless behavior, a window was opened so that CO_2 level decreased and after 30 to 45 minutes the sleep of the Alzheimer patient was observed to be restful again.

Much more research should be carried out in the near future if the relation between CO_2 level and restlessness also exists for a larger group of Alzheimer patients, or wider for a group of people suffering from dementia or even for healthy people.

7 REFERENCES

Strøm-Tejsen P, Wargocki P, Wyon DP and Kondracka A.(2014). *The effect of air quality on sleep*. In: Proceedings of the 13th International Conference on Indoor Air Quality and Climate – Indoor Air 2014, Hong Kong, Paper HP0506.

J.M. Logue, P.N. Price, M.H. Sherman, B.C. Singer (2011), *Why We Ventilate*; LBNL-5093E

AIR CHANGE RATE MEASUREMENTS USING INDOOR/OUTDOOR RATIO OF PM2.5

Lingshan Li1,Xiaofeng Li*2, Pok Lun Cheng 1

1 Tsinghua University
Dept. Of Building Science, School of Architecture, Tsinghua University
Beijing, 100084, China

2 Tsinghua University
Dept. Of Building Science, School of Architecture, Tsinghua University
Beijing, 100084, China
**Corresponding author: xfli@tsinghua.edu.cn*

Abstract

According to past researches, most people spend 80%-90% of their time indoors. The ventilation is very important to people's health and the comfortable surroundings around us. From the viewpoint of energy saving, mechanical ventilation will consume a large amount of additional energy. So variety of ways measuring natural ventilation is worth considering. In fact, in real life, many people tend to have their windows shut rather than open, and the reasons are complex.

Some reasons are due to safety, maintaining a good thermal environment, harmful outdoor pollutants, shielding from outside noise etc. Thus, it is necessary to set an index of low ventilation rate, in order to simulate the situation of windows shut or slightly opened. The ACH is controlled between 0.5 and 1, by adjusting the window openings. As measurement technique, the constant injection method can be used to measure the infiltration rate of natural ventilation. One useful way is the dry ice method, in which dry ice in an insulated container is used as a constant injection of tracer gas. Apart from dry ice method, the PM2.5 is considered to be another kind of tracer gas. The main sources of PM2.5 come from burning of fossil fuels. If there are no indoor sources, e.g. smoking or waste-combustion, we can assume that there are no initial indoor sources of PM2.5, where all sources come from outdoors. Then the I/O ratio of PM2.5 can be used to determine different ventilation rates.

In this study, the ACH from tracer gas method and I/O ratio of PM2.5 was compared. The relevance between ACH and I/O ratio will be verified through a specific equation to get further conclusion, through which we can get ACH directly by means of measuring I/O ratio of PM2.5.

Keywords

ACH; PM2.5; Natural ventilation; Constant injection; Dry ice method; I/O ratio

1 INTRODUCTION

Evidence indicates that most people spend 85~90% of their time indoors, and the quality of air is decided by the concentration of all particles. Natural ventilation is an important way to improve IAQ, while the extent of the windows' opening has a great influence on the ventilation rate.

Concerning about the measurement of natural ventilation, many researchers have summarized many kinds of methods. Many of them are used to measure the air changes per hour (ACH), such as simulation, tracer gas methods, etc.

Many studies have introduced the background, theory, usage conditions and economic benefits to measure ACH by tracer gas, such as Heidt and Werner (1986), Sherman (1990), Roulet & Vandaele (1991) and McWilliams J. (2003)

The decay method is widely used in past researches, using only the decay period of the tracer gas to get the ACH, where the time interval depends on the ACH. Since the measured ACH range is lower than 1.0, the time to change all the air in room is more than an hour, so this method is not suitable for low unsteady ventilation rates. When measuring unsteady ventilation, only the constant injection method is applicable.

Recent years, PM2.5 becomes the hot spot of scientists. Especially in Beijing, the outdoor concentration of PM2.5 is high. According to weather statistics of Beijing Municipal Environmental Protection Bureau, 58 days in 2014 and 45 days in 2013 are heavy polluted, in which the concentration of PM2.5 is higher than 150 mg/m3.days when the concentration is over 100mg/m3 are nearly 30%. So it is an excellent tracer source to use in field measurements.

Chen and Zhao studied the possibility of using particles such as PM10, PM2.5 to get a relevance between indoor and outdoor (I/0) ratio of PM and ACH both unsteady- and steady-state differential equations were put forward in their study

Assuming there's no indoor sources of PM2.5, where particles on surfaces and clothes being negligible, it can be considered that all particles are from outdoors to indoors by natural ventilation.

Our aim is to measure the ACH with a constant injection method by using CO2 released from dry ice, and get the I/0 ratio of the PM2.5 concentration at the same time. Then, the relevance between these two parameters can be compared.

2 METHODOLOGY

This part will explain the two methods used in the comparison for the calculation of ACH. A well-known tracer gas method was used to prove the ACH calculated with the PM2.5 method. Since the reliance of PM2.5 method is unknown, the ACH taken from the tracer gas method is set as standard and regarded as a-true. So in this experiment, it is vital to get the correct result of a-true with great caution, or the standard error will be large.

2.1 Tracer gas method

2.1.1 ACH calculation method

The constant injection method using a steady release rate of tracer gas can be expressed as:

$$\Delta C = \frac{\Delta \tau}{V_{zone}}(F - NV_{zone}(C_1 - C_e)) \qquad (1)$$

ΔC:Change of tracer gas concentration,
Δτ :Time interval,
Vzone :Zone volume
F :Tracer gas injection rate
N :number of ACH
C1 :indoor tracer gas concentration
Ce :outdoor tracer gas concentration.

The only unknown parameter is N, with all the rest as input values. The calculation was repeated for each consequent time interval to get a theoretical curve.

Between the measured and theoretical values, a curve fit equation was used, by using the solver tool in Excel:

$$\mathrm{Error}\left(C_{i,t}\right) = \left(C_1 - C_{i,t}\right)^2 \qquad (2)$$

The calculated sums of the errors were small due to this method, and varied between 1-5%, so the curve fit had an error of 1-5%. This calculation method is applicable with a variable tracer gas concentration, when the ACH is unsteady for the constant injection.

2.1.2 Tracer gas CO2

The dry ice in an insulated box works as a constant injection tracer source. The sublimation rate of dry ice is affected by the insulation of the box and indoor temperature. If the dry ice is placed in an insulated box, the heat transfer will eventually reach steady state.
The weight loss of the dry ice box was used to calculate the emission rate of CO2 with the following formula:

$$F_{CO_2} = \dot{m}\frac{V_m}{M_m} \quad (3)$$

FCO2: Volumetric release rate of CO2
Vm : Molar volume of CO2
Mm : Molar mass of CO2.

In this study, the dry ice box was positioned in the center of the room and weighted twice every day (night and morning). The weight has an accuracy of ±1 g.
CO2 concentrations were measured with Telaire 7001. Measuring range of the CO2 sensors was between 0-2500 ppm with an accuracy of ±5%. The CO2 concentration was measured with a time interval of 1 min. Two sensors were positioned inside in the different places in the office. The average was used between the two sensors, with a position difference of less than 5 %. One CO2 sensor was used to measure the outdoor concentration, positioned on the rooftop.

2.2 PM2.5 method

2.2.1 ACH calculation

According to the conservation of mass, the differential equation can be written as follow:

$$V\frac{dC_{in}}{dt} = aPVC_{out} - aVC_{in} - KVC_{in} + S \quad (4)$$

V: Room volume
T: Time
A: ACH
P: Particle penetration factor
K: Particle deposition rate
S: Indoor particle emission rate.
It should be noted that the suspension of particles is neglected in this equation.
The steady-state equation can be simplified as:

$$\frac{C_{out}}{C_{in} - C_s} = \frac{K}{P}\left(\frac{1}{a}\right) + \frac{1}{P} \quad (5)$$

Bennett and Koutrakis (2006) developed an alternative method for calculating the infiltration factor using time-dependent indoor and outdoor particle concentrations and ACH. Assuming there are no indoor particle sources:

$$C_{in,t} = \frac{aP}{K+a}C_{out}\left(1 - e^{-(K+a)\Delta t}\right) + C_{in,t-\Delta t}e^{-(a+K)\Delta t} \quad (6)$$

The two important parameters in the equation are P and K. They are discussed as follow.

2.2.2 The particle penetration factor——P

P is the particle penetration factor, defined by the ratio of the last and former concentration. It can be changed by the different kind of windows and opening angle, or even by the air flow rate.

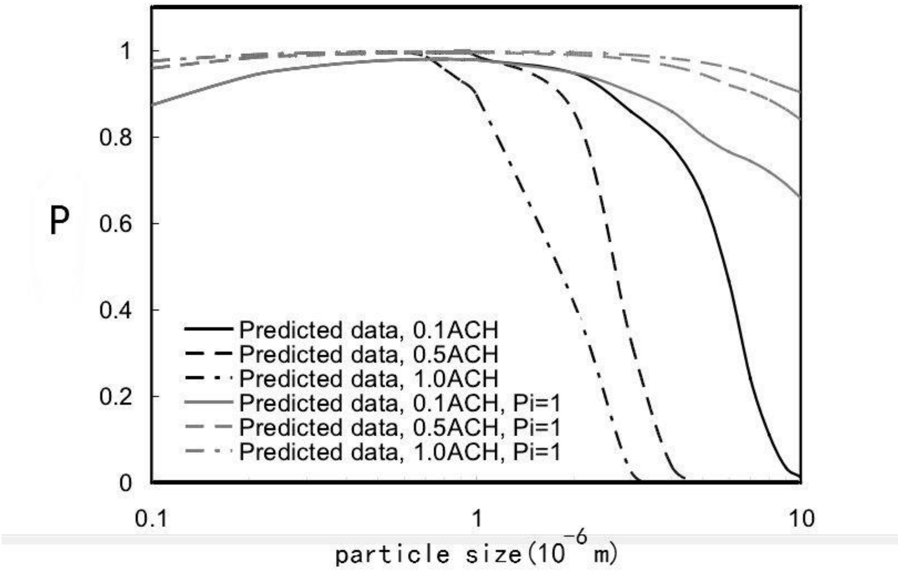


Figure 26: P for different ACH

Chun Chen gives the common distribution of P for small rooms, shown in Fig. 1.
It can be seen that when the ACH is in the range of 0.5~1.0, predicted data for P is close to 1.0. So in the beginning, P is set as 0.9/0.95/1.0 for individual calculation for analyzing.

2.2.3 The particle deposition rate——K

Influencing parameters of K is complex. The most important aspect is particle size. Others like airflow distribution, temperature, humidity, surface of the table and wall also influences K.
Due to other researches can't be decided by a constant value to become a parameter of the formula, but statistics show that K always is between 0.05~0.5.
Now, the very important task is to get a proper single value K from the experiment. Constant value doesn't mean that K is not changeable. It is just for the convenience of calculation method. During the calculation process, the ACH from the dry ice method is set as a-true. So the perfect result is when the ACH from PM2.5 method is the same as a-true. The error can also be used to judge if the relevance is conceivable. By comparison of different K, we can get the most accurate one to be the parameter of the formula.

2.2.4 Indoor and outdoor (I/O) ratio of PM2.5

The I/O ratio is defined as：

I/O Ratio = $\frac{C_{in}}{C_{out}}$ (7)

To get the PM2.5 concentration, the instrument AM510 was used with a logging interval of 1 min. The sensor was zero calibrated every 24 hours.
The error contains two parts: one is the mechanical error, and another part is caused by the temperature. For the mechanical part, the instrument has a preset correction factor. Comparing with the standard concentration of meteorological observatory, we can set one certain factor for PM2.5. The factor is not accurate for many reasons, so error comes. For the temperature part, the standard temperature is set during zero calibration when the temperature change and error appears. Taking into account these errors, only the concentrations more than 100mg/m3 was considered.
After getting both the indoor and outdoor concentration of PM2.5, the I/0 ratios can directly be used in the iterative calculation to get the ACH. The calculation part was executed with the help of MATLAB.

2.3 Test room chosen for experiment

The test room chosen is a 3 story office building on the second floor, in Beijing, China. The room faces north-west, with two windows facing both north and west direction, as shown in Fig 2.

The room volume is 57.4 m^3, (4.35m x 3.30m x 4.00m) which is an important parameter of tracer gas method. Most of the surface is white wall or smooth wooden desks, knits are little. So the deposition rate K is not so big in this occasion.

The air flow rate was controlled by changing the angle of the windows 3 and 4. The remaining openings were sealed by transparent adhesive tape to avoid multi-zone condition. So all the windows use single-sided ventilation, where windows are faced towards one direction. When the windows are opened, the influence of window frames is negligible, making P equal to 1.0.

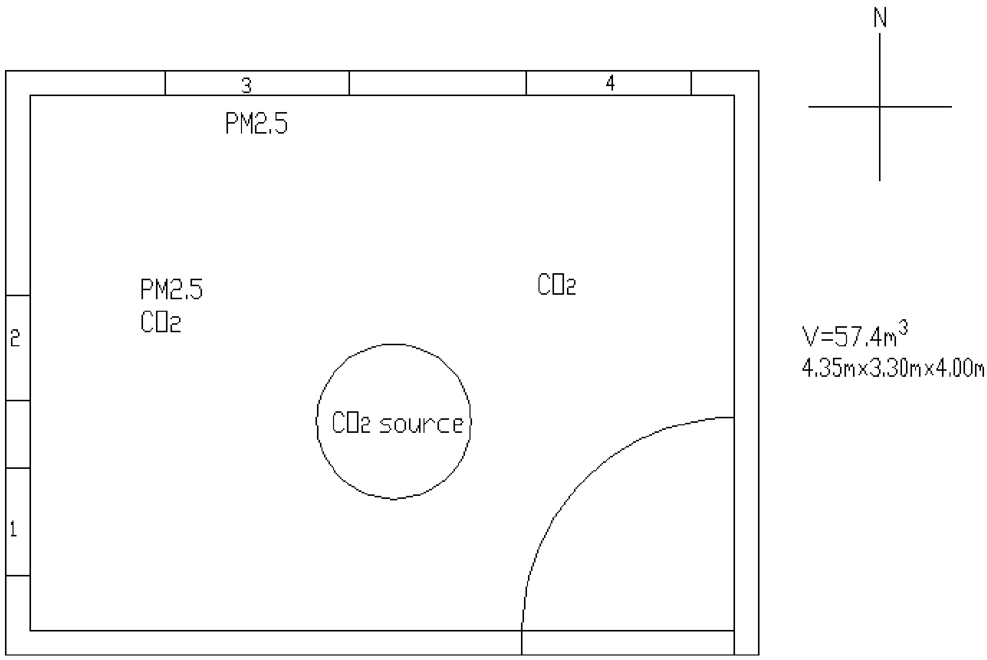


Figure 2: Room dimensions

3. RESULTS

This part shows the results from the experiments, which was conducted between April to May 2015.

3.1 Calculation of ACH using tracer gas method

The indoor CO2 graph was divided between: step-up, constant and decay, for the calculation of ACH (shown in Fig. 3).This was done in order to calculate the ACH with minimal curve fit errors. The calculated ACH is shown directly next to the indoor CO2 concentration graph, with brackets representing the time interval chosen for the calculation part.

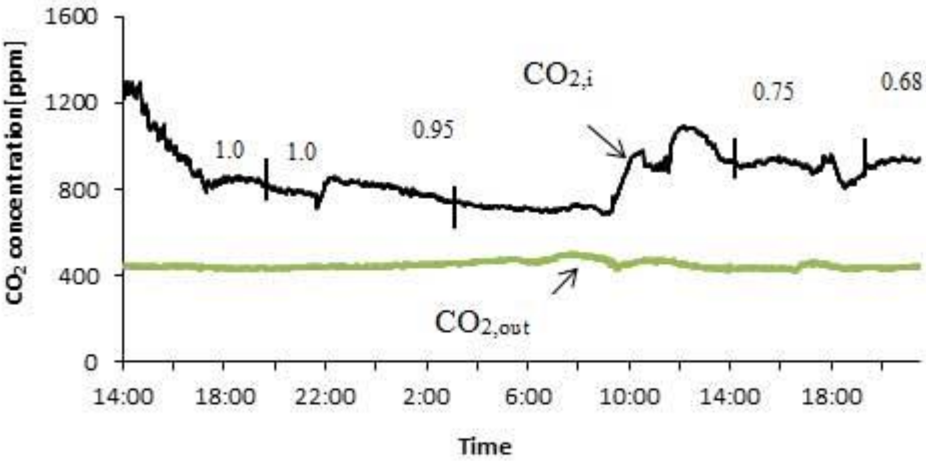


Figure 27: Calculation of ACH using dry ice method

There were no people or other disturbances present during the measurements, except two times per day weighting the dry ice which only took half minute per time. This short amount of time of disturbance was considered to have no effect on the calculated ACH.

3.2 Calculation of ACH using PM2.5

The indoor and outdoor PM2.5 concentration for a typical day is shown in Fig. 4.

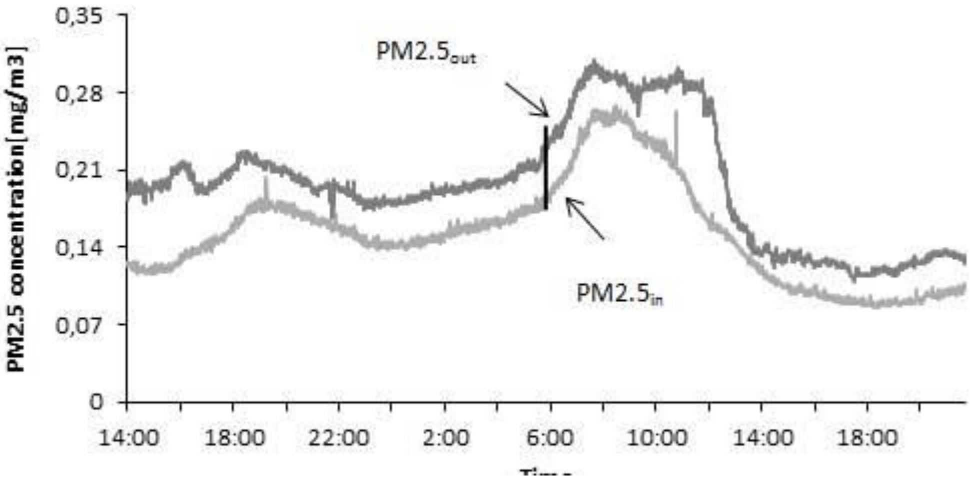


Figure 28: Concentration of PM2.5

When the indoor and outdoor concentrations have a nearly parallel trend, the data can be used for the calculation of a(PM2.5). Also, when the ACH is under 1.0, the PM2.5 concentration should be steady for at least one hour.
Similar to the calculation of dry ice method, the calculation part was divided into parts. The time intervals chosen were 2 hours during to the consideration of accuracy.

3.3 Verification of PM2.5 method

When P is constant, the only unknown parameter remaining is K, which can be found through statistical method.
For the comparison of the two methods, the ACH from the dry ice and PM2.5 method is set as a-true and a(PM2.5), respectively.
The aim at first is to figure out which combination of P and K can serve the most reliable formula. So the data points where P and K can measure closest to a-true in one period of time

was chosen, as shown in Fig 5. In this figure, different combinations of P and K are shown, which needs further analyzing for the most rational P and K.

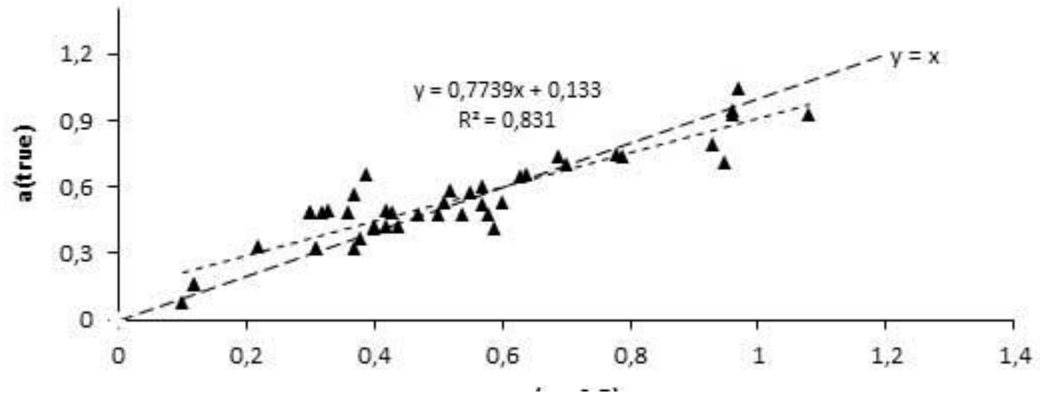


Figure 29: Comparison of the two methods with different K.

The data was further divided between ACH<0.4 and ACH>0.4, with the statistics of P and K for ACH<0.4 shown in Table 1 and 2. The ACH below 0.4 was excluded, due to inconsistent relevance. It would also take too long time for the indoor air to change 1 time and getting a steady I/O ratio of PM2.5.

Table 1: Statistics of P and K

P	1	0.95	0.9
Occurrences	27	11	12
K	0.05~1	0.1(not included)~0.2	0.2(not included)~0.5
Occurrences	8	33	9

For the value of K, the most occurrences appears between 0.1~0.2, which is used in the calculations further on.

When we set P=1.0 and K=0.15, and left out the points who’s ACH<0.4, and get a figure:

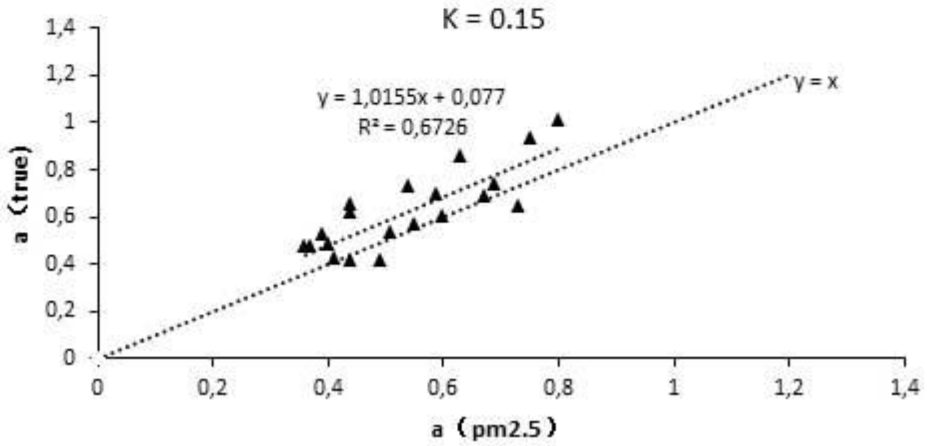


Figure 6 Comparison of ACH from the tracer gas and PM2.5 method.

The result is acceptable for most users. In the 21 points, six of them are located close to y=x, others stay parallel with the standard line, and show a good linear relationship.
But other values of K give not so good relevance, as shown in Fig. 7 .

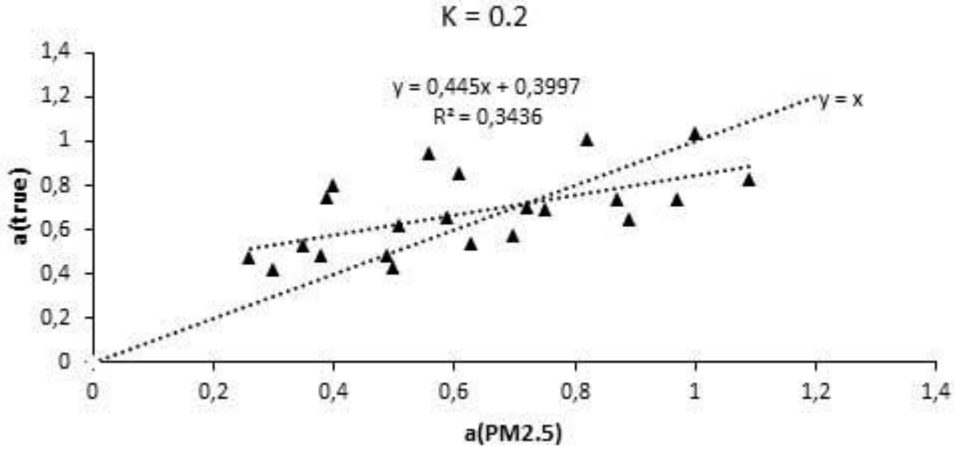


Figure 7:K=0.2

After analyzing, it was decided to use K= 0.15, and P = 1.0, so the final equation can written as follow:

$$V\frac{dC_{in}}{dt} = aVC_{out} - aVC_{in} - 0.15VC_{in} + S \qquad (8)$$

And the calculation formula becomes:

$$C_{in,t} = \frac{a}{0.15+a} C_{out}\left(1 - e^{-(0.15+a)\Delta t}\right) + C_{in,t-\Delta t} e^{-(a+0.15)\Delta t} \qquad (9)$$

Iterative methods can get a solution with certain programs. The suitable “a” is between 0.4 and 1.0. By measuring the concentration of PM2.5 both indoors and outdoors, we can get conceivable ACH.

4 DISCUSSION

The field experiment of the ventilation rate measurement can have several deviations, because K is decided by different reasons, which are too complex. And the influence of K on a is not linear. In most occasions, the airflow is always changing so the deposition rate changes. Also, the concentration of PM2.5 in the surroundings can’t be controlled easily by people. Thus, when it is windy or rainy this method is limited, because the concentration will show a sharp decline.
According to the statistics of Beijing Municipal Environmental Protection Bureau, the result can be used 30% days of a year in Beijing area to get the approximate ACH.
Other people can use this formula to get ACH without producing tracer gas on purpose when windows open only a little (AHC between 0.4 and 1.0). This will help a lot with financial and physical reasons.
This experiment only uses one value of K. Future research can add more parameters influencing it, by including , the roughness of indoor surface. Then K can be decided more accurately.

5 CONCLUSIONS

People are searching for different methods to know more about our surroundings, including the natural ventilation rate, so PM2.5 is considered. The concentration of PM2.5 indoors and outdoors was used to get ACH, compared with a traditional tracer gas method. The relevance between these two methods was compared, and a new formula with constant K and P was found. Thus, the formula can be used to measure ACH directly with the help of high concentrations of PM2.5, especially occasions with windows slightly open.
In most developing industrial cities China, PM2.5 is especially outstanding issue. With the high concentration of PM2.5 in such area like Beijing, it can be a free source of tracer source to use anytime for great convenience. This method is also easy to use, due to the easy measure steps.

REFERENCE:

Beijing Municipal Environmental Protection Bureau(2014) announce the days of heavy pollution.
Stavova P. (2011) “Experimental Evaluation of Ventilation in Dwellings by Tracer Gas CO2”. Czech Technical University, Czech Republic.
Chun Chen, Bin Zhao (2011) “Review of relationship between indoor and outdoor particles: I/O ratio, infiltration factor and penetration factor”. Atmospheric Environment 45 (2011) 275 e288
Cheng, P. L. and X. F. Li (2014). "Air Change Rate Measurements using Tracer Gas Carbon Dioxide from Dry Ice." International Journal of Ventilation 13(3): 235-245.
Heidt FD, Werner H. (1986) “Microcomputer-aided measurement of air change rates”. Energy Bldgs.’, Vol 9.
Roulet C-A, Vandaele L. (1911). “Air flow patterns within buildings – measurement techniques”. Technical Note AIVC 34
A.C.K. Laia, W.W. Nazaroff (2005) “Supermicron particle deposition from turbulent chamber flow onto smooth and rough vertical surfaces”. Atmospheric Environment 39 (2005) 4893 – 4900
Christopherm.Long ,Helenh.Sun （2001）”Using Time-and Size-Resolved Particulate Data To Quantify Indoor Penetration and Deposition Behavior”. Environ . Sci. Technol. 2001 , 35, 2089 - 2099.
O.O. Ha¨nninena, , E. Lebretb, V. I1acquaa, K. Katsouyannic,N. Ku¨nzlid,f, R.J. Sra´me, M. Jantunena （2004）”Infiltration of ambient PM2.5 and levels of indoor generated non-ETS PM2.5 in residences of four European cities”

MITIGATING OCCUPANT EXPOSURE TO $PM_{2.5}$S EMITTED BY COOKING IN HIGH OCCUPANCY DWELLINGS USING NATURAL VENTILATION STRATEGIES

Catherine O'Leary[1], Sally Lofthouse[1], and Benjamin Jones[*1]

1 Department of Architecture and Built Environment
Lenton Firs House, University of Nottingham
NG7 2RD, United Kingdom
**Corresponding author: benjamin.jones@nottingham.ac.uk*

ABSTRACT

The long term exposure to fine particulate matter with a diameter of ≤2.5 μm ($PM_{2.5}$) is linked to numerous health problems, including chronic respiratory and cardiovascular diseases, and cancer. In dwellings, a primary emission source of $PM_{2.5}$ is cooking, an activity conducted several times per day in most households. People spend over 90% of their time indoors and more time in their homes than any other type of building. Therefore, they are at risk of exposure to elevated levels of $PM_{2.5}$ emitted by cooking if these particles are not removed at source. This is particularly important in dwellings with a high occupancy density where cooking periods are longer than average. In the UK, overcrowding is a significant issue with 1.1 million (5%) of the 23.4m households in England and Wales considered to be overcrowded. This issue affects families the most because more than 66% of overcrowded households have dependent children. Here, the consequences of $PM_{2.5}$ exposure could be significant for two key reasons: (i) children breathe greater volumes of air relative to their body mass than adults, and (ii) their tissue and organs are still growing.

This paper uses theoretical and semi-empirical modelling approaches to investigate the concentrations of $PM_{2.5}$s emitted by cooking in UK dwellings of high occupancy density. Firstly, the effects of increasing the rate of $PM_{2.5}$ emission from cooking in proportion to occupancy density are investigated in an archetypal dwelling located in London where there is evidence of elevated levels of overcrowding that can lead to and poor indoor air quality. It is assumed that the number of cooking activities increases with the number of occupants and that they are consecutive, thus increasing the cooking period while keeping the emission rate constant. The results predict that the average weighted $PM_{2.5}$ concentration emitted by cooking is approximately constant as the number of occupants increases. Although academically interesting, this highlights the need to identify realistic cooking schedules. Therefore a second approach records IAQ parameters and occupant behaviour over a 12-day period in a high occupancy student house in Nottingham. These data inform a pollutant transport model that predicts temporal variations of $PM_{2.5}$ concentrations attributable to cooking. The observed occupant behaviour is predicted to lead to daily weighted concentrations well in excess of the WHO limit. To mitigate the exposure risk, it is found that kitchen windows should be left open for a minimum of 20 minutes after cooking to reduce pollutant concentration to safe levels.

KEYWORDS

Overcrowding, health, ventilation, indoor air quality, kitchen.

1 INTRODUCTION

A household is defined as one or more persons sharing living accommodation (a living room or a sitting room) and who are not necessarily related by blood or marriage (ONS, 2013). In

England and Wales 1.1 million (5%) of its 23.4 million households are considered to be overcrowded (ONS, 2014). It is an increasing problem exacerbated by a shortage of suitable and affordable housing (CIH *et al.*, 2012) and there are examples of Rackmanism (Butler, 2015), the practice of charging extortionate rents for inferior properties, especially to poor, disadvantaged or immigrant tenants. The occupants of over-occupied dwellings are subjected to a range of pollutants, diseases, and social situations that can negatively affect their health and wellbeing, academic performance, and economic productivity. These increase the burden on national health care and reduce economic output.

The UK's definition of overcrowding is enshrined in law. The *Room Standard* dates from 1935 (HM Government, 1985) and divides the number of occupants by the number of rooms, but excludes bathrooms, toilets, halls, landings and storage spaces. Accordingly, a kitchen is considered acceptable sleeping accommodation. An overcrowded household is defined as one with more than one person per room (ppr). Many housing authorities use a modern measure of overcrowding (CIH *et al.*, 2012) known as the *Bedroom Standard*, which requires a separate bedroom for each married or cohabiting couple, adult aged over 21 years, pair of adolescents aged 10-20 years of the same gender, and pair of children under 10 years regardless of gender. Thus it defines an overcrowded household as one lacking one or more bedrooms and so it is more likely to consider a household overcrowded than the Room Standard.

In the UK, people spend over 70% of their time inside their own homes (Lader *et al.*, 2001). Buildings contain airborne pollutants emitted internally by building material and furnishings and by activities undertaken in the building, and also those emitted externally. There is a growing awareness that some bio-accumulating semi-volatile organic compounds and ultrafine particles may negatively affect the health of occupants (Logue *et al.*, 2011) and so building air quality is a cause for concern. The most dangerous pollutant is estimated to be particulate matter with a diameter of ≤2.5μm ($PM_{2.5}$) (Logue *et al.*, 2011). The particles are small enough to bypass biological defences and are linked to chronic respiratory and cardiovascular diseases, and cancer (Lewtas *et al.*, 2007). The WHO (2006) recommends mean maximum $PM_{2.5}$ concentrations of $25\mu g/m^3$ per day and $10\mu g/m^3$ per year. In the USA the inhalation of $PM_{2.5}$s in dwellings is estimated to be responsible for approximately 1000 disability adjusted life years lost per year per 100,000 people (Logue *et al.*, 2011). Primary $PM_{2.5}$ sources in dwellings are cooking and tobacco smoking, but also candle and incense burning, open and stove fires, gas fires, hobs and ovens, and scented oil devices. Following deposition, $PM_{2.5}$s are re-suspended by vacuuming, dusting, and sweeping. Of these sources, cooking is of particular interest because it is an essential activity undertaken in most dwellings.

Elevated risks of lung cancer, particularly in women, are associated with the use of solid cooking fuels (Lissowska *et al.*, 2005) as well as emissions from foods being cooked (Wang *et al.*, 2009). Cooking using traditional Woks in kitchens without extractor hoods is associated with increased lung cancer risk for non-smoking Taiwanese women (Ko *et al.*, 1997). There is an increase in the rate of emission of $PM_{2.5}$s when comparing the use of gas cookers to electric, and when cooking meats rather than vegetables (Dennekamp *et al.*, 2001). In the UK approximately 30% of UK households use gas ovens and 70% use electric, whereas around 40% use electric hobs and 60% use gas (BRE, 2013). Dennekamp shows that they are both associated with $PM_{2.5}$ emissions and so their contribution to poor IAQ and associated negative health effects should be considered. The UK's statutory Approved Document F (ADF) (HM Government, 2010) prescribes intermittent ventilation rates of 30l/s adjacent to the hob (using a hood) or 60l/s elsewhere, or a continuous rate of 13l/s. In new dwellings these are a requirement, whereas existing dwellings only have to maintain ventilation systems present when its kitchen is refurbished.

It is likely that the scale of the cooking activity increases with the size of the household and so the risk of exposure to cooking related $PM_{2.5}$s could be higher in overcrowded dwellings. Accordingly, this paper uses theoretical and semi-empirical modelling approaches to investigate the concentrations of $PM_{2.5}$s emitted by cooking in UK dwellings of high occupancy. Firstly, the effects of increasing the rate of $PM_{2.5}$ emission from cooking in proportion to occupancy density are investigated theoretically using an archetypal dwelling located in London where there is emerging evidence of a relationship between overcrowding, indoor air quality (IAQ), and negative health effects (Kyle, 2011). Secondly, IAQ parameters and occupant behaviour are recorded over a 12-day period in a student house in Nottingham. These data inform a pollutant transport model that predicts temporal concentrations of $PM_{2.5}$s attributable to cooking. Together they ask the question: is there a risk of elevated $PM_{2.5}$ concentrations in high occupancy dwellings attributable to cooking?

2 METHOD

To explore indoor $PM_{2.5}$ concentrations in over-occupied UK dwellings, a validated multi-zone ventilation and pollutant transport tool known as CONTAM is used (Walton & Dols, 2013). Both the theoretical (see Section 2.1) and semi-empirical (see Section 2.2) approaches make identical modelling assumptions in some notable areas. Air exchange between the dwelling and its external environment is assumed to occur through air leakage paths (ALPs) located in facades exposed to the external environment and via open windows. All openings assume one-way-flow and a power law relationship between the airflow rates and the pressure differences across them. The windows are assumed to be sharp-edged and have a flow exponent of n=0.5 whereas the ALPs have n=0.65 (Sherman & Dickerhoff, 1998). All ALP parameters follow Jones *et al*. (2013, 2015). An ALP is located at the floor and ceiling height of each external façade of each zone. Adjacent dwellings are assumed to experience identical environmental conditions and so air is not exchanged between them. Bespoke MATLAB code (MathWorks, 2013) processes the CONTAM predictions to determine weighted daily and annual averages of $PM_{2.5}$ concentrations that can be compared against WHO guidelines.

2.1 Theoretical Approach

Table 1: Dimensions of rooms and whole dwelling (Oikonomou *et al*., 2012)

	Hall	**Stair (x2)**	**Kitchen**	**Living**	**Toilet**	**Storage**	**Bathroom**	**Bed 1**	**Bed 2**
Area (m^2)	6.4	9.9	16.7	29.7	2.5	1.9	8.1	23.0	22.2
Volume (m^3)	17.8	27.7	46.7	83.2	7.0	5.3	22.8	64.5	62.2
Dwelling envelope area (m^2): 317.2				Permeability ($m^3/h/m^2$): 12.1			Volume (m^3): 364.6		

Overcrowding is most common in one and two bedroom homes (Shelter, 2005). Accordingly, CONTAM is used to model the $PM_{2.5}$ concentrations in an archetypal two-bedroom terraced house (Oikonomou, 2012) whose dimensions are given in Table 1. The house is first modelled as a single zone and then disaggregated into zones, where each zone represents a room. This is to examine the effect of open plan living on occupant exposure and to investigate the importance of a multi-zone approach.

Internal doors are assumed to remain closed and so are modelled as cracks with length 760mm and width 10mm (HM Government, 2010). Cooking emission rates and schedules follow Shrubsole *et al.* (2012). Households in over-occupied dwellings are likely to comprise multiple groups who each cook separately and so cooking is assumed to be a consecutive activity rather than a concurrent activity. The ambient $PM_{2.5}$ concentration is 13μg/m^3. $PM_{2.5}$s are emitted during cooking at a rate of 1.6mg/min and have a deposition rate of 0.39h^{-1} in each zone. Cooking commences at the same time each day, regardless of the occupancy level, but the duration increases with occupancy. For lunch and dinner we assume 15 minutes plus 5 minutes per occupant, and for breakfast 7.5 minutes plus 2.5 minutes per occupant. The internal air temperature is 21.1˚C for ground floor zones, following Jones *et al.* (2015) and 18˚C for first floor zones following (CIBSE, 2006). The internal air temperature of the single zone model is 21.1˚C. The CIBSE Test Reference Year (TRY) (CISBE, 2002), a synthesised typical year of weather data for London, provides external air temperatures and wind velocity inputs. The dwelling is considered to be in an urban location and the wind speed is scaled appropriately by CONTAM. Wind pressure coefficients are estimated using an algorithm (Jones *et al.*, 2015) that gives a normalized average wind pressure coefficient for long-walled low-rise dwellings, which is a function of the angle of incidence of the wind and local sheltering. General periods of occupancy are given in Table 2 and room patterns for cook and non-cook occupants in the multi-zone model are given in Table 3.

Table 2: Building occupancy periods (hours)

	Occupied Times	
Weekday	0000-0820	1740-2400
Weekend	0000-2400	

Table 3: Indoor occupancy patterns (% time)

	Cook		Non-Cook	
Room	**WD**	**WE**	**WD**	**WE**
Living	56%	40%	62%	58%
Kitchen	8%	10%		
Bed 1	36%	50%		
Bed 2			38%	42%

WD, week day; WE, weekend.

2.2 Semi-Empirical Approach

Table 4: Dimensions of rooms and dwelling

	Kitchen	Living Room	Bedroom	Hallway	Dwelling
Average Height (m)	2.6	2.6	2.6	2.6	
Width (m)	2.7	3.5	3.5	1.0	
Length (m)	6.5	3.7	4.0	7.7	
Floor area (m^2)	17.2	13.0	14.0	7.3	51.3
Volume (m^3)	48.2	32.4	35.0	18.3	133.9
Surface area (m^2)					131.6

A high occupancy dwelling located in Nottingham is investigated during the heating season to investigate its internal $PM_{2.5}$ concentrations over time. Both occupant behaviour and internal and external air temperatures were recorded during a 12 day period from 27th October to 16th November 2014. These data are used to inform a multi-zone CONTAM model that predicts $PM_{2.5}$ concentrations in each room.

The dwelling is a typical 1930s red brick terrace whose geometry is given in Table 4. Average room heights are given because they vary. The kitchen and living room dimensions are given separately but the rooms comprise a single open plan space. The dwelling's air permeability is estimated to be $13m^3/h/m^2$ based on its age and condition.

The kitchen contains an electric oven and 4 gas rings and occupants recorded their use. Each appliance is assumed to generate $PM_{2.5}$ at a rate of 1.6mg/min (Shrubsole *et al.*, 2012). A maximum of 3 appliances were used concurrently over the cooking period and so the maximum generation rate used is 4.8mg/min. The ambient $PM_{2.5}$ concentration is $12\mu g/m^3$ (DEFRA, 2014) and each zone contains a sink with a deposition rate of $0.39h^{-1}$ (Shrubsole *et al.*, 2012).

The use of ventilation was recorded. The kitchen contains a fan above the cooker, two small windows, a large window and a door. None of these were used during monitoring and so a *closed window* scenario provides an appropriate baseline. For comparison, three other scenarios are considered: *partially open*, where the small windows are open 50% during cooking; *fully open*, where one small window and the large window are open 100% during cooking; and *always open*, where all windows are 100% open all of the time.

Air temperatures were recorded at 1 minute intervals with dataloggers (Onset, 2015) that have a precision and error at 25˚C of 0.1˚C and ±0.4˚C, respectively, and averaged for each hour of the day. The internal temperature was recorded at head height in the middle of the kitchen away from heat sources. Appropriate wind velocity data is taken from the CIBSE TRY for Nottingham (CIBSE, 2002). Temperatures for all other rooms are assumed: 14˚C for the bedroom and 12˚C in the hallway. The recorded occupancy schedule is aggregated to produce a typical occupancy schedule, see Table 5. Occupants spend time in the kitchen/living area when cooking and to eat and socialise. The remainder of their time spent is spent in their bedroom or out of the house, see Table 6.

Table 5: Dwelling occupancy (hours)

	Occupied from	Occupied until
WD	0000	1000
	1600	2400
WE	0000	1200
	1500	2100
	2400	2400

WD, week day; WE, weekend.

Table 6: Indoor occupancy patterns (% time)

Room	WD	WE
Kitchen	20%	40%
Bedroom	80%	60%

WD, week day; WE, weekend.

3 RESULTS

3.1 Theoretical Approach

The predicted weighted annual mean exposures for the open plan and multi-zone models are given in Table 7. They show that the annual mean total exposures for the open plan (OP) scenario and for non-cooks in the multi-zone (MZ) scenario increase linearly with occupancy, but that this increase is minimal, and that all remain below the WHO maximum annual mean concentration of 10μg/m^3. Although this is academically interesting, it highlights the need to identify more realistic cooking schedules. The mean concentrations are higher for occupants in the OP scenario than for non-cooks in the MZ scenario. This suggests increased risks of $PM_{2.5}$ exposure for those in open plan dwellings and shows the importance of considering building geometry and occupancy patterns. Most notably, cooks in the MZ scenario are exposed to much higher $PM_{2.5}$ concentrations, which increase with occupancy and exceed the WHO guideline at all occupancy levels.

Table 7 also shows the number of days per year where the predicted weighted internal $PM_{2.5}$ concentrations exceed the WHO 24-hour mean guideline of 25μg/m^3. The data reflect the annual mean exposures and show that the OP and MZ non-cook scenarios never have a daily mean that exceeds 25μg/m^3. However, the number of days where the daily mean exceeds 25μg/m^3 increases non-linearly with occupancy for the MZ cook scenario reaching 311 days when the household comprises 12 occupants.

Both metrics show increased risks for cooks but not to occupants who spend no time in the kitchen (see Table 3). This suggests a need for additional kitchen ventilation to remove contaminants at source. CONTAM assumes uniform air within a zone (Walton & Dols, 2013), however, in reality the distribution of $PM_{2.5}$ is likely to be non-uniform and so actual occupant exposures could vary. The external $PM_{2.5}$ concentration is assumed to be constant and is considered to be both the default concentration for all simulations and the concentration of $PM_{2.5}$s to which occupants are exposed when they are away from the house.

Table 7: Weighted annual mean total exposure and days exceeding 24hr Mean guideline total exposure 25μg/m^3

Number of Occupants	Weighted annual mean exposure (μg/m^3)			Days exceeding 24hr Mean Guideline		
	Open Plan	Cook	Non-Cook	Open Plan	Cook	Non-Cook
3	9.00	14.73	4.95	0	0	0
6	9.06	19.44	5.02	0	2	0
9	9.28	24.27	5.09	0	104	0
12	9.63	28.92	5.15	0	311	0

3.2 Semi-Empirical Approach

The observations of occupancy in the monitored house show that its occupants are most likely to cook alone. This is significant because it provides support for the consecutive cooking behaviour used to develop the theoretical models of $PM_{2.5}$ concentrations in Section 3.1. The behaviour also has a significant impact on the generation of $PM_{2.5}$s in the house. UK households cook for an average of 34 minutes per day (min/day) (Jackson, 2014) whereas each occupant of the Nottingham House cooks a meal separately. During the twelve days of monitoring, the mean cooking time was 126.3min/day, nearly 4 times the household average.
The weighted mean total $PM_{2.5}$ concentrations over the 12-day period for each scenario are given in Table 8. When compared against the WHO annual mean guideline of 10µg/m^3 all scenarios are shown to exceed it, except the always open scenario. However, the extrapolation of 12 days of data to a whole year is problematic because weather, occupant behaviour, and the probability of the use of ventilation openings will all vary throughout the year effecting airflow rates and $PM_{2.5}$ concentrations within the building.
The weighted daily mean $PM_{2.5}$ concentrations over the 12-day period are given in Table 9. The WHO 24-hour guideline of 25µg/m^3 is exceeded on 11 of the 12 days by the *closed* scenario, whereas this is reduced to 4 days by the *partially open* scenario. This highlights that the provision of a small amount of additional purpose-provided ventilation can significantly lower occupant exposures to $PM_{2.5}$. The WHO guideline is only exceeded once by the *fully open* scenario coinciding with the longest recorded cooking time. The WHO guideline is never exceeded by the *always open* scenario.

Table 8: Whole period average concentration by scenario (µg/m^3)

Scenario	Total Weighted Mean
Always Closed	52.67
Partially Open during cooking	21.95
Fully Open during cooking	16.14
Always Open	9.05

Table 9: Weighted daily mean exposure [November 2014] (µg/m^3)

Day	Scenario			
	Always Closed	**Partially Open during cooking**	**Fully Open during cooking**	**Always Open**
5	14.49	11.23	10.20	7.20
6	38.83	25.63	16.55	9.14
7	68.59	35.16	24.40	11.10
8	59.75	30.55	19.54	10.26
9	102.14	45.21	24.06	14.11
10	41.97	17.95	11.97	7.57
11	51.15	15.85	9.83	7.30
12	64.29	12.93	8.50	7.48
13	76.78	20.25	27.40	8.62
14	41.06	19.18	19.46	7.16
15	41.97	17.84	12.86	9.72
16	31.23	11.71	8.98	8.93
Days >25µg/m^3	11	4	1	0

4 DISCUSSION

The paper uses a modelling approach to investigate the risk of elevated $PM_{2.5}$ concentrations in high occupancy dwellings. Although care has been taken to use model inputs with provenance this has not always been possible and so there is great uncertainty in the predictions.
Das *et al*. (2014) use a two-zone CONTAM model of an apartment and a probabilistic framework to estimate the variation in indoor $PM_{2.5}$ concentrations attributable to cooking. A sensitivity analysis shows that they are most sensitive (in order of significance) to window opening area, $PM_{2.5}$ generation rate, and IAT. It is reassuring that the window area and IAT were monitored and used as empirical inputs to the semi-empirical model. However, a probabilistic approach would have helped to establish uncertainty in assertions made here.
The emission rate of $PM_{2.5}$ is likely to be a function of many parameters, such as the make, model and cleanliness of the cooker and hobs, and the food types (Lewtas, 2007). However, these variables were not recorded and so it is impossible to quantify uncertainty in the predicted

concentrations. Only a modest number of studies in the literature present measurements of $PM_{2.5}$ emission rates from cooking. They are given as mean values and so, in the absence of further information, they are assumed to be constant or Gaussian when a standard deviation is given. There is a need for probability density functions of emission rates so that uncertainty in the mean can be evaluated. It is also likely that emission rates of different cooking activities change with time and so there is a clear need for data that reflects this. A new measurement approach and the creation of a large dataset are required.

A constant deposition rate is assumed for $PM_{2.5}$ from cooking and external sources. However, recent research suggests lower deposition rates that vary according to their source may be appropriate (Shrubsole *et al.*, 2015). These would concurrently increase mean indoor $PM_{2.5}$ concentrations and occupant exposure.

Constant external $PM_{2.5}$ concentrations are assumed for the theoretical and semi-empirical models. Urban $PM_{2.5}$ concentrations are higher than those in rural areas and are affected by traffic. Annual concentrations vary considerably; for example, annual means varied between 12μg/m^3 and 22μg/m^3 across Greater London for the period 2008 to 2015 (DEFRA, 2015). They are also likely to vary over short periods of time and so a static value represents a source of uncertainty.

Neither the theoretical nor the semi-empirical models consider the use of a kitchen extractor fans. The semi-empirical model used observed occupant behaviour to justify this omission, whereas the theoretical model is used to investigate a worst case scenario. Not all UK houses have kitchen extractor fans and there is no evidence that they function correctly when present; for example, many are recirculating types that require high-efficiency filters to improve IAQ. The IAQ in UK domestic kitchens is currently unknown and requires further investigation. Here, it is important to identify the appropriateness of the ventilation rates currently required in domestic kitchens and the potential dangers associated with high or over-occupancy of dwellings. If cooking behaviour is consecutive, as modelled here, then a prescribed ventilation rate may suffice. Concurrent cooking could lead to $PM_{2.5}$ concentrations that are significantly higher than those predicted here.

Section 3.2 shows that natural ventilation can effectively reduce concentrations of $PM_{2.5}$. A time-step analysis of the *always closed* scenario shows that $PM_{2.5}$ concentrations only return to ambient levels after many hours. In the *partially* and *fully open window* scenarios, the time taken is 2.5 and 1.5 hours, respectively, and just 10-20 minutes for the *always open* scenario. This suggests that windows should be left open for a minimum of 20 minutes after cooking. However, natural ventilation offers no control over the direction of airflow and so pollutants can spread to living areas. The concurrent use of kitchen extractor fans and windows could increase $PM_{2.5}$ in living spaces and an investigation is required. A ventilation strategy that minimizes occupant exposure to $PM_{2.5}$ must simultaneously extract cooking related pollutants at source and limit airflow out of the kitchen into other zones using depressurization.

Internal $PM_{2.5}$ concentrations occasionally fell below the ambient concentration. This can appear counterintuitive but occurs when the $PM_{2.5}$ deposition rate is greater than both their indoor emission rate and the rate at which they are brought into the dwelling via ventilation. Occupants who spend time in the spaces with lower $PM_{2.5}$ concentrations have a reduced mean exposure to cooking related $PM_{2.5}$ and associated health risks.

The occupancy patterns in Section 2 assume that the occupants do not occupy their dwelling between 0820-1740hrs. However, income poor households, or those with three or more children, are much more likely to be living in persistently overcrowded accommodation (Barnes et al., 2011) and so the occupancy patterns used here may not be appropriate. Changes to sleeping patterns observed in many overcrowded households, where they become disrupted and irregular, or where household members frequently sleep in the living room, kitchen, or hall (Shelter, 2005) were not examined. Additionally, window opening behaviour is assumed to occur year-round and does not consider variations in IAT or security. Internal doors are assumed to remain closed, which may prevent the spread of pollutants between zones. The semi-empirical model uses occupancy patterns that are simplified and aggregated models of observed daily routines. However, the presence of each occupant varied and so their individual exposures will be different to those predicted here.

Shrubsole *et al*. (2012) consider 2010 and 2050 exposures of occupants to $PM_{2.5}$ in London dwellings. The 2010 scenario finds average total exposures of 28.4μg/m^3, 60.5μg/m^3, and 15.5μg/m^3 for the whole household, cook, and non-cook, respectively. When compared to the semi-empirical model the household average is similar to the 21.95μg/m^3 period average for the partially open scenario. The theoretical model predicts annual averages of 14.73μg/m^3 and 4.95μg/m^3 are predicted for a cook and non-cook, respectively, when there are 3 occupants, which are very different to those of Shrubsole. This emphasises the importance of model inputs, such as occupancy schedules, window areas, and building geometry. However, both studies agree that cooks are exposed to higher $PM_{2.5}$ concentrations than non-cooks.

Fabian *et al*. (2012) predict annual mean concentrations of 27.8μg/m^3 for households in dwellings without an extractor fan, and 10.9μg/m^3 when they are installed. This compares well to 21.95μg/m^3 predicted by the semi-empirical model, but we note that they are higher than the theoretical model averages. This emphasises the potential benefits of local extraction.

There is a growing concern about high external $PM_{2.5}$ concentrations in London, which had an annual mean of 14.9μg/m^3 in 2014 (DEFRA, 2014). However, this is significantly lower than some of the annual mean indoor $PM_{2.5}$ concentrations predicted here and by others. People spend the vast majority of their time inside buildings and so air quality should be a cause for concern for policy makers who seek to minimize its negative health effects and the consequent burden on national health care and tax payers. Furthermore, Section 3 shows that over-occupied dwellings could have $PM_{2.5}$ concentrations that are significantly above those of households that conform to the Bedroom Standard. More than 66% of overcrowded households have dependent children and so families are affected the most. Children breathe greater volumes of air relative to their body mass than adults and their tissue and organs are still growing. The negative health consequences attributable to exposure to elevated levels of $PM_{2.5}$ could last a lifetime, bringing emotional, educational, and financial costs for both individuals and society. This highlights the importance of effective ventilation, and the need for ventilation strategies that adequately and efficiently remove harmful pollutants from any dwelling, particularly those that are over-occupied.

5 CONCLUSIONS

This paper uses theoretical and semi-empirical approaches to investigate the risk of elevated $PM_{2.5}$ concentrations in high occupancy dwellings attributable to cooking. Firstly, an archetypal dwelling located in London is used to investigate the effects of increasing the rate of $PM_{2.5}$ emission from cooking in proportion to occupancy density. It is assumed that the number of cooking activities increases with the number of occupants and that they are consecutive, thus increasing the cooking period while keeping the emission rate constant. It is predicted that the average weighted $PM_{2.5}$ concentration attributable to cooking is approximately constant as the number of occupants increases. A comparison of the predicted $PM_{2.5}$ concentrations in open and zoned representations of the dwelling shows that non-cooks are exposed to higher concentrations when the building is open plan, although they are below WHO annual maximum levels. The zonal approach further increases the risk to cooks because $PM_{2.5}$ concentrations are above WHO annual maximum levels when there are 9 occupants.

Secondly, IAQ parameters and occupant behaviour are recorded over a 12-day period in a naturally ventilated high occupancy house in Nottingham where cooking behaviour is indeed found to be consecutive. These data inform a pollutant transport model that predicts temporal variation of $PM_{2.5}$ concentrations attributable to cooking. The observed occupant behaviour is predicted to lead to daily weighted concentrations well in excess of the WHO limit. A low ventilation rate through available windows during, but not beyond, the cooking period is shown to reduce daily mean concentrations to safe concentrations after 1.5-2.5 hours. However, to mitigate the risk of exposure to $PM_{2.5}$ it is found that kitchen windows should be left wide open during cooking and for a minimum of 20 minutes after cooking to reduce pollutant concentration to safe levels.

Overcrowded and high occupancy dwellings represent more than 5% of UK households and there is there is emerging evidence of a relationship between overcrowding, IAQ, and negative health effects. Accordingly, the cooking related IAQ in these dwellings should be a cause for concern for policy makers who seek to minimize national health care costs and increase national economic output.

REFERENCES

Barnes M, *et al.* (2011) The Duration of Bad Housing and Children's Well-being in Britain. Housing Studies. 26:155-76.

BRE. (2013) Report 9: Domestic appliances, cooking & cooling equipment. Building Research Establishment, UK.

Butler, P. (2015) Housing raid finds 26 people living in three-bedroom east London home. *The Guardian*: 25th June.

CIBSE. (2002). Guide J - Weather, Solar and Illuminance data. London, CIBSE Publications.

CIBSE. (2006) Guide A - Environmental Design. 7 edition. London: CIBSE Publications.

CIH, NHF, Shelter. (2012) The Housing Report, edition 2. The Chartered Institute of Housing, National Housing Federation, and Shelter.

Das P, *et al.* (2014) Using probabilistic sampling-based sensitivity analyses for indoor air quality modelling. Building and Environment. 78:171-82.

DCLG. (2012) Allocation of accommodation: guidance for local housing authorities in England. Department of Communities and Local Government.

DEFRA. [Viewed November 8, 2014] http://uk-air.defra.gov.uk/data/exceedence?

Dennekamp M, *et al*. (2001) Ultrafine particles and nitrogen oxides generated by gas and electric cooking. Occupational and Environmental Medicine.58:511-6.

Fabian P, *et al.* (2014) Simulating indoor concentrations of NO_2 and $PM_{2.5}$ in multifamily housing for use in health-based intervention modelling. Indoor Air. 22(1):12-23.

H.M. Government. (2010) The Building Regulations 2000 (2010 edition): Approved Document F: Ventilation. Crown Copyright 2010.

H.M. Government. (1985) The Housing Act 1985, Section 324. Crown Copyright 1985.

Jackson P. (2015) Recent trends in UK food consumption. University of Sheffield.

Jones BM, *et al.* (2013) The Effect of Party Wall Permeability on Estimations of Infiltration from Air Leakage. International Journal of Ventilation.12:17-29.

Jones B, *et al.* (2015) Assessing Uncertainty in Housing Stock Infiltration Rates and Associated Heat Loss: English and UK Case Studies. Building and Environment. 92:645-56.

Ko YC, *et al.* (1997) Risk factors for primary lung cancer among non-smoking women in Taiwan. International Journal of Epidemiology.26:24-31.

Kyle RG, *et al.* (2011) Childhood disadvantage and emergency admission rates for common presentations in London. Archives of Disease in Childhood. 96:221-6.

Lader D, Short S, Gershuny J. (2006) The time use survey, 2005: how we spend our time. Office for National Statistics, London, UK.

Lewtas, J. (2007). Air pollution combustion emissions. Mutation Research/Reviews in Mutation Research, 636(13):95-133.

Lissowska J, *et al.* (2005) Lung Cancer and Indoor Pollution from Heating and Cooking with Solid Fuels. American Journal of Epidemiology. 162:326-33.

Logue JM, *et al.* (2011) A Method to Estimate the Chronic Health Impact of Air Pollutants in U.S. Residences. Environmental Health Perspectives. 120:216-22.

MathWorks (2014). MATLAB Version 8.4.0.150421 (R2014b). The MathWorks Inc.

Oikonomou E, *et al.* (2012) Modelling the relative importance of the urban heat island and the thermal quality of dwellings for overheating in London. Building & Environment. 57:223-38.

ONS. (2013) Households, families and people (General Lifestyle Survey Overview - a report on the 2011 General Lifestyle Survey). Office for National Statistics, UK.

ONS. (2014) Overcrowding and under-occupation in England and Wales. Office for National Statistics, UK.

Onset. (2015) HOBO U10-001 data logger. http://www.onsetcomp.com

Shelter. (2005) How overcrowded housing affects families. Shelter. London, UK.

Sherman MH, Dickerhoff DJ. (1998) Airtightness of U.S. dwellings. ASHRAE Transactions. 104:1359-67.

Shrubsole C, *et al.* (2012) Indoor PM2.5 exposure in London's domestic stock. Atmospheric Environment. 62:336-43.

Walton GN, Dols WS. (2013) CONTAMW 3.1 User Guide and Program Documentation, NISTIR 7251. Gaithersburg, MD: National Institute of Standards and Technology.

Wang X-R, *et al.* (2009) The roles of smoking and cooking emissions in lung cancer risk among Chinese women in Hong Kong. Annals of Oncology. 20:746-51.

WHO. (2005) WHO Air quality guidelines for particulate matter, ozone, nitrogen dioxide and sulfur dioxide. Word Health Organization.

VENTILATION EFFECTIVENESS COMPARISON BETWEEN EXTRACT VENTILATION AND BALANCED VENTILATION IN A SCALE MODEL

Bart Cremers

Zehnder Group Nederland
Lingenstraat 2
8028 PM Zwolle, The Netherlands
bart.cremers@zehndergroup.com

ABSTRACT

The differences between extract ventilation and balanced ventilation are subject of many discussions in sales markets where both solutions have their share. Often, the differences are marked in terms of energy, because balanced ventilation is normally accompanied by heat recovery. But there is another difference in terms of the ventilation effectiveness of the system.

This document reports experiments in a scale model of a house showing the difference between extract ventilation and balanced ventilation in ventilation effectiveness, and therefore in achievable indoor air quality. The ventilation effectiveness is measured in terms of the cleaning time, i.e. the time it takes for smoke to be completely extracted from individual rooms.

The results indicate that for an undisturbed (design) system the cleaning time for individual rooms is independent on the ventilation system. But in disturbed situations like an open window or wind pressure on the building, the cleaning time is different for various individual rooms, and dependent on the ventilation system.

The conclusion is that for balanced ventilation, the ventilation effectiveness is not reduced by occupant behavior or wind conditions. On the other hand, for extract ventilation the ventilation effectiveness is lower in particular individual rooms as a result of these disturbances.

KEYWORDS

ventilation effectiveness, extract ventilation, balanced ventilation

1 INTRODUCTION

The differences between mechanical extract ventilation (MEV) and mechanical ventilation with heat recovery (MVHR) are subject of many discussions especially in the Dutch, Belgian and French markets where both solutions have their share. Often, the differences in terms of energy are marked. It is well known that heat recovery saves about 80-90% of the heating demand caused by the fresh air entering a building in the heating season.

But there is another difference in terms of indoor air quality, caused by the ventilation effectiveness of the system. In a monitoring campaign, these influences have been investigated thoroughly by Van Holsteijn et al (2015). This paper presents results of the differences in ventilation effectiveness concluded from scale experiments in a miniature house.

2 METHOD

For the scale experiments, the miniature house from fig. 1 was used. The house is 65 cm wide, 70 cm tall, and 25 cm deep. At the first floor, a master bedroom was modelled on the left, and a child's bedroom on the right, separated by a hallway. The hallway has three doors, of which two are leading to the master bedroom and child's bedroom. The third door at the back leads to a cavity at the back side of the house, modelling a bathroom. All three doors are made with a small slit underneath the door to allow cross flow between rooms even when the doors are closed.

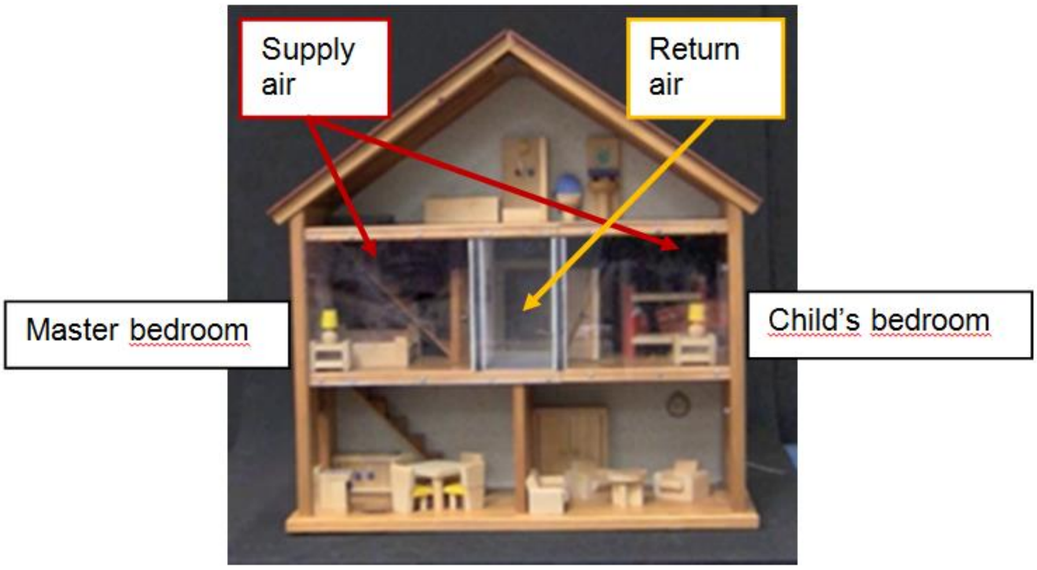


Figure 1. The miniature model used in the scale experiments.

The entire first floor has been closed air tight with glass. At the sides of the house, both in master bedroom and in child's bedroom are rectangular windows of 8 cm x 8 cm that can be closed or opened using tape. A small hole in the tape could be made to allow natural air passage (modelling a window grille).

Two identical small axial fans with a diameter of 75 mm are mounted at the sides of the cavity at the back side of the house. One of the fans (the 'return' fan) extracts air from the house via the bathroom. A second fan (the 'supply' fan) brings 'fresh' air into the building via two plastic ducts with internal diameter of 5 mm. One duct ends in the master bedroom and one duct ends in the child's bedroom. The ends of both ducts stick out of the back wall of the rooms for about 2 cm and are located 1 cm from the ceiling and 7 cm from the exterior wall.

The following special arrangements have been taken in preparation of the experiments. All of these arrangements are necessary in the scale model as well as in the real world for a normal building.

1. In order to have an air tight house, the cracks in interior and exterior walls were sealed.
2. In order to have balance between return air and supply air, the return flow was partly obstructed to match the resistance of the supply flow.
3. In order to have equal supply air volume in master bedroom and child's bedroom, the lengths of the supply ducts have been made equal to match each other's resistance.

The ventilation effectiveness of a ventilation system under various conditions has been investigated by injecting smoke to the master bedroom and the child's bedroom. Just after the bedrooms have been filled with smoke, the ventilation system is started. For MEV, only the return fan is started, and for MVHR both fans are started. Figures are shown for three conditions with MEV on the left of the figure and MVHR on the right of the figure.

The first condition is without disturbance, the second condition is with open window in the master bedroom and the third condition is with wind pressure on the façade of the building where the master bedroom located. The ventilation effectiveness of a system is expressed with the so-called "cleaning time", defined as the time it takes for a room to be fully clean (i.e. without smoke).

The exact amount of the air flow rates are unknown. But we can make an estimation of the used ventilation rate in the scale model. In real world ventilation, a typical cleaning time for an individual room is about 4 hours. In the scale experiments, the cleaning time of the undisturbed condition is about 10 minutes. This means that the ventilation rate in the scale experiments is about 24 times as high as in real life.

Note that the term MVHR is used in this document to indicate a system with mechanical supply of fresh air as opposed to natural supply of fresh air. However, in the scale model, there is no heat exchange taking place because the energy implication are out of scope for these experiments.

3 RESULTS AND DISCUSSION

Condition 1: Without disturbance

The first comparison is made without disturbance, i.e. the situation as intended in the design phase of a ventilation system. Figure 2 shows that after the start of the ventilation system, stale air in the bedrooms is replaced for fresh air. For MEV the fresh air is entering via the window grills and for MVHR the fresh air is entering via the supply tubes.

The cleaning time for both bedrooms and for both ventilation systems are all equal. It can be concluded that, without disturbance, the cleaning time depends only on the fresh air flow rate.

The smoke can be observed to disappear as a layer of smoke on the floor that is decreasing in height as time goes by. Although the ventilation rate in the scale experiments is high, it appears that the smoke is not mixed in a uniform way in a bedroom. This is likely to be caused by the high density of the smoke compared to the density of clean air. In real life, the mixing of fresh air with stale air in the room is much more apparent.

a) 

b) 

c) 

Figure 2. Frames of the situation without disturbance 1 minute (a), 3 minutes (b), and 10 minutes (c) after start of the ventilation system. In every frame left is the house with MEV and right is the house with MVHR.

Condition 2: Open window in the master bedroom

The second comparison is made for occupant behavior during a typical winter night situation. For both MEV and MVHR, the parents leave the window half open to allow fresh air to enter the bedroom (on top of the fresh air brought by the ventilation system). For the MEV case, the window grille in the child's bedroom is closed to avoid direct draught and the internal door is closed to ensure a silent, good sleep for the child.

Figure 3 shows that the cleaning time in the master bedroom is decreased because the half open window is increasing the amount of fresh air coming into the room. In the child's room the closed window grille is obstructing the incoming flow of fresh air and consequently the cleaning time increases with respect to the intended situation in the design phase.

For MVHR, the cleaning time of the child's room is still equal to the cleaning time of the undisturbed condition. It means that the supply of fresh air into a room never gets smaller by the opening of windows or doors in other parts of the house.

a) 

b) 

c) 

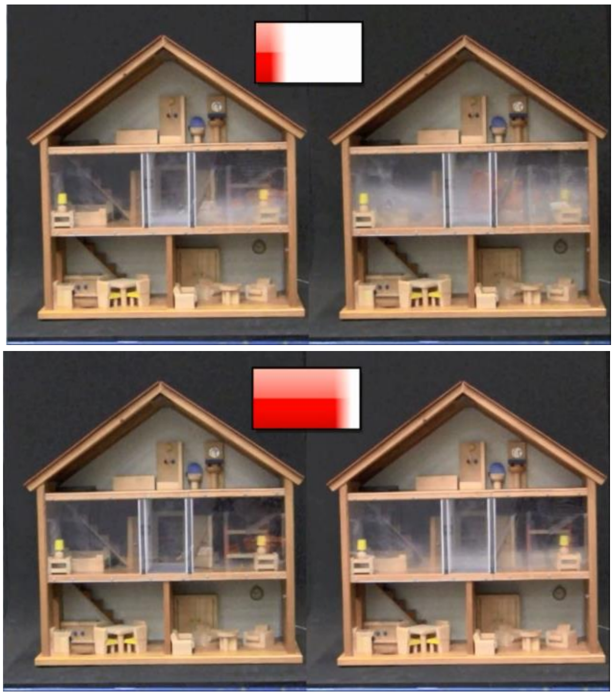
Figure 3. Frames of the situation with an open window in the master bedroom 1 minute (a), 3 minutes (b), and 10 minutes (c) after start of the ventilation system. In every frame left is the house with MEV and right is the house with MVHR.

Condition 3: Wind pressure on the master bedroom

The third condition is the occurrence of wind, with the master bedroom at the windward side and the child's bedroom at the lee side. In the scale model, the wind is made by a table fan placed approximately 2 meters from the scale house. For MEV, all window grilles are open. For both ventilation systems all windows are closed and all internal doors are halfway open.

Figure 4 shows that for MEV, the open window grilles at the windward side and the lee side allow a cross flow through the house. Stale air from the master bedroom is flowing to the child's bedroom. Figure 4 also shows that for MEV the natural air flow through the window grille is taking place in the reversed direction (!) as intended in the design phase, so that indoor air with smoke flows out of the house via the grille. Unlike a typical night, in the scale model there is no continuous source of contamination (CO_2, moisture, etc.) in the master bedroom. However, one can conclude from the figures that he cross flow through a house with MEV causes an increased fresh air supply in rooms at the windward side and a decreased fresh air supply in rooms at the lee side of the house. The cross flow also has its effect on draught experiences in the master bedroom and on the entire energy consumption to heat the building but these energy implication are beyond the scope of these experiments.

a) 



b)

c)

Figure 4. Frames of the situation with wind attack on the master bedroom 1 minute (a), 3 minutes (b), and 10 minutes (c) after start of the ventilation system. In every frame left is the house with MEV and right is the house with MVHR.

4 SUMMARY OF RESULTS

The comparison between MEV and MVHR can best be seen in table 1. This table summarizes the cleaning time in the child's room for the experiments with the scale model. The cleaning time is used as a measure of the ventilation effectiveness, with obviously cleaning time decreasing when ventilation effectiveness for an individual room is increased.

Table 1. Cleaning time of the child's bedroom.

	MEV	**MVHR**
No disturbance	8 minutes	8 minutes
Open window in master bedroom	20 minutes	8 minutes
Wind on master bedroom	1 minute	8 minutes

The cleaning time of a room is increased by an open window elsewhere in the house. Contrary to people's belief that windows may not be opened for an MVHR ventilation system, it is shown that the ventilation effectiveness of an MEV system may be negatively influenced by open windows somewhere in the house.

The cleaning time of a room at the lee side of the building is seemingly decreased by the wind in the scale experiments. However, with a continuous source of contamination elsewhere in the house, the room at the lee side does not get a supply of fresh air via the window grille, but is filled with stale air from elsewhere in the house. Because of the air tight envelope of a house with MVHR, the cleaning time of both rooms is not influenced at all by wind around the building.

In real life the number of conditions is much larger. Depending on the outside temperature and wind conditions, and on the specific position of grilles, windows, internal/external doors, the driving force and the air flow paths change in a drastic way. With natural supply of air as in

MEV ventilation systems, the corresponding ventilation effectiveness for an individual room is largely influenced by the combination of all these conditions.

The conditions as shown in these scale experiments are merely chosen as examples of the negative influences they can have. The real life effect of all these conditions can be seen in large monitoring campaigns as in Van Holsteijn et al (2015).

5 CONCLUSIONS

For an air tight house with MVHR (mechanical extract and mechanical supply), the ventilation effectiveness is neither changed by the occupants' behavior nor by wind effects around the building. The mechanical supply in a room is therefore necessary to maintain a continuous supply of fresh air into the room. The conclusion is that the mechanical supply in MVHR systems (as opposed to the natural supply from MEV systems) lead to a constant indoor air quality regardless of occupants' behavior or wind effects around a house. Another advantage is the obvious energy saving by the heat recovery of MVHR ventilation systems.

6 REFERENCES

Van Holsteijn RCA, Li WL, Valk HJJ et al (2015) Improving the energy- and IAQ performance of ventilation systems in Dutch residential dwellings. Proceedings of the International Conference Healthy Buildings 2015 Europe, 18-20 May 2015, Eindhoven

NUMERICAL EVALUATION OF THE IMPACT OF HEMP LIME CONCRETE MOISTURE-BUFFERING CAPACITY ON THE BEHAVIOUR OF RELATIVE HUMIDITY SENSITIVE VENTILATION SYSTEM

Bassam Moujalled[*1], Driss Samri[1], Laurent Guldner[2], Fabrice Richieri[3] and Emma Stephan[1]

1 Cerema
Rue Pierre Ramond, CS60013
33166 Saint-Médard-en-Jalles Cedex, France
**bassam.moujalled@cerema.fr*

2 DGALN
Tour Séquoia
92055 La Défense Cedex, France

3 SNIA
Avenue Pythagore, BP 70285
33697 Mérignac Cedex, France

ABSTRACT

Hemp Lime concrete (HLC) is a bio-based material, which knows currently a growing development. HLC is a low embodied energy material. It has an excellent moisture buffer performance and is considered as good indoor climate regulators. Recent field study has confirmed the ability of HLC to maintain hygrothermal conditions at winter and summer comfort levels.

On the other side, relative humidity sensitive ventilation systems help to save building heat energy by reducing the amount of the exhaust airflow during unoccupied periods or low activities depending on the level on the indoor relative humidity. Previous studies showed the ability of such systems to save energy without compromising IAQ throughout the whole year. They are widely used in new French residences. However, the moisture-buffering capacity of bio-based material such as HLC can modify the behaviour of humidity sensitive ventilation as it lessens the variations of indoor relative humidity.

The objective of this paper is to assess the impact of the moisture buffering capacity of hemp lime concrete on the behaviour of humidity sensitive ventilation system in single detached dwellings. A numerical approach based on the energy simulation tool TrnSys coupled to the multi-zone air-flow and contaminant transport model COMIS was used. A real case of HLC dwelling was studied. First, measured indoor relative humidity is compared to the simulated results in order to assess the moisture buffer model of Trnsys. Second, humidity sensitive ventilation system is added in order to analyse the dynamic interaction between the moisture buffering capacity of HLC and the performance of humidity sensitive ventilation system in terms of energy efficiency and indoor air quality. Results show that in the case of humidity sensitive ventilation, the moisture buffer capacity of hemp lime concrete helps to maintain the indoor relative humidity within the range of comfort zone between 40% and 60% through the whole year. They confirm that the use of moisture-buffering materials is a very efficient way to reduce the amplitude of daily moisture variation. However, the yearly average exhaust airflow is slightly higher, yet the heat load increase is less than 5%.

KEYWORDS

Coupled heat and air-flow simulation, Relative-humidity-sensitive (RHS), Humidity control, Moisture-buffering, Energy

1 INTRODUCTION

Hemp Lime concrete (HLC) is a bio-based material, which knows currently a growing development. HLC is a low embodied energy material. It has an excellent moisture buffer performance and is considered as good indoor climate regulators (Collet, 2012). Recent field study (Moujalled, 2015) has confirmed the ability of HLC to maintain hygrothermal conditions at winter and summer comfort levels while outside temperature and relative humidity (RH) daily variations are up to 15°C and 50% respectively. It has confirmed the global idea that bio-based materials are good indoor climate regulators. The measurements inside walls showed a high thermal inertia, which allowed them to dampen the daily variations by 90% and to delay the effects of peak values up to 11 hours.

On the other side, relative humidity sensitive (RHS) ventilation systems help to save building heat energy by reducing the amount of the exhaust airflow during unoccupied periods or low activities depending on the level on the indoor relative humidity. They are widely used in new French residences as they help to meet the requirements of the latest thermal regulation. Previous studies showed the ability of such systems to save energy without compromising IAQ throughout the whole year. Woloszyn (Woloszyn, 2009) has demonstrated that the combined effect of ventilation and wood as buffering material can help to keep the indoor RH at a very stable level, between 43% and 59%. Tran Le (Tran Le, 2010) has compared different ventilation strategies (with constant and variable ventilation rates) and he found that the use of a RHS ventilation strategy with hemp concrete can reduce energy consumption about 15% because of the decreasing of cold fresh air flowing into the room.

However, the RHS ventilation systems are designed for non-hygroscopic building materials. When used with hygroscopic materials such as HLC, the ventilation system operation can be modified by the moisture-buffering capacity of HLC as it lessens the variations of indoor relative humidity. Thus the RHS ventilation system would not function properly as designed and the building risks being under or over-ventilated.

The objective of this paper is to assess the impact of the moisture buffering capacity of hemp lime concrete on the dynamic behaviour of humidity sensitive ventilation system in single detached dwellings through a numerical approach based on the energy simulation tool TRNSYS coupled to the multi-zone air-flow and contaminant transport model COMIS.

2 METHOD

A numerical approach based on the energy simulation tool TRNSYS coupled to the multi-zone air-flow and contaminant transport model COMIS was used. A real case of HLC dwelling was studied. First, measured indoor relative humidity is compared to the simulated results in order to assess the moisture buffer model of TRNSYS. Second, humidity sensitive ventilation system is added in order to analyse the dynamic interaction between the moisture buffering capacity of HLC and the performance of RHS ventilation system in terms of energy efficiency and indoor air quality.

2.1 Studied building

The simulation case is based on a real 2-floor single-detached dwelling that hosts 5 people and has a surface of 250 m^2 (Figure 1). It is located close to Perigueux in South West to France. Its envelope is made of 30cm thick HLC sprayed into walls of timber frame structure. HLC is also

used in roof and intermediate floor in 10cm and 15cm thickness respectively. The walls are internally and externally protected with lime-sand plasters. The composition of the walls is presented in Table 1.
A pellet boiler coupled with 12.6 m² of solar collectors provides energy for heating and domestic hot water. The heat is distributed in the house through radiant floor at the ground level and radiant walls at the top level. A balanced ventilation system with heat recovery is used for air renewal. The total air renewal is 0.3 h^{-1}.
The building was monitored between February and December 2012. Temperature and Relative Humidity (RH) were measured in each room with 15 minutes time step.

Figure 1: View of the south façade of the building

Table 1: Wall composition

Wall	Composition	U value
External Wall	3 cm lime sand plaster + 30 cm HLC + 2 cm lime sand plaster	0.28 W/(m^2.K)
Ceiling	2 cm OSB + 10 cm HLC + 21 cm cellulose wadding + 2cm roof tile	0.15 W/(m^2.K)
Floor	4 cm concrete + 7cm Polyurethane + 4cm concrete	0.26 W/(m^2.K)
Windows	Low emissivity double-glazing with wooden frame	1.1 W/(m^2.K)

Mechanical RHS ventilation is studied following technical agreement specifications (CSTB, 2009). It is composed of humidity sensitive air-inlets in main rooms and humidity-controlled air-outlets in service rooms. Besides, it enables boosting airflow rate in the kitchen (135 m^3/h) during cooking and in the toilets (30 m^3/h). Table 2 presents the characteristics of the RHS ventilation system.

Table 2: RHS ventilation system characteristics

System	Inlets		Outlets		
	Living room	Bedrooms/office	Kitchen	Bathroom	Toilet
RHS	2 x (6-45 m^3/h) at RH 45-60%	6-45 m^3/h at RH 45-60%	10-45 m^3/h at RH 24-59% (135 m^3/h; 30’)	10-40 m^3/h at RH 36-66%	5 m^3/h (30 m^3/h; 30’)

Moisture vapour and sensible heat schedules were created for each zone according to the activities of a 5-person family (2 adults and 3 children) - activities related to human metabolism

and domestic. For each schedule, the amount of water vapour and sensible heat were calculated hour by hour based on the values given by (Richieri, 2013).
Dwelling is supposed to be heated from 1st October to 20th May with an 20.4°C ambient temperature during occupied period (18h-9h on weekdays and all the week-end). Set point temperature is 3°C reduced during unoccupied period (weekdays).

2.2 Simulation tools

The energy simulation tool TRNSYS is coupled to the multi-zone air-flow and contaminant transport model COMIS. This coupling allows the calculations of heat transfer, airflow and pollutant transport (e.g., moisture) in a multi-zone building under transient boundary conditions. Besides, it enables dynamic modelling of closed-loop control for different systems (especially RHS ventilation system) and humidity buffering of porous materials. Figure 2 presents the coupling between TRNSYS and COMIS.

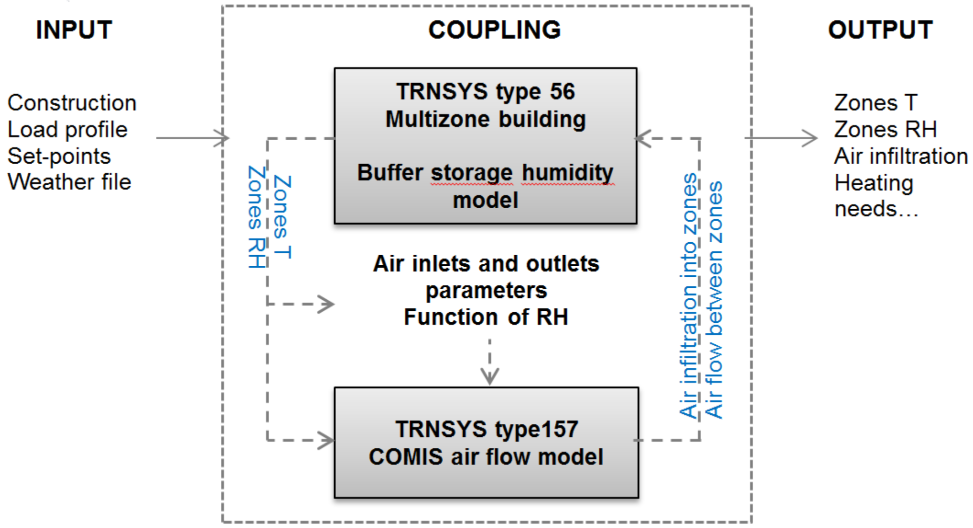


Figure 2: Overview of coupling between TRNSYS and COMIS-model

2.2.1 Air flow model

The building is considered as a zone network linked by airflow components such as cracks on envelope, inlets and outlets of ventilation systems, and internal circulations between 8 zones representing the different rooms of the dwelling. For each main room, inlets are located in accordance with plans, positioned at 2.2 m height. Specific behaviour of humidity sensitive inlets is modelled as defined in Table 2.
Leakages are distributed over the 338 m² building envelope (A_{Tbat}). In each zone, two cracks have been defined on external walls at 0.63 m and 1.88 m height, and one crack on the ceiling at 2.5 m height as described in (Richieri, 2013). The cracks characteristics are as follows:

- fixed 0.65-flow exponent,
- flow coefficient calculated from the global building air permeability $Q_{4pa\text{-}surf}$ (0.6 $m^3/(h.m^2)$) proportionally to the ratio of the façade area to the total envelope area A_{Tbat}.

The zones are interconnected by 90 cm wide doorways considered closed during simulations. To ensure air circulation from dry to wet rooms, each doorway is considered with a 1 cm undercut for all inner doors. The door undercut is modelled as a large crack with 0.5 flow exponent.
The Cp values are considered in accordance with the AIVC guide on ventilation for the case of shielding conditions “surrounded by obstructions” (Liddament, 1996).

2.2.2 Buffer storage humidity model

In TRNSYS two models are available for the calculation of the moisture balance. The first, called effective capacitance humidity model (EC), is a simplified model which considers sorption effects with an enlarged moisture capacity of the zone air. The second, called buffer storage humidity model (BS), is a more sophisticated model which offers a surface and a deep moisture buffer in the walls of the zone (SEL, 2007). A short description of the equations, assumptions and limitations of both models are given by (Steeman, 2010) and (Woloszyn, Kalamees, 2009). The buffer storage model is based on the effective moisture penetration depth model and is divided into a surface and a deep layer to account for both short-term and long-term exchanges, e.g. daily and yearly variations. The moisture penetration depth can be calculated from the material properties and the cycle of the periodic humidity variation as given in (Steeman, 2010). It is in the order of several millimetres for daily variations and centimetres for yearly variations. Each layer is characterised by the following parameters:

- ξ : gradient of sorptive isothermal line of surface/deep buffer [kg_{water} /$kg_{material}$ /RH],
- d : penetration depth of surface/deep buffer [m],
- β : the exchange coefficient between zone and surface storage for the surface buffer, and between storage and deep storage for the deep buffer [kg/(h.m²)].

Table 3 shows the values of the parameters for the HLC of the studied case. The gradient of sorptive isothermal line is calculated from the sorption isotherms curves for HLC given by (Tran Le, 2010). The penetration depths are calculated with a cycle of one day for the surface buffer and 365 days for the deep buffer as indicated above.

Table 3: Defintion of the parameters of the buffer storage humidity model

	ξ	**d [mm]**	**β [kg/(h.m²)]**
Surface storage	0.06	7.3	3
Deep storage	0.06	140.2	1

3 RESULTS

3.1.1 Comparison of simulation models with measurement results

In the first step, the simulation tool was compared against experimental data in order to evaluate the two moisture balance models of TRNSYS. Indoor relative humidity was calculated with the effective capacitance model (without buffering) and buffer storage model (with buffering) using a balanced ventilation system as in the real case.

Figure 3 shows an example of indoor air RH for a typical week during winter. The measured RH is stable and varies slightly around 40%. The daily variations of the calculated RH are more important with both models, especially for the effective capacitance (EC) model without buffering. The buffer storage model (BS) helps to reduce the amplitude of daily moisture variations and the discrepancy between the calculation and the experiment.

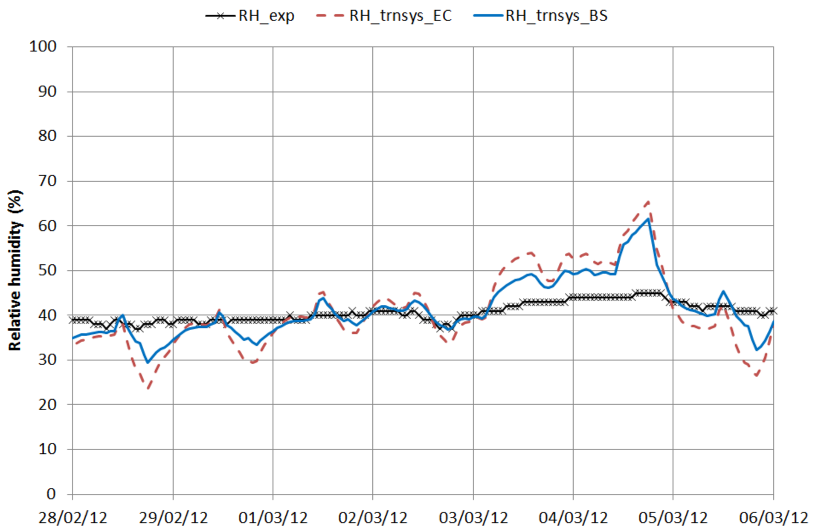
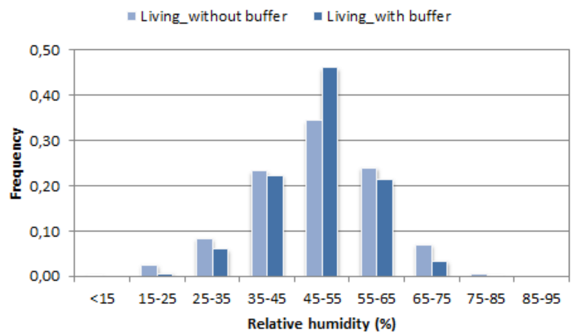


Figure 3: Comparison of measured indoor RH with simulation results for the effective capacitance model without buffering (EC) and the buffer storage model (BS)

3.1.2 Impact of buffer storage humidity on RHS ventilation system operation

In this part, the balanced ventilation system is replaced by RHS ventilation system which is modelled as defined in Table 2. In order to evaluate the impact of buffer storage humidity of HLC, the calculations were made with the EC model (without buffering) and BS model using the parameters as described in Table 3.

Figure 4 compares the variations of calculated RH between both models for the living room and the main bedroom. Globally both models present the same mean values of RH (between 45% and 55%), but the variations of RH with the EC model are more important than the BS model. As expected, the moisture buffering capacity helps to reduce the variation of the indoor RH.



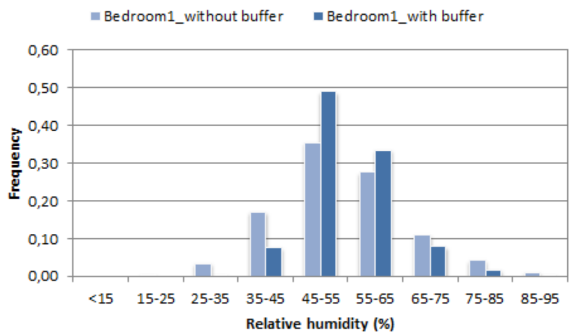


Figure 4 : Comparison of variations of calculated RH for the living room (left) and main bedroom (right)

Figure 5 shows the comparison of the variations of the total airflow rate between the two models. The mean value of the BS model is slightly higher than EC model (99 m^3/h against 97 m^3/h), whereas the extreme values (min and max) are more important for the EC model.

In the case of airflows above 160 m^3/h, the histogram shows that the variations are similar for both models. Indeed the higher airflow rates are imposed by the boosting airflow of the ventilation system and are independent from the humidity.

For the lower values, the airflow rates are more often in the extreme ranges for EC model than BS model. On the contrary, the airflow rates are more often in the middle ranges for BS model than EC model. Indeed, RH values of BS model fall more often within the functioning limits of air inlets and outlets (RH 45-60% for the living room inlet and RH 24-59% for the kitchen outlet) and thus the lowest airflow rate is rarely applied.

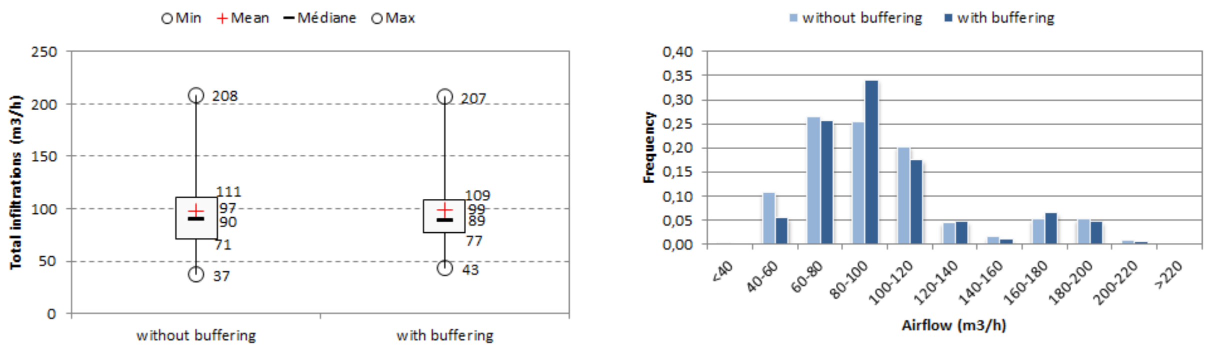


Figure 5: Comparison of total airflow rate between the cases without buffering (effective capacitance model) and with buffering (buffer storage model)

Figure 6 and Figure 7 compare the evolution of RH and airflow rate in the main bedroom between EC and BS models during a typical week in winter. For BS model, the daily variation of RH is about 20% against more than 40% for EC model. This confirms that the moisture buffer capacity of the BS model helps to reduce the spread between the minimum and the maximum values of RH as in real case. Therefore the lowest airflow rate (10 m^3/h) is rarely applied with BS model as shown on Figure 7.

Finally, the heating needs were calculated for both models. For BS model, the heating needs are 16.2 kWh/(m².year) against 15.6 kWh/(m².year) for EC model. The moisture buffer capacity results with a slight increase of heating needs (around 4%) because the slight increase of cold fresh air flowing into the dwelling.

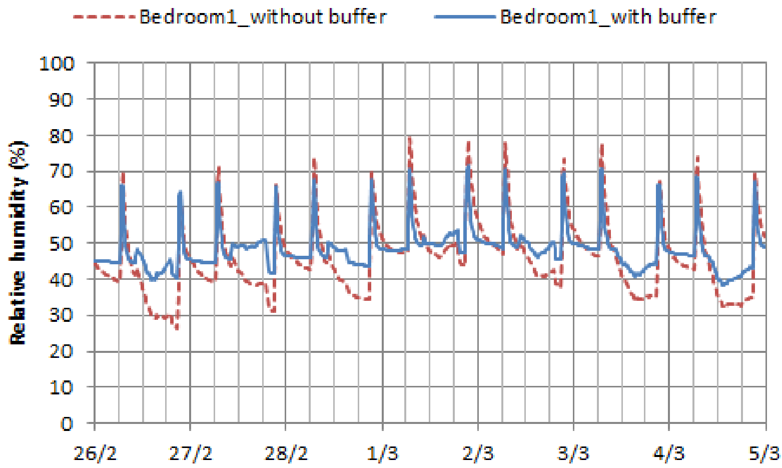


Figure 6: Comparison of the evolution of RH in the main bedroom between the cases without buffering (EC model) and with buffering (BS model)

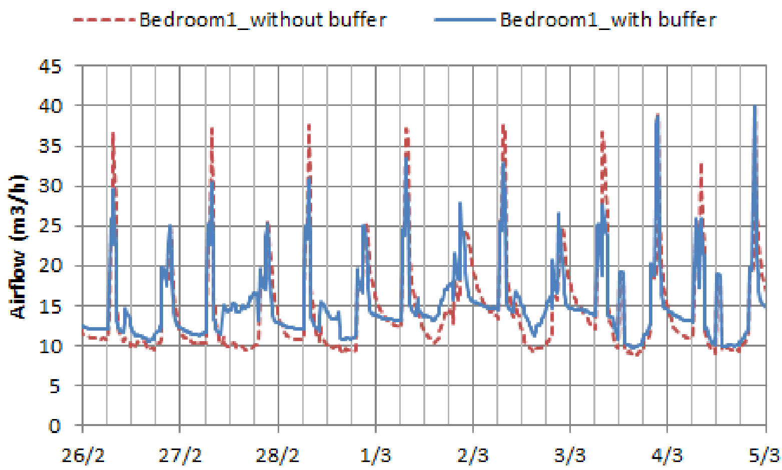


Figure 7: Comparison of the evolution of the airflow rate in the main bedroom between the cases without buffering (EC model) and with buffering (BS model)

4 CONCLUSIONS

A coupled multi-zonal model TRNSYS-COMIS has been developed to study the impact of the moisture buffering capacity of hemp lime concrete on the behaviour of relative humidity sensitive ventilation system in single detached dwellings based on a real case. Results show that the moisture buffer capacity of HLC helps to maintain the indoor relative humidity within the range of comfort zone between 40% and 60% through the whole year. They confirm that the use of moisture-buffering materials is a very efficient way to reduce the amplitude of daily moisture variation. The combination of the moisture buffering of HLC with relative humidity sensitive ventilation system modify slightly the functioning of the ventilation system by increasing moderately the yearly average exhaust airflow, which results by a slight increase of heat losses by air renewal (less than 5%).

5 ACKNOWLEDGEMENTS

The authors want to thank the French Ministry of Sustainable Development for its financial contributions. The sole responsibility for the content of this publication lies with the authors. It does not necessarily reflect the opinion of the Ministry. The authors thank also DB Chanvre Company that provided this construction.

6 REFERENCES

Collet, F., & Pretot, S. (2012). Experimental investigation of moisture buffering capacity of sprayed hemp concrete. *Construction and Building Materials, 36*, 58-65.

CSTB. (2009). *Systèmes de ventilation mécanique hygroréglable FADIS et FRANCE AIR. Avis technique n° 14/07-1194*V200.* Paris: CSTB.

Liddament, M. (1996). *A guide to energy efficient ventilation.* Coventry: AIVC.

Moujalled, B., Aït-Ouméziane, Y., Samri, D., Stéphan, E., Moissette, S., Bart, M., et al. (2015). Experimental and numerical evaluation of the hygrothermal performance of a hemp-lime building. *First International Conference of Bio-based Building Materials, June 22-24 2015, Clermont-Ferrand, France* (pp. 534-541). Bagneux: RILEM Publications.

Richieri, F., Moujalled, B., Samri, D., Bourdassol, M., & Carrie, R. (2013). Numerical evaluation of the airtightness impact on energy needs in mechanically ventilated dwellings. *8th International BUILDAIR-Symposium, June 7-8 2013, Hannover, Germany.*

SEL;TRANSSOLAR;CSTB;TESS. (2007). *TRNSYS 16 – Multizone Building modeling with Type56 and TRNBuild.* Madison, USA: University of Wisconsin.

Steeman, M., Janssens, A., Steeman, H., Van Belleghem, M., & De Paepe, M. (2010). On coupling 1D non-isothermal heat and mass transfer in porous materials with a multizone building energy simulation mode. *Building and Environment, 45*, 865-877.

Tran Le, A., Maalouf, C., Mai, T., Wurtz, T., & Collet, F. (2010). Transient hygrothermal behaviour of a hemp concrete building envelope. *Energy and Buildings, 42*, 1797-1806.

Woloszyn, M., Kalamees, T., Abadie, M., Steeman, M., & Kalagasidis, A. (2009). The effect of combining a relative-humidity-sensitive ventilation system with the moisture-buffering capacity of materials on indoor climate and energy efficiency of buildings. *Building and Environment, 44*, 515-524.

OPTIMAL WINDOW OPENING BASED ON NATURAL VENTILATION MEASUREMENTS

Yue Zhang[1], Xiaofeng Li [*2], Pok L. Cheng [1]

1 Tsinghua University
Dept. Of Building Science, School of Architecture,
Tsinghua University
Beijing, 100084, China

2 Tsinghua University
Dept. Of Building Science, School of Architecture,
Tsinghua University
Beijing, 100084, China
**Corresponding author: xfli@tsinghua.edu.cn*

ABSTRACT

From the energy point of view, buildings should be as tight as possible. But lack of ventilation will result in high level of indoor pollutants, which is harmful for occupants. Numerous studies find that lack of ventilation could cause symptoms for occupants, which are characterized by World Health Organization as Sick Building Syndrome.

There are lots of real-time ventilation data and rational ventilation standards in the world. However, these kinds of measurements lacks in China, so more data is needed. In China, most public schools have neither air condition nor mechanical ventilation, so the main form of ventilation is natural ventilation. Thus, it is crucial to do research in measuring the natural ventilation rate in order to improve the indoor environment.

Natural ventilation is mainly driven by heat buoyancy and wind pressure. The lowest air change rate happens when the wind speeds and temperature differences are low. If the quantity of fresh air can satisfy the ventilation standard under these circumstances, the natural ventilation can satisfy most conditions.

According to climate data, the days in which wind speed is below 0.5m/s accounts for about 26%. Most of the school buildings aren't located in open areas. Furthermore, in consideration of the emergency evacuation for the children, the school buildings are built only with 1-3 floors. So, the disturbances of the surrounding buildings contribute to low wind speeds around the school building.

The time period between cooling and heating seasons usually comes with low indoor and outdoor temperature difference, where most classrooms have open windows for ventilation. It is therefore important to find out whether the ventilation rate is sufficient during this period of time.

The measuring method is based on the release of a stable rate of the tracer gas CO_2 given off by solid CO_2 (dry ice) in an insulated box. In theory, the dry ice will sublime at a constant rate as long as there is sufficient dry ice in the box. Thus, the dry ice sublimation rate should remain constant at any time once the steady state heat transfer condition has been reached.

The results of the measurement can enrich the database of the Chinese current ventilation situation. The early-design-stage predictions of natural ventilation performance in renovated buildings provide reference to the design of natural ventilation systems.

KEYWORDS

Natural ventilation; Tracer gas; Constant injection; Optimal window opening; Energy saving

1 INTRODUCTION

Indoor environment is an integral part of people's lives as we spend most of our lifetime indoors, making ventilation an important role in built environment (Sundell 2004).

Also, ventilation affects heating and cooling seasons, which has a huge impact on energy loads of buildings.

Nowadays, energy saving is becoming a priority in many countries, especially in China. In this case, tightening and insulating of a building envelope as well as ventilation reduction have been the main saving tendencies. Laws and standards on buildings' air-tightness are ratified, and people applied the approaches in most new and renovated buildings.
From the energy point of view, buildings should be as tight as possible. So, buildings have become significantly tighter. But the consequences of these steps, unfortunately, were not always thoroughly considered. For instance, low infiltration rate is a serious problem resulting from the relentless tightening of building envelopes.

On the other hand, the role of ventilation in school buildings plays an important role because the amount of supplied fresh air not only have a direct effect on health, but also indirectly influences the presence and concentrations of allergies' risk factors, such as dampness and mold, home dust mites, environmental tobacco smoke, etc. Young children are also more sensible to most indoor pollutants. (Bornehag, Sundell et al. 2005, Ridley, Pretlove et al. 2006, Kolarik, Naydenov et al. 2008)Furthermore, studies have confirmed that there's strong relationship between study efficiency and supply of fresh air. Thus, sufficient ventilation is essential to peoples' health as well as their study performance (Bakó-Biró, Clements-Croome et al. 2012).

In China, most public schools don't have mechanical ventilation, so the main form of ventilation is through natural ventilation, where people get fresh air from opening windows.
But according to questionnaires collected from primary school teachers, most teachers have the sense of ventilation, but they don't have fixed habits of opening windows, because they have no idea when and how much to open the windows to ensure proper amount of fresh air.
Therefore, a theoretical background for how many windows (based on window area) have to be opened during a natural ventilated classroom for a year is required, in order to maintain a healthy indoor environment. Only by this way, a balanced ventilation rate to ensure both energy saving and good indoor environment is ensured.
In order to know the amount of window openings required, the ventilation rate has to be measured. The minimum required ventilation rate is taken from the national standard. Thus, conducting quantitative analysis upon this indoor environment factor, ventilation rate is introduced. The usual ways to measure the natural ventilation rate includes blower door test、CFD simulation and tracer gas (TG) method. This paper uses field measurements by tracer gas (TG) method to get the ventilation data.
The TG methods can be classified based on the type of control and injection method. Three of the frequently used methods are: decay-, constant injection (emission)-, and long term average method – passive perfluorocarbon tracer gas (PFT) method.
The measurement is designed to find out the relationship between the temperature difference and ventilation rate for the specific classrooms, so plenty of ventilation data under different temperature differences are need. Thus, the decay method isn't a suitable for long-term measurements. Moreover, the low control of taking samples (the sampling time is chosen approximately according to the time of tracer gas injection) decreases the accuracy of the method rapidly during unsteady ventilation rate. In passive perfluorocarbon tracer gas (PFT) method, little amount of tracer gas is needed, the analyses procedure is complicated and recently there are only few laboratories in the world, which can perform PFT analyses.
So when measuring long-term unsteady ventilation rates, the only method possible method is the constant injection method (Sherman 1990, Roulet and Vandaele 1991, McWilliams 2002).
Unlike the heating and cooling season, where the indoor temperature is kept constant, the time period in-between usually comes with low indoor and outdoor temperature difference, where most classrooms have open windows for ventilation. It is therefore necessary to find out the indoor temperature for these periods of time. In order to get the whole years' data of indoor temperature, simulation is applied. The parameters are set based on the real conditions. The temperature difference and ACH relevance was measured during April, with no heating or

cooling present. The results of this measurement give the minimum temperature difference for efficient natural ventilation driven by buoyancy.
The results contribute in enriching the database of the current Chinese ventilation situation. Furthermore, according to the results, under the present situation in renovated buildings, the paper provides a window opening guidance to rectify peoples' behaviour in natural ventilation
And the early-design-stage predictions of natural ventilation performance in renovated buildings could provide reference to the design of natural ventilation systems.

2 METHOD

2.1 Description of test site

One type of classroom, with a total of 4 classrooms, was measured in a primary school, the corridor outside and the inside picture of the classroom are showed in Fig.1 and Fig.2. The school is situated in Beijing, China. The dimensions of the classrooms are showed in Fig. 3, where the room volume is 17 m^3.
The heating season is from Nov.15 to Mar.15, with an indoor temperature about 18 °C; the cooling season is from June to August, where the indoor temperature is mostly maintained at 22-26 °C. The remaining days of the whole year are concluded in the transition season, during which there are no heating and cooling applied in the classrooms.
Classes last from 8:00 to17:00, Mon-Fri, and summer and winter vacation both lasts 7 weeks.
All field experiments were conducted during transition season with no heating or cooling approaches.
All classrooms have no mechanical ventilation, with windows facing both north and south direction. The field measurement was conducted with doors shut and 3 windows on one side widely open.

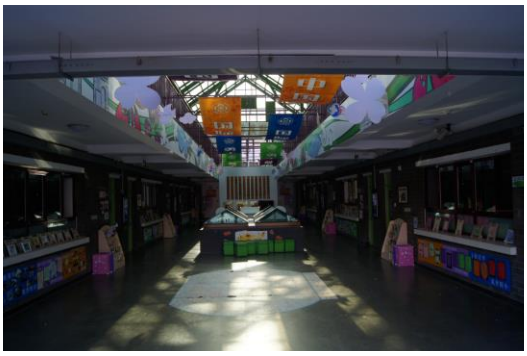


Figure 1: Corridor outside the classrooms.

Figure 2: Indoor picture of a classroom.

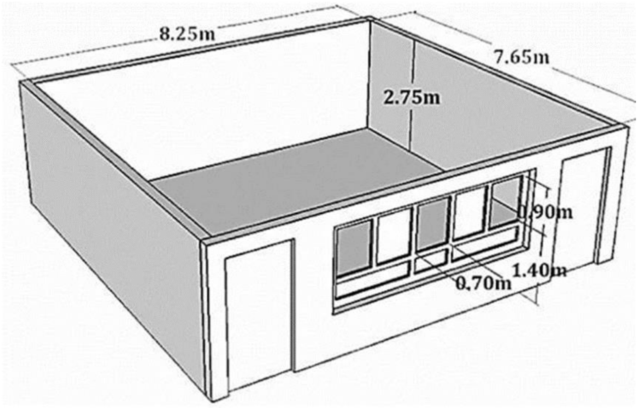


Figure 3: Classroom dimensions.

2.2 Calculation of ventilation rate

Based on the conservation of mass, the relationship can be presented as follows:

$$F + Q \times C_e = V_{zone} \times \frac{dc}{d\tau} + Q \times C \qquad (1)$$

The ACH calculation process for the constant injection method is based on a parametric iteration technique, where the change in the tracer gas concentration between two measurements (indoor and outdoor concentrations) can be expressed as (Šťávová 2012) :

$$\Delta C = \frac{\Delta\tau}{V_{zone}} \times (F - N \times V_{zone} \times (C_1 - C_e)) \qquad (2)$$

ΔC- change of tracer gas concentration
$\Delta\tau$- time interval,
V_{zone}- zone volume,
F – constant tracer gas injection rate,
N- number of ACH,
C_1- indoor tracer gas concentration,
C_e- outdoor tracer gas concentration.

The unknown parameter is N, with all the rest as input values. The calculation was repeated for each consequent time interval to get a theoretical curve. The C_1 is the first concentration measured of the time interval, followed by a theoretical concentration $C_{2,t}$ at the end of the time interval calculated as a sum of the initial concentration (C_1) and the individual step increase ΔC. The calculation process is repeated for every single following time interval, giving a theoretical exponential curve of $C_{2,t}$.

At last, Eq. 2 was applied to calculate the difference between the measured and theoretical values. The Solver tool Add-in in Microsoft Excel was then used for the curve fit to minimize the sum of the errors between the measured and theoretical values, so that the value of N is found.

$$\min\left\{\sum_i Error(C_{i,t})\right\} \rightarrow min\left\{\sum_i (C_1 - C_{i,t})^2\right\} \qquad (3)$$

Index i=0…k, where k is the number of measured concentrations

The ventilation rate, Q, was then calculated with Eq. 4.

$Q = N \times V_{zone}$ (4)

The measured indoor CO_2 concentration was divided up between step-up, constant and decay concentration in order to calculate the ventilation rate. The calculated sums of the errors were small due to this method, and varied between 1-5%, so the curve fit had an error of 1-5%. This calculation method is applicable with a variable indoor tracer gas concentration, when the ventilation rate is unsteady.

2.3 Tracer gas

The tracer chosen for the experiment was CO_2. The dry ice in an insulated box works as a constant injection tracer source, releasing a constant rate of CO_2 (Cheng and Li 2014). The box was rectangular shaped with 5 cm thick polyurethane insulation material, with exit hole at the top, as shown in Fig. 4.

Figure 4: Dry ice box.

Each classroom had one or two boxes with 15 kg dry ice, which was enough to satisfy the indoor tracer gas concentration for 2-3 days. Each box was refilled with dry ice when necessary. The weight loss of the dry ice box was used to calculate the emission rate of CO_2, F_{CO2}, with the following equation:

$$F_{CO_2} = \frac{m \times V_m}{M_m}$$

(5)

CO_2 concentrations were measured with *Telaire* 7001 sensors. The measuring range of the sensors is between 0-2500 ppm with an accuracy of ±5%. The CO_2 concentration was measured with a logging interval of 1 min. Four sensors were positioned inside each classroom at different positions in order to verify that the indoor concentration was uniform. The average was used between the four sensors. Two CO_2 sensors were used to measure the outdoor concentration, positioned in the corridor.

2.4 Temperature

The ventilation rates in the measurement are mainly influenced by the buoyancy effect only. In order to get the indoor and outdoor temperature difference, two temperature sensors were positioned in the centre of the room and one outdoors. The temperature was measured with *Air Temperature and Humidity Meter WSZY-1*, with an accuracy of ±0.1 °C.

2.5 Wind speed

To ensure that natural ventilation is driven by buoyancy pressure regardless of wind pressure, a wind sensor, *Wind Micrometer 5825* with an accuracy of ±0.1m/s, was used to measure the wind speed. The ACH data used are under the condition that the wind speed then is no more than 0.5 m/s. The sensor was positioned in the middle of the corridor, facing towards the outside.

2.6 Simulation of indoor temperatures

Since outdoor temperature can be known from climate data, simulation of indoor temperature is required, in order to get the buoyancy pressure conditions for the entire year.
The indoor temperature can be found through different ways. Due to limited time, this study used simulation.
Regardless of the wind pressure, the simulation used a *Building Information Modeling* software named *DeST (Yan, Xia et al. 2008)*. The building envelope parameters are based on the real-time conditions. Parameters of heat generations are shown in Table 1.

Table 1: Heat generation parameters.

Parameter	Heat generation [W]
People (36 total)	62*
Lights	360
Equipment	0

*Number indicates the heat generated per person.

3 RESULTS

This part shows the results from the field experiment in four classrooms using single-sided ventilation. There were no people, or equipment, nor other disturbances present during the measurements, except two times per day weighting the dry ice which only took half minute per room.

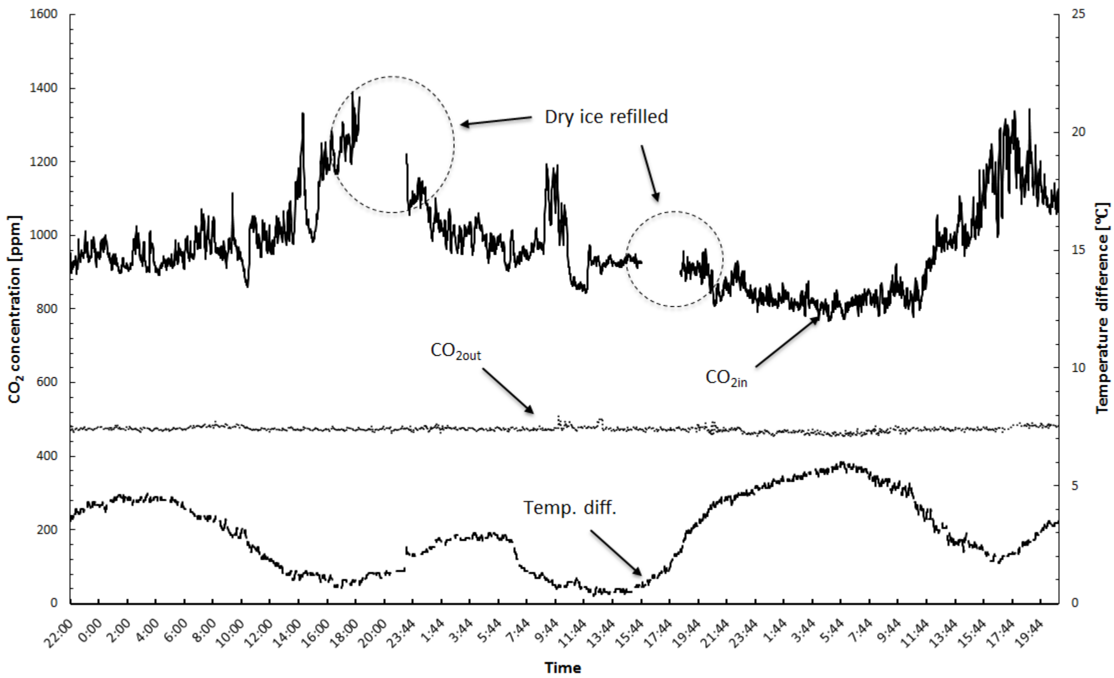


Figure 5: Indoor and outdoor CO_2 concentration and temperature difference.

Fig. 5 shows the data for the calculation part for 3 continuous days. The indoor CO_2 concentration was averaged, because the sensors had a position difference of less than 10 %. For the calculation of ventilation rate, the indoor CO_2 data was divided between: step-up, constant and decay. In order to calculate the ventilation rate with minimal curve fit errors, the time interval was set to 0.5h for the calculation part, which the temperature remains constant during the interval. So the temperature difference during the interval was averaged.

The calculation results are showed in Fig. 6, consisting of over 500 data points.

The horizontal axis represents the temperature gradient $\sqrt{\left(\frac{T_{\mathrm{i}}-T_{\mathrm{o}}}{T_{\mathrm{i}}}\right)}$ and the vertical axis represents the ventilation rate in the classroom.

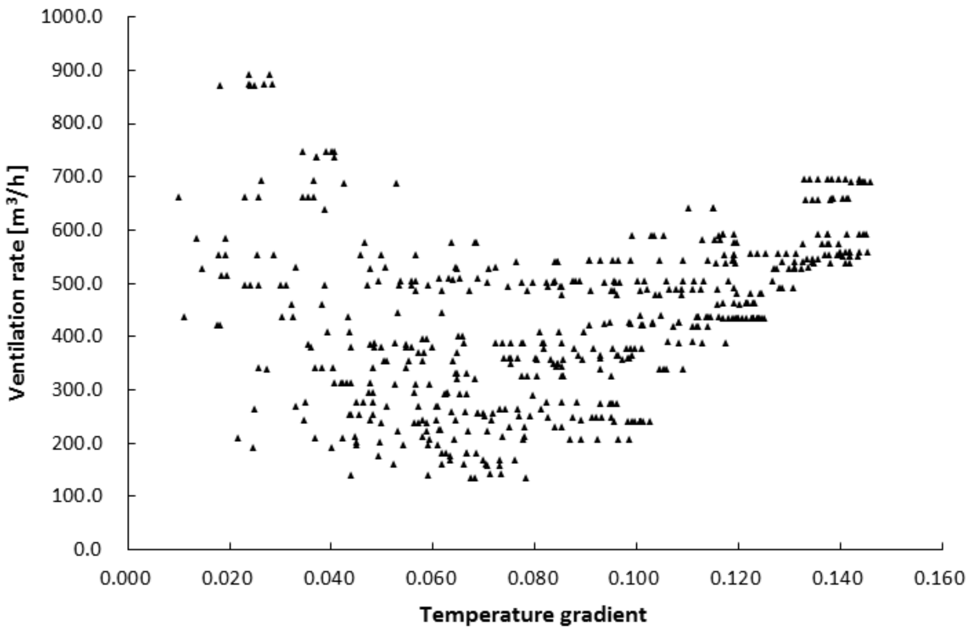


Figure 6: Ventilation rate and temperature gradient comparison.

After excluding the data for wind speeds over 0.5 m/s, the linear relationship is shown in Fig. 7, consisting of over 350 data points.

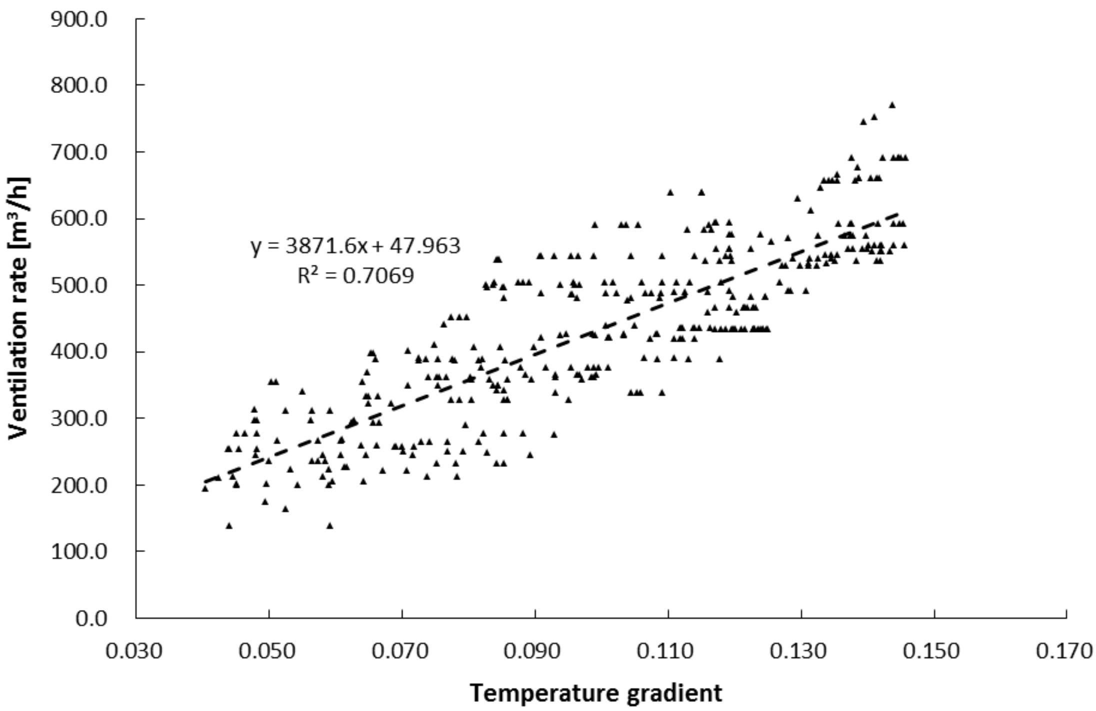


Figure 7: Ventilation rate and temperature gradient comparison.

Comparison of results between the tracer gas method and Bernoulli equation is showed in Fig. 8.

$$Q = \frac{1}{3} C_{\mathrm{D}} A \sqrt{gH \frac{T_{\mathrm{i}}-T_{\mathrm{o}}}{T_{\mathrm{i}}}} \qquad (5)$$

Where discharge coefficient $C_D = 0.7$; area of the window is A=3*0.7*0.9 ; the height of the window is H=0.9 ;
Since most of ventilation rates are between 0.75 and 1.25 theoretical values, the error ratio is considered less than 25%.

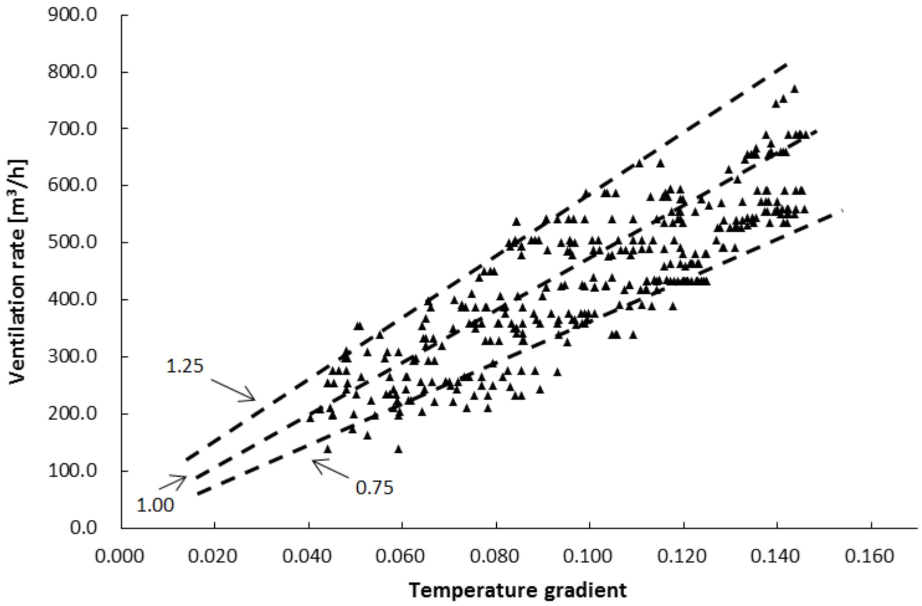


Figure 8: Comparison of measurement results and Bernoulli equation.

Based on an entire year temperature difference from the simulation results during day time, the proper window opening suggestions are given in Table 2.

Table 2: Open-window suggestions

temperature gradient $\sqrt{\left(\frac{T_i - T_o}{T_i}\right)}$	**Suggested number of opened windows**
>0.29	1
0.14-0.29	2
0.09-0.14	3

4 DISCUSSION

The field experiment of the ventilation rate measurement can have several deviations, since there was no artificial mixing. Thus, the uniformity of the tracer gas in the classrooms can’t be 100% ensured. Furthermore, the interval for the ventilation rate calculation is 0.5h, during which, the temperature may change. Lastly, the simulation model can’t represent the real-time indoor temperature of the classroom.
The results enrich the ventilation database in China, and could increase peoples’ attention to their own behavior in opening of windows. Also, the results are available in the early-design-stage of the building design. The predictions of natural ventilation performance in renovated buildings could provide reference to the design of natural ventilation systems.
Others can use the field measurement to obtain the relationship between ventilation rates and temperature under low wind speed. Then find out the temperature inside and out for several temperature differences, for example, 0-5, 10-15 and 20-25°C, where field measurement is also available: by measuring a day or two in typical months. At last, a window opening guide for the teachers can be given.

The measurements purely consider the effect of buoyancy pressure, regardless of the wind pressure. Future work can test cross-ventilation as well, since single-sided and cross ventilation is totally different. Cross ventilation is much more wind sensitive, making it hard to measure in real-time situations.

5 CONCLUSIONS

This paper measured ventilation rates with constant injection method to get natural ventilation rates driven by buoyancy pressure in classrooms in a Chinese primary school located in Beijing. Based on the data measured, the relationship between ventilation rate and temperature difference is found. Then, simulation is applied to find the indoor temperature in the classrooms for an entire year. Combined with the climate data of the outdoor temperature, the temperature difference of the whole year can be known. At last, a window opening strategy depending on the indoor and outdoor temperature difference for natural ventilation is provided, in order to save energy.

The results are also available in the early-design-stage of the building design. The predictions of natural ventilation performance in renovated buildings could provide reference to the design of natural ventilation systems.

REFERENCES

Bakó-Biró, Z., D. J. Clements-Croome, N. Kochhar, H. B. Awbi and M. J. Williams (2012). "Ventilation rates in schools and pupils' performance." Building and Environment 48: 215-223.

Bornehag, C. G., J. Sundell, L. Hagerhed-Engman and T. Sigsgaard (2005). "Association between ventilation rates in 390 Swedish homes and allergic symptoms in children." Indoor Air 15(4): 275-280.

Cheng, P. L. and X. F. Li (2014). "Air Change Rate Measurements using Tracer Gas Carbon Dioxide from Dry Ice." International Journal of Ventilation 13(3): 235-245.

Kolarik, B., K. Naydenov, M. Larsson, C.-G. Bornehag and J. Sundell (2008). "The association between phthalates in dust and allergic diseases among Bulgarian children." Environmental health perspectives: 98-103.

McWilliams, J. (2002). "Review of air flow measurement techniques." Lawrence Berkeley National Laboratory.

Ridley, I., S. Pretlove, M. Ucci, D. Mumovic, M. Davies, T. Oreszczyn, M. McCarthy and J. Singh (2006). Asthma/Dust mite study-Final report, Building Research Technical Report 7/2005, Department for Environment, Food and Rural Affairs.

Roulet, C.-A. and L. Vandaele (1991). Airflow patterns within buildings, measurement techniques: air leakage measurement methods, air flow measurement methods, measurement methods related to efficiency, measurements on ventilation systems, Air Infiltration and Ventilation Centre, University of Warwick Science Park.

Sherman, M. (1990). "Tracer-gas techniques for measuring ventilation in a single zone." Building and Environment 25(4): 365-374.

Sundell, J. (2004). "On the history of indoor air quality and health." Indoor air 14(s7): 51-58.

Yan, D., J. Xia, W. Tang, F. Song, X. Zhang and Y. Jiang (2008). DEST - An integrated building simulation toolkit Part I: Fundamentals. Building Simulation, Springer.

Šťávová, P. (2012). "Experimental Evaluation of Ventilation in Dwellings by Tracer Gas CO2."

EVALUATION OF THE REFURBISHMENT POTENTIAL OF MEDITERRANEAN SCHOOLS TOWARDS NZEB

Niki Gaitani[*1], Laia Cases Fabregas[2], Elena Mastapostoli[1], and Mat Santamouris[1]

1 National & Kapodistrian University of Athens, Group of Building Environmental Research, Building of Physics-5, University Campus, 15784, Athens, Greece
**Corresponding author: ngaitani@phys.uoa.gr*

2 Fundació Privada Ascamm, Parc Tecnològic del Vallès, Av. Universitat Autònoma, 23 - 08290 Cerdanyola del Vallès, Barcelona, Spain

ABSTRACT

EU energy policy encourages member states and public authorities to start converting building stock into nearly Zero Energy Buildings (nZEB) and adopting exemplary actions. ZEMedS project focuses on the issues related to the refurbishment of schools to nearly Zero Energy Buildings (nZEB) in France, Greece, Italy and Spain. Presently, there is a gap in national regulation of Mediterranean countries to embody the 2012/27 EED as far as renovation rates of public buildings are concerned. As there is no clear definition of nZEB concept, a roadmap for nZEB, with numerical indicator for energy demand and the share of renewable energy sources is needed. In this context, ZEMedS project aims to cover a complete renovation path, tackling strategies for the envelope, the systems and renewable energy applications as well as the energy management and users' behavior. Ten case studies of typical schools from Catalonia, Tuscany, Athens, Ancona, Montpellier, have been analyzed in terms of the energy efficiency and cost optimality so as to contribute to the ongoing development of a methodology on how to achieve energy efficient and cost optimal nearly zero energy schools while ensuring the IEQ aspects. The case studies are existent schools that do not fit the minimum energy standards and need a renovation. A number of measures dealing with the building envelope and energy systems, were taken into account and examined through energy auditing and simulations tools.

The results presented here ease the understanding of design and construction decisions on the rising cost of energy. In order to set up a common nZEB strategy and to analyze the available funding resources, this study proposes also an integrated Technical and Financial Toolkit. ZEMedS's Toolkit intents to clarify and unify various definition of nZEB, identifies main obstacles and restrictions that face its application and analyses possible social, environmental and economic impacts. Going one step further, the ZEMedS's Toolkit attempts to set-up requirements related to the annual energy balance, the share of RES and the IEQ issues for the renovation of MED Schools.

KEYWORDS

Schools, Renovation, nZEB, Mediterranean

1 CONCEPT OF ZEMEDS PROJECT

Buildings represent the largest available source of cost effective energy saving and CO_2 reduction potential within Europe. The aim to reduce energy consumption in buildings has led to Zero Energy Building (ZEB) concept. Within the European legislative framework [1, 2] nearly

Zero Energy Buildings (nZEB) are arising much interest nowadays and European Union is committed to implement energy efficiency in buildings. This commitment requires efforts from all Member States to contribute to energy efficiency in the building sector, through the adoption of suitable regulatory and policy instruments.

In Mediterranean regions of Italy, Greece, Spain and France, there are approximately 87.000 schools, consuming in a rough estimation around 2Mtoe/year.

ZEMedS (Zero Energy MEDiterranean Schools) [3], is 3-year Project Co-funded by the European Commission within the Intelligent Energy Europe Programme (IEE), which focuses on the issues related to the refurbishment of schools to nZEB. Currently, there is no clear definition of nZEB concept in national regulation of Mediterranean countries to embody the 2012/27 EED as far as renovation rates of public buildings are concerned. A roadmap for nZEB, with numerical indicator for energy demand and the share of renewable energy sources is needed.

The aim of ZEMedS project is to map the energy conservation potentials in Mediterranean schools in relation to the environmental quality perspectives. School buildings feature poor indoor air quality while their energy consumption and overall environmental quality could be improved significantly. Many studies have identified the lack of date base, knowledge, experience and best-practice examples as barriers in refurbishment projects.

The specific objectives of the project are:

- To increase the knowledge and know-how on the nZEB renovation of schools in Mediterranean climates and give support to several new initiatives on the nZEB refurbishment of schools in Mediterranean climate regions
- To promote the necessary actions for the renovation of school buildings in a Mediterranean climate to be nearly zero-energy buildings
- To ensure a reduced energy demand, to be partially covered by renewable energy sources and, at the same time, guarantee a good indoor environment that will impact positively on occupants' health and result in higher learning outcomes for the pupils concerned

2 APPROACH

ZEMedS gives priority to deep renovation; nonetheless well-designed step-by-step procedures can pave the way to nZEB when problems related to funding or schedule are encountered. In this context, this article presents the first results of ZEMedS toolkits and case studies.

Toolkits are addressed to building designers and policy makers and contain technical and financial resources, whereas case studies are real school buildings that have been energy and cost-analyzed so as to define specific renovation strategies. These include energy upgrade of the envelope, enhanced ventilation, re-sizing of heating and lighting equipment, installing renewable energy, controls and user behavior.

The nZEB's approach is ambitious as intends to both comply with zero energy consumption and current standards for indoor environments.

ZEMedS numerical indicators could be summarized along these lines:

Requirement 1. Reduction of energy demand & increase of RES share:
Annual Primary Energy Consumption **PE** (heating, cooling, ventilation, DHW and lighting) covered by RES

$$\mathbf{C_{PE} - Prod_{RES} \leq 0} \quad (1)$$

- C_{PE}: Primary energy consumption yearly for all uses. In accordance with national primary energy factors

- $Prod_{RES}$: Renewable energy supply

Requirement 2. Annual Final Energy Consumption (heating, cooling ventilation & lighting) **FE**:

$$\mathbf{C_{FE} \leq 25\ kWh/m_{reference\ area}^{2}.year}\ (2)$$

- Heating/Cooling and Ventilation: $C_{HVAC} \leq 20$ kWh/m².year
- Lighting: $C_{lighting} \leq 5$ kWh/m².year

Requirement 3. Indoor Air Quality guaranteed:

Concentration of $CO_2 \leq 1000$ ppm
Overheating should be limited to 40 hours annually:
$\mathbf{T_{air\ \ above\ 28\ °C} \leq 40\ hours/year}$ (3)

3 CASE STUDIES

Ten typical schools (Figure 1) from the regions of Catalonia, Tuscany, Athens, Ancona and Montpellier, have been analyzed in terms of the energy efficiency and cost optimality so as to contribute to the ongoing development of a methodology on how to achieve energy efficient and cost optimal nearly zero energy schools while ensuring the IEQ aspects.
A number of measures dealing with the building envelope and energy systems, were taken into account and are listed below:

- Renovation of the façade: External wall insulation system avoiding thermal bridges (with additional application of cool coating products);
- Renovation of the roof: (i) For terrace roofs, external roof insulation system including wind/moisture barriers with new tiles (cool roofs applied in two cases), (ii) For pitched roofs with unheated space under cover, insulation system applied internally;
- Replacement of existing windows with more efficient;
- Installation of external solar protections;
- Replacement of existing lighting with LED technology and installation of daylighting dimming control;
- Installation of ventilation system (i) natural, (ii) mechanical without heat recovery, (iii) mechanical with heat recovery;
- Change of heating system;
- Installation of PV systems

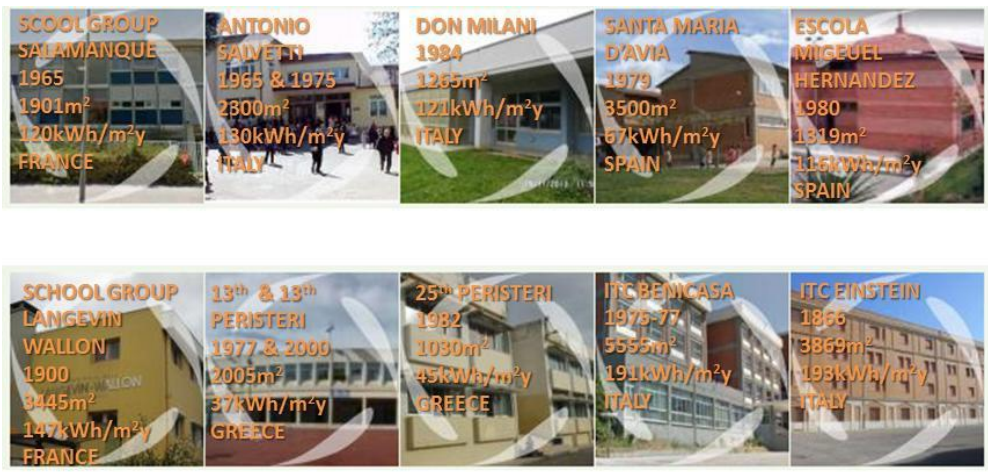


Figure 1: Photos of the ten case studies with values of construction year, heated surface & total consumption according to the bills (average 3 years)

According to the ZEMedS database the total energy consumption from bills varies between
37 kWh/m2y to 193 kWh/m2y (average 118kWh/m2y) and the indoor conditions are not sufficient.

The energy performance of the selected school buildings was calculated with Energy plus simulation program [4]. The measures were analyzed in order to comply with the ZEMedS nZEB requirements and payback calculation has been released for each case. The thickness of the insulation together with the windows quality, were examined step wisely, as fundamental variants.

Table 1 Basic variants related to the examined U values of walls, roofs and windows

U thermal transmittance (W/m^{2}K)	**Variant 1**	**Variant 2**	**Variant 3**
Walls	0.40	0.30	0.20
Roofs	0.30	0.22	0.15
Windows and external doors	1.80	1.50	1.30-1.40

Additionally:
- Thermal bridges considered in building envelope of existing building
- Occupancy: classrooms (0.44 per/m2), offices (0.21 per/m2), dining room (1 per/m2)
- Mechanical ventilation when occupancy at 6,5 l/s person
- Infiltration rate of 30 m3/h m2 at 50 Pa when simulating the existing building and 6 m3/h m2 at 50 Pa when simulating the building with window renovation.
- Natural ventilation for opened windows (5 ACH)

The simulations have been performed step wisely (ITC Benincasa School, Figure 2)

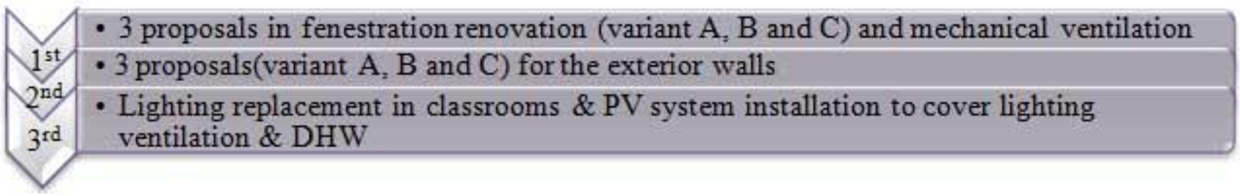


Figure 2: Deep renovation measures in 3 steps approach_ITC Benincasa

The input data of the simulations is presented at the table below.

Table 2 Input data_ Case Study 1_ITC Benincasa

		Variant A		Variant B		Variant C		Current regulation D.Lgs. 311/06
Step 1	Uwindows and doors	1.8	6mm/16mm(AIR)/6mm le (<0.04) aluminium window frame (with thermal break) . Ug=1.64 Uf= 2.2	1.5	6mm/16mm(ARGO)/6mm le (<0.04) aluminium window frame (with thermal break) . Ug=1.34 Uf= 2.2	1.4	6mm/16mm(ARGO)/6mm le (<0.04) wooden window frame. Ug=1.34 Uf= 1.8	Window Uw=2.2; Ug=1.9
	Solar protection	Mobile slats		Mobile slats		Mobile slats		Mobile slats
	Mechanical Ventilation	Ventilation system with heat recovery (control when occupancy) 6.5 l/s person, 70% heat recovery						no mechanical ventilation
Step 2	Uroof	current (U roof 0.42)						0.29 (interior insulation system)
	Uwall	0.4	Ventilated facade with insulation system	0.3	Ventilated facade with insulation system	0.2	Ventilated facade with insulation system	0.36 (exterior insulation system)
	Ugroundfloor	current (U groundfloor 0.9)						current (U groundfloor 0.9)

		Variant A	Variant B	Variant C	Current regulation D.Lgs. 311/06
Step3	Lighting	replacing T8 tubes for LED tubes in classrooms 6.3 W/m2. Daylight Regulation (dimming).			22 kWp / 6 kWh/m2 (50% of expeted energy consumption for DHW, heating, cooling)
	Heating system & DHW	current			
	Cooling system	current			
	PV system	34 kWp / 10 kWh/m2			

4 MAIN RESULTS

The results presented here ease the understanding of design and construction decisions on the rising cost of energy.

Here presented for four case studies the energy consumption from bills and simulations:

Table 3 Results for 4 School Buildings

Name school	Location	Conditioned area (m2)	Energy consumption in gas (kWh/m2) of current situation (bills)	Energy consumption in electricity (kWh/m2) of current situation (bills)	Energy consumption in gas (kWh/m2) of current situation (simulation)	Energy consumption in electricity (kWh/m2) of current situation (simulation)
Group school Bedarieux	Bédarieux, France	3445	155	24	107	28
Group school Salamanque	Montpellier, France	2303	81	18	86	21
ITC Benincasa	Ancona, Marche, Italy	4942	138	21	96	27
School Miguel Hernandez	Badalona, Catalonia, Spain	1147	92	42	81	35

A deep renovation strategy for each case is summarized at Table 4.

Table 4 Examined deep renovation strategies

Name school	Deep renovation strategies with energy measures in
Group school Bedarieux (result 1)	envelope + heating system (biomass boiler) + PV system covering (lighting, DHW by electricity)
Group school Bedarieux (result 2)	envelope + lighting and dimming control + heating system (biomass boiler) + PV system covering (lighting, DHW by electricity)
Group school Salamanque	envelope + lighting + mechanical ventilation with heat recovery + PV system covering (heating by natural gas, lighting, ventilation, DHW by electricity)
ITC Benincasa	envelope + mechanical ventilation with heat recovery + lighting and dimming control + PV system covering (heating by natural gas, ventilation, lighting, DHW by natural gas)
School Miguel Hernandez (result 1)	envelope + mechanical ventilation with heat recovery + lighting + PV system covering (heating by natural gas, ventilation, lighting, DHW by electricity (primary school) and natural gas (pre-school))
School Miguel Hernandez (result 2)	envelope + mechanical ventilation with heat recovery + lighting + heating & DHW system (gas condensing boiler) + PV system covering (heating by natural gas, ventilation, lighting, DHW by electricity (primary school) and natural gas (pre-school))
School Miguel Hernandez (result 3)	envelope + mechanical ventilation with heat recovery + lighting + heating & DHW system (geothermic) + PV system covering (heating by electricity, ventilation, lighting, DHW by electricity)

The results corresponding to the ZEMedS requirements are presented at Table 5.

Table 5 Results

	Energy balance in PE (kWh/m2 y) (heating, cooling, vent., DHW & lighting) (ZEMedS requirement 1) (data from simulations)			Energy result in FE (kWh/m2 y) (heating, cooling, vent. & lighting) per conditioned area (ZEMedS requirement 2) (data from simulations)			Payback (years)
Name school	Var 1	Var 2	Var 3	Var 1	Var 2	Var 3	-
Group school Bedarieux (result 1)	2	2	2	38	36	33	15-16
Group school Bedarieux (result 2)	1	1	1	31	28	26	18-19
Group school Salamanque	1	0	-1	40	39	38	26-27
ITC Benincasa	3	-1	-3	23	19	17	18-19
School Miguel Hernandez (result 1)	4	2	0	23	21	19	19-20
School Miguel Hernandez (result 2)	1	-1	-2	19	17	16	20-21
School Miguel Hernandez (result 3)	-1	-1	-1	10	9	9	22-23

The results to accomplish requirement 2 [$C_{FE} \leq 25$ kWh/m $_{reference\ area}{}^2$.year] depend strongly on the current situation of the case studies. According to the results obtained, the requirement
2 can be accomplished for the schools with relatively lower energy consumption of gas in the current situation (as School Miguel Hernandez). Furthermore, energy strategies such as the reduction of the internal loads with the improvement of lighting with daylight dimming control (ITC Benincasa) can contribute to less than 25kWh/m^2 in FE.
As regards to the requirement 1, the examined PV systems have been dimensioned to be able to cover the energy consumption established with a surface of panels more than 250 m^2. This leads to payback values no lower than 15 years.
In all cases, the indoor air quality is improved through mechanical ventilation and good design of solar protections, although the use of the building is fundamental for the well-being of the users. Payback calculations were implemented for the packages of examined measures. The main difficulty was a lack of information for the cost in maintenance and replacements regarding the existing buildings.

Table 6 Payback analysis for Case Study 1_ITC Benincasa

	Expected savings in gas	**Expected savings in electricity**	**Overall cost of gas €/year**	**Overall cost of electricity €/year**	**Investment in €**	**Overall maintenance cost €/year**	**Cost of replacement in €**	**Items to be replaced**	**Payback (year)**
variant 1 (fenestration & solar protection) & MVHR	72%	0%	14514	26990	617324	1325	0	-	13
variant 2 (fenestration &solar protection) & MVHR	73%	0%	13996	26990	624852			-	13
variant 3 (fenestration &solar protection) & MVHR	73%	0%	13996	26990	745642			-	15
variant 1 (fenestration & solar protection & ext. wall) & MVHR	86%	0%	7257	26990	1124335	1795	0	-	18
variant 2 (fenestration & solar protection & ext. wall) & MVHR	89%	0%	5702	26990	1157737			-	18
variant 3 (fenestration & solar protection & ext. wall) & MVHR	91%	0%	4665	26990	1307899			-	19
variant 1 (fenestration & solar protection & ext. wall) & MVHR & lighting dimm & PV	84%	34%	8294	17813	1266292	2677	76781	lighting T5 tubes (15-20 years lifetime) /inverters PV (15 years lifetime)	18
variant 2 (fenestration & solar protection & ext. wall) & MVHR & lighting dimm & PV	88%	34%	6220	17813	1299694				18
variant 3 (fenestration & solar protection & ext. wall) & MVHR & lighting dimm & PV	90%	34%	5184	17813	1449856				19

Although improving the building efficiency is often profitable, investments are hindered by barriers.

5 CONCLUDING REMARKS

The ZEMeds project intents to elucidate the relationship between nearly zero-energy and cost- optimal measures and to develop an argumentation on how to ensure a smooth transition from current MED schools to nearly zero energy school buildings. In this context, ten case studies

of school buildings have been analyzed in terms of the energy efficiency and cost optimality so as to define a detailed renovation action plan.
The ZEMeds integrated Technical and Financial Toolkit offers multiple best practices, techniques and methods that guide the implementation of nZEB actions in accordance to the unique necessities of each school, region and country in the Mediterranean area. Also the Toolkit attempts to set- up energy performance and IEQ requirements of MED schools with a view to achieving cost-optimal levels. These requirements have been validated through the examined case studies.

- Typical Mediterranean school built in the period 60-80's may consume around
  118kWh/m^2/y (final energy), count with many overheating hours, has glare problems and inefficient ventilation;
- Once efficiency measures have been incorporated, the remaining energy needs can be met using renewable energy technology;
- With the suggested measures, classrooms used during summertime reduce overheating hours to less than 40h;
- The payback period varied 15-25 years

The hierarchy of priorities for building renovation of the examined schools is based on different needs (safety, maintenance, spatial requirements, energy savings, etc.) and profoundly depends on the budget availability and the existing funding channels.

6 REFERENCES

[1] European Parliament and Council (2010) DIRECTIVE 2010/31/EU of 19 May 2010 on the energy performance of buildings (recast). Official Journal of the European Union, 18.6.20
10
[2] Directive 2012/27/EU of the European Parliament and of the Council of 25 October 2012 on energy efficiency, amending Directives 2009/125/EC and 2010/30/EU and repealing Directives 2004/8/EC and 2006/32/EC
[3]. ZEMedS:
http://www.zemeds.eu/
[4]. Energy Plus:
http://apps1.eere.energy.gov/buildings/energyplus/

HOW TO CONSTRUCT A DOMESTIC PITCHED ROOF WITH HIGH THERMAL QUALITY?

Staf Roels* and Jelle Langmans

Department of Civil Engineering
Building Physics Section, University of Leuven
Kasteelpark Arenberg 40 – bus 02447
BE-3001, Heverlee, Belgium
**Corresponding author: staf.roels@bwk.kuleuven.be*

ABSTRACT

The paper at hand collects research findings on the impact of air flow on the thermal performance of pitched roof assemblies. Air flows in these components are typically a mixture of: 1) in/exfiltration, 2) natural convection and 3) wind-washing. In the current building practice the necessity of an air barrier to guarantee the thermal and hygric performance of roofs is well established. Yet the need for a continuous wind barrier to avoid wind-washing of the insulation layer is still often underestimated in practice. In addition the literature review shows that already small leakage paths around the insulation layer may induces an important reduction of the thermal performance due to buoyant driven air loops which is often overlooked in today's building methods. Based on the findings in the literature the present article puts forward guidelines on how to construct pitched roofs with a robust high thermal performance.

KEYWORDS

Wind-washing, natural convection, pitched roofs, guidelines, literature review

1 INTRODUCTION AND PROBLEM STATEMENT

A sound thermal insulation is one of the key factors to reduce the energy use of new and existing buildings. Consequently most European member states increased the U-value requirements for opaque building components to values of 0.25 W/m²K or even better. These requirements are typically evaluated in the design phase: the theoretical performance of the building component is calculated based on the thickness and thermal conductivity of the composing layers. It is however well-known that the actual achieved thermal performance is not only a matter of the thickness of the insulation layer. Other phenomena than pure heat conduction can strongly spoil the performance on site (a.o. Hens et al. 2007, Janssens & Hens 2007, Lowe et al. 2007).

For pitched roofs air movement through the building component is considered as the malefactor, often resulting in an on-site thermal quality much lower than the design value. Recently the British Board of Agrément, one of UK's leading certification bodies, published a report on the air movement and thermal performance of pitched roof constructions (BBA, 2012). The results and conclusions were based on experimental research in laboratory

conditions making use of a modified Hot Box. By comparing the thermal performance as a function of imposed air speed at the cold side of the roofs of three so-called ‘standard build practice’ roof configurations, a significant performance gap was observed for all roof types. On average across all three configurations, a 90% increase was reported between calculated and measured U-value at higher wind speeds. Most striking in this study are not the obtained results as such – similar results have been reported in previous studies –, but the fact that the pitched roof constructions tested are considered as standard building practice. The huge impact of air movement on the actual thermal performance of building enclosures has been studied extensively for decades. Already in 1989 Powell et al. published a literature survey on the influence of air movement on the effective thermal resistance of porous insulations. Ever since, several studies investigated the effect of different air movement patterns on the thermal performance and durability of (mainly lightweight) building components (e.g. Brown et al. 1993, Di Lenardo 1995, Uvlsok 1996 and Janssens & Hens 2007). Several of those studies, resulted in guidelines and performance requirements of the composing material layers (a.o. NBC 1995, Straube and Burnett 2011, Uvsløkk et al. 2010). However, it seems that these (fragmented) recommendations did not led to a roof construction resilient to air flow effects in day to day building practice.
Therefore, the current article reviews previous research work on air movement in light weight building components in general and pitched roofs in particular. In this study we focus on lightweight insulated sloped roofs, consisting of a wooden framing structure, insulated by filling the structural cavity in between the framing. Based on the literature, the first part of this paper reiterates the common air flow patterns and their effect on the heat transmission losses. In the second part, the observed findings are compiled in specific performance requirements and guidelines for good building practice for a pitch roof construction. When following these guidelines, the performance gap can be minimized and the actual thermal performance of the roofs will be in line with the design values.

2 COMMON AIR FLOW PATTERNS AND THEIR EFFECT ON THERMAL PERFORMANCE

The impact of air movement on the thermal performance of lightweight building components has been extensively studied in the literature. For pitched roofs, three different kind of air flow patterns have to be accounted for:

- air leakage (diffuse and concentrated)
- natural convection
- wind washing

The first corresponds to forced convection due to deficiencies in – or even the lack of – an airtight layer. As a result pressure differentials across the building envelope (be it induced by an indoor-outdoor temperature difference, by wind or by mechanical ventilation) will force the air to flow through the building component. For pitched roofs forced exfiltration conditions are most dominant, resulting not only in increased heat losses, but often also in moisture problems (Janssens & Hens, 2007, Langmans et al. 2013). The second pattern causes a reduced thermal performance of the building component due to free heat convection as a result of buoyancy driven air rotation in and around the insulation layer. Driving force is the temperature difference across the building component. The last air flow pattern, wind washing, results from the fact that commonly no stringent airtightness requirements are put forward for the exterior protective layer (the underlay). As a result, even with a good overall airtightness achieved, exterior air may flow through air permeable insulation materials, again jeopardizing the designed thermal quality.

Figure 1 shows the different kind of air movements observed in sloped roofs: air leakage (diffuse and concentrated), natural convection in and around the insulation layer, and wind washing. Though these mechanisms typically occur as mixed processes, the next sections will try to quantify the impact of each of the different air flow patterns. Based on a literature review, the present article will conclude on the necessary requirements to avoid these air movements.

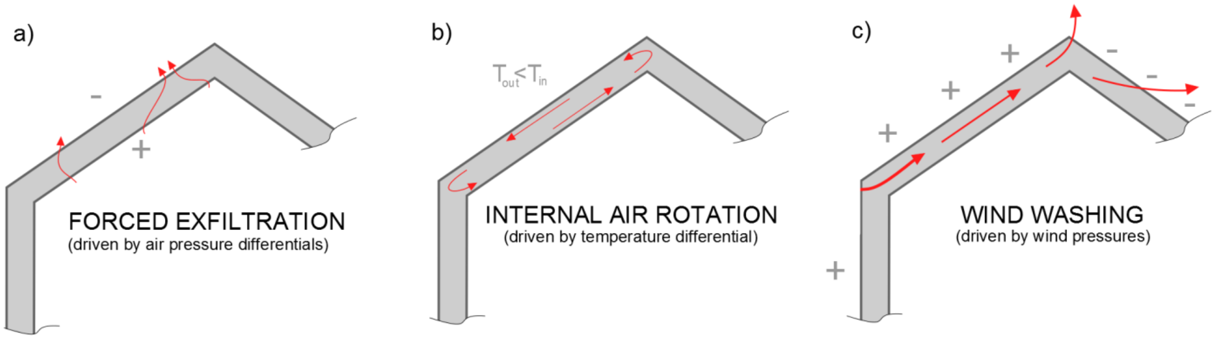


Figure 1: Possible air flow phenomena reducing the thermal performance of pitched roofs: a) forced exfiltration through the pitched roof, b) internal air rotation in and around the insulation layer and c) wind-washing of the insulating layer.

2.1 Forced exfiltration

Most straightforward form of air transport within sloped roofs is forced convection. This occurs when an air pressure differential is established across a building component including air leakages. The overall air pressure difference across the roof is induced by wind, temperature differences and/or mechanical ventilation. The distribution and order of magnitude of the leakage paths within the pitched roof determines the resulting air flow pattern. Desmarais et al. (2000) conducted a detailed laboratory investigation in which they distinguished between: a) long paths, b) concentrated paths and c) distributed (diffuse) leakage paths. The shape of the air flow pattern highly influences the impact on the thermal performance of the building component. A concentrated air flow path will for example correspond to higher heat losses than a distributed air flow path with the same overall air leakage level.

In- and exfiltration in light weight building components can be readily avoided by installing a continuous airtight layer, commonly called the air barrier. Because in practise air barriers consist of different elements and joints, Straube and Burnett (2011) propose to use the term air barrier system. Already in 1985, Di Lenardo et al. prescribed target values for the air permeability of air barrier systems, including anticipated joints and perforations. At that moment, an upper limit of 2.7 10^{-6} $m^3/(m^2sPa)$ was put forward for an air barrier system including joints. This value was based on numerical simulations for the Canadian climate, considering both limit state values for moisture accumulation as well as an increase of the conductive heat transfer (up to 15%) due to air leakages. This value has been tightened by Straube and Burnett (2011), proposing an upper limit of 1.3 10^{-6} $m^3/(m^2sPa)$. Apart from the requirements for the overall airtightness of the air barrier system, they stress the importance of other criteria, such as continuity, strength, durability and stiffness.

2.2 Internal air rotation

Most of the existing studies on natural convection in light weight building components are related to stud walls. Apart for an inclination angle, the composition of pitched roofs is very similar to stud walls. As a consequence the research results regarding internal air rotations in stud walls can be easily transformed to pitched roofs taking into account the inclination angle.
The first studies investigating the effects of air movement on the thermal performance of permeable insulation systems occurred in the late fifties in Norway (Lorentzen & Brendeng 1959). These experimental studies demonstrated that low density insulations may lead to convection loops in and around the insulation layer, significantly reducing its thermal performance. Hereafter many authors have examined the same effect for various configurations and temperature differences across the components (e.g. Bankvall 1972, Silberstein et al. 1990, Klarsfeld & Combarnous 1980, Lecompte 1989, Dyrbøl et al. 2002). A first comprehensive literature review on this topic has been written by Powell et al. (1989). Their review mainly distinguishes between two configurations, studied at that time: (1) a cavity filled with an open porous insulation material and (2) an air cavity partly filled with a thermal insulation layer.
In summary, the studies under review, generally conclude that the impact of natural convection on the thermal performance is limited when at least one of the adjacent layers is airtight, air gaps along the interface between the sheathing and insulation are avoided and sufficiently dense insulation materials (>20-30 kg/m3) are applied.

2.3 Wind washing

The previous section restricted the discussion to internal air rotations effects, presuming an ideal situation in that the insulation layer is closed between two airtight layers. Nevertheless, in reality the exterior protective layer (wind barrier) is most often not sufficiently airtight. As a consequence exterior air may flow through permeable insulation materials driven by pressure gradients on the building envelope. First laboratory investigations on the thermal impact of wind-washing emerged in Scandinavia by Timusk et al. (1991) and Uvsløkk (1996). Timusk et al. (1991) performed experiments on full-scale corners of wood frame walls exposed to wind conditions. They found that defects in the wall sheathing resulted in a significant decrease of its thermal performance. Later, Uvsløkk (1996) conducted full-scale laboratory measurements on wind-washing effects in timber frame walls. The applied exterior pressure difference, based on in situ measurements, was realized by imposing a pressure difference at the walls cladding in and outlet. Based on these measurements, Uvsløkk (1996) proposed a maximum overall air permeability of the wind barrier system of 14 10^{-6} $m^3/(m^2$ sPa). Ojanen & Kohonen (1995), at their turn, found similar threshold values by numerical simulation. These additional simulation results illustrated that compartmentation of the insulation layer reduced wind-washing. As a consequence, Ojanen & Kohonen (1995) suggested a maximum air permeance of 10 10^{-6} $m^3/(m^2$ sPa) when strong corner convection is possible and 25 10^{-6} $m^3/(m^2$sPa) when the building envelope is divided in separate structures.
In addition to these laboratory measurements, Janssens & Hens (2007) studied windwashing effects on in situ pitched roofs in Belgium. They examined the overall air flow patterns with tracer gas tests and temperature and heat flux sensors installed at three heights. Results revealed a clear correlation between the exterior wind speeds and the reduction of the thermal performance. Most significant impact on the thermal performance was located at the level of the ridge. The measured reduction of the overall thermal resistance of roofs for a wind of 4 m/s coming from a direction perpendicular to the roof was 10% for entirely insulated elements (compact roofs) and 40% for partly insulated roof elements (vented roofs). Based on

these in situ measurements Janssens & Hens (2007) recommended to seal underlay materials in duo-pitched roofs to improve their air permeance levels to the proposed values of Uvsløkk (1996) and Ojanen & Kohonen (1995). These postulated recommendations notwithstanding, the vast majority of the current building practice still not seals the wind barrier to prevent wind-washing.

3 PRACTICAL GUIDELINE FOR PITCHED ROOFS

The literature review in the previous section revealed that the thermal performance of pitch roofs may be significantly affected by air flow patterns. At the same time, based on their findings, several researchers put forward some guidelines and performance targets to make the roofs less vulnerable to air flow effects. This section compiles for each of the air flow patterns all outcomes to come up with general guidelines on how to construct a pitch roof with a robust thermal performance.

First of all, to **avoid air leakage**, the construction should contain a continuous airtight layer, the so-called air barrier system. This air barrier system has to preclude in- and exfiltration through the roof component. As upper limit a maximum air permeability of 1.3 10^{-6} $m^3/(m^2sPa)$ can be maintained. Note however, that in addition to the airtightness of the material layer (which can easily be measured and achieved (e.g. Langmans et al. 2010), the continuity of the air barrier system is of far more importance, guaranteeing an overall airtightness of the system including joints, overlaps,… Therefore, a typical interior finishing as for instance coated gypsum board (that easily fulfils the air permeability requirements) is often not considered sufficient since achieving continuity is hard at service penetrations, wall-roof interfaces, etc.. Furthermore, cracks in the inner lining might appear due to wind gust loads on the roof. That is why, common guidelines propose to separate the interior lining and air barrier system, and to install them at different positions to avoid damage of the air barrier system when perforating the lining. In European countries often an air-vapour barrier system is used at the warm side of the insulation to avoid both air leakage through the component, as well as vapour diffusion and convection. Alternatively, a warm roof construction (similar to a flat roof design) could be applied, in which the location of the air (vapour) barrier system, insulation and outside finishing is placed on top of the structural sheathing and framing.

Once forced exfiltration is excluded by a sound air barrier system, it is important to **avoid air rotation by natural convection** within the roof component. Section 2.2 has shown that the thermal resistance of insulation layers can be significantly degraded by natural convection. Convection loops were found to be highly triggered by air cavities or even small air gaps between insulation layer and boundary surface (e.g. Brown (1993) and Janssen (1997)). Therefore, a tight contact between insulation and both interior and exterior surface will be important to reduce the risk on air rotation. This asks for a compact roof. But even when air gaps along underlay and interior air barrier system are avoided, internal air loops through low-density fibrous insulation layers may occur (Powell et al., 1989). To reduce the impact of natural convection, Langmans (2013) proposes the application of denser insulation materials (> 20 kg/m^3). Also blown in insulation was found to be more resilient to air looping than insulation blankets. Not only is the risk on small air gaps lower when insulation is blown into the compartment, often also higher insulation densities are applied.

Last, but not least, and unfortunately often overlooked, wind-washing should be avoided. Even with an interior air barrier system fulfilling passive house standards[1], cold outside air might penetrate in the insulation layer due to wind induced forced convection and leave the construction again at another position, but now as warm air. Pitched roofs are rather

[1]Passive house standard explicitly require an overall building airtightness of n50=0.6 1/h

susceptible to windwashing as wind will induce steep pressure gradients along the roof sides. To avoid this, an additional air barrier is foreseen at the outside, the so-called wind barrier. Following Uvslokk (1996) and Ojanen & Kohonen (1995) an upper limit of 10-15*10^{-6} $m^3/(m^2sPa)$ is put forward for the air permeability of the wind barrier. Most roof underlay materials fulfil these requirements and can take up the role of wind barrier. In addition to the air permeance requirements of the underlay, the field study of Janssens & Hens (2007) stressed the importance of a continuous wind barrier. This is important, as for its original function of drainage plane, no sealed joints were requested, but an overlap sufficed. The airtight continuity of the underlay often results in adapted building details. Whereas a continuous wind barrier is feasible in the roof surface itself, it is harder to achieve at eave and ridge joints.. In the eave detail a pre-installed airtight strip is used to make a continuous airtight connection between wind barrier and interior air barrier system, securing the insulation layer for wind-washing.

4 DISCUSSION & CONCLUSIONS

Contrary to the recent conclusions of the (BBA, 2012) in which a significant performance gap was observed for all kind of warm roof constructions, the previous section illustrated that it is possible to construct a well-performing and robust pitched roof with high thermal quality and resilient to air flow patterns. To do so, the following advices should be taken into account:

- To avoid wind washing and air looping, create a compact roof, i.e. fill the cavity between the underlay and the air and vapour barrier over the total height of the rafters and use an insulation material with a high enough density (> 20 kg/m^3) and some compressibility.
- Apply a continuous air (and vapour) barrier system at the inside of the insulation layer, in order to assure the airtightness of the roof construction when the internal lining is perforated. The overall air permeability of the air barrier system should be lower than 1.3 10^{-6} $m^3/(m^2sPa)$. Furthermore, make sure that the internal lining and air barrier are at different positions, so that the lining can be perforated without damaging the air barrier.
- Apply a sufficiently airtight underlay (air permeance < 10-15 10^{-6} $m^3/(m^2sPa)$) with sealed joints (or airtight tongue and groove system). Pay attention to an airtight detailing of the eaves and the ridge.
- To increase the thermal quality even more, the cavity between air and vapour barrier and interior lining can be insulated. Air flow has hardly any impact on the thermal performance of this cavity and the extra insulation layer gives additional support to the air barrier system, increasing its lifetime and durability.

5 REFERENCES

Bankvall, C. (1972), *Natural convective heat transfer in insulated structures*, PhD thesis, Division of building technology, Lund institute of technology, Sweden, Lund, Sweden.

BBA (2012) Air movement and thermal performance – An investigation into the effect of air movement on the thermal performance of domestic pitched roof constructions. British Board of Agrément, Technical Report nr. 1

Brown, W. C., Bomberg, M. T., Ullett, J. M. & Rasmussen, J. (1993), *Measured thermal resistance of frame walls with defects in the installation of mineral fibre insulation*, Journal of Building Physics 16(4), 318–339.

Desmarais, G., Derome, D., & Fazio, P. (2000). *Mapping of air leakage in exterior wall assemblies*. Journal of Building Physics, 24, 132–154.

Dyrbøl, S., Svendsen, S. & Elmroth, A. (2002), *Experimental Investigation of the Effect of Natural Convection on Heat Transfer in Mineral Wool*, Journal of Building Physics 26(2), 153–164.

Hens, H., Janssens, A., Depraetere, W., Carmeliet, J., Lecompte, J. (2007). *Brick Cavity Walls: a Performance Analysis Based on Measurements and Simulations,* Journal of Building Physics, 31 (2), 95-124.

Hens (2003). *Interstitial condensation due to air Leakage: a sensitivity analysis*. Journal of Thermal Envelope and Building Science, 27(1), 15–29.

ISO 10456:2007. Building materials and products -- Hygrothermal properties -- Tabulated design values and procedures for determining declared and design thermal values

Janssen, H. (1997), *Thermal performance of highly insulated wood frame walls*, Master thesis, Norwegian University of Science and Technology, Norway.

Janssens, A., & Hens, H. (2003). *Interstitial condensation due to air Leakage: a sensitivity analysis*. Journal of Thermal Envelope and Building Science, 27(1), 15–29.

Janssens, A. & Hens, H. (2007), *Effects of wind on the transmission heat loss in duopitched insulated roofs: A field study*, Energy and Buildings 39(9), 1047–1054.Janssens, A., & Hens,

Kalamees, T., & Vinha, J. (2003). *Hygrothermal calculations and laboratory tests on timber-framed wall structures*. Building and Environment, 38(5), 689–697.

Klarsfeld, S. & Combarnous, M. (1980), *Analyse des transferts thermiques convectifs dans les isolants poreux perméables*, Revue generale de thermique 228, 977–983.

Langmans, J., Klein, R., Roels, S. (2012). *Hygrothermal risks of using exterior air barrier systems for highly insulated light weight walls: A laboratory investigation*. Building and Environment, 56 (10), 192-202.

Langmans, J., Klein, R., & Roels, S. (2012). *Hygrothermal risks of using exterior air barrier systems for highly insulated light weight walls: a laboratory investigation*. Building and Environment, 56(10), 192–202.

Langmans, J., Klein, R., & Roels, S. (2010). *Air Permeability Requirements for Air Barrier Materials in Passive Houses*. In International Symposium on Building and Ductwork Air-tightness. Copenhagen, Denmark: Energie + Umwelt Zentrum.

Lecompte, J. (1989), *The influence of natural convection on the thermal quality of insulated cavity walls constructions (in Dutch)*, PhD thesis, Departement of Civil Engineering, KU Leuven, Belgium.

Lorentzen, G. & Brendeng, E. (1959), *On the influence of free convection in insulated vertical walls*, in '10th International congress of refrigeriation', p. 294.

National Building Code of Canada 1995, Institute for Research in Construction, National Research Council Canada, Appendix A-9.25.1.2.

Powell, F., Krarti, M. & Tuluca, A. (1989), *Air movement influence on the effective thermal resistance of porous insulations: a literature survey*, Journal of Building Physics 12(3), 239–251.

Riesner, K. (2003), *Natural convection in exterior walls (in German)*, Phd thesis, Univeristy of Rockstock, Germany.

Rousseau J. (1998). *Air permeance of Building Materials, Research Highlights*. Technical Series: 98-109. (Available on http://www.cmhc.ca/publications/en/rh-pr/tech/98109.htm)

Timusk, J., Seskus, A. & Ary, N. (1991), *The control of wind cooling of wood frame building enclosures*, Journal of buildings Physics 15(1).

Uvsløkk, S. (1996), *The Importance of Wind Barriers for Insulated Timber Frame Constructions*, Journal of Building Physics 20(1), 40–62.

Straube, J. 2011. *High Performance Building Enclosures*. Somerville, MA: Building Science Press

Silberstein, A., Langlais, C. & Arquis, E. (1990), *Natural convection in light fibrous insulating materials with permeable interfaces: onset criteria and its effect on the thermal performances of the product*, Journal of Building Physics 14(1), 22–42.

Silberstein, A., & Hens, H. (1996). *Effects of air and moisture flows on the thermal performance of insulations in ventilated roofs and walls*. Journal of Building Physics, 19(4), 367–385

Wahlgren, P. (2007). *Overview and literature survey of natural and forced convection in attic insulation*. Journal of Building Physics, 30(4), 351–370.

DETAILED NUMERICAL MODELLING OF MOIST AIR FLOW THROUGH A COMPLEX AIRTIGHTNESS DEFECT

Clément Belleudy[*,1,2], Monika Woloszyn[1], Matthieu Cosnier[2]

1 LOCIE, CNRS 5271, Université Savoie Mont Blanc
Campus Scientifique Savoie Technolac
Bâtiment Helios, Avenue du Lac Léman,
73376 Le Bourget du Lac, France

2 Centre Scientifique et Technique du Bâtiment
24 rue Joseph Fourier
38400 Saint Martin d'Hères, France
**Corresponding author: clement.belleudy@cstb.fr*

ABSTRACT

Mastering building airtightness is essential to meet the requirements of current and future building codes, not only for saving energy but also for ensuring moisture safety. Perfect airtightness is difficult to achieve: failures are often observed, due to bad design or poor workmanship. Some published investigations proved that leaking air mostly flows through porous material and thin air channels, due to material imperfections and construction tolerances. In addition, air inlet and outlet are not necessarily close to each other, which makes air leakage paths through the building envelope multidimensional and difficult to map. Very few existing models enable such complex air leakage geometries to be dealt with.
In this article, a recently developed detailed model coupling heat air and moisture (HAM) transfer is presented and used to analyse moist air flows due to airtightness defects. The model is able to deal with anisothermal airflow through complex 2D building assemblies, including both air permeable porous materials and thin air channels. The model is then applied on a 2D air leakage configuration subjected to infiltration and exfiltration scenarii. Results are analysed in terms of moisture risk and energy impact. A parametric study on boundary conditions is carried out. The results show higher moisture risk in case of air exfiltration through the airtightness defect.

KEYWORDS

HAM, air leakage, moisture, airtightness, modelling

1 INTRODUCTION

Building airtightness is a major issue to meet the targeted requirements of low energy buildings. Bad design and poor workmanship can lead to unintended air leakage across building envelopes, which leads not only to an increase in energy consumption, but also to additional potential moisture damages inside building assemblies. It is of importance to better assess the impact of air leakage on the hygrothermal performance of the envelope. There is a need to provide quantitative results of this impact in terms of energy and moisture, which may contribute to raise building actors' awareness about a better integration of airtightness into the building construction process.

Modelling coupled heat, air and moisture transfer is a demanding task, especially as leaking air flows through the building envelope by combined air channels and air permeable porous insulation materials. Moreover, these air paths are multidimensional and difficult to map. Measurement campaigns conducted in France on a sample of buildings enabled to locate most frequently encountered air leakage (CETE de Lyon 2010). From these campaigns, booklets presenting 2D sections of the sensitive points towards air leakage have been released for main typologies of buildings. To the best of our knowledge, there are in literature very few numerical models able to deal with anisothermal HAM transfer through air permeable porous material and thin air channels. For example, a simplified approach to model HAM transfer in thin air channels in contact with airtight materials has been introduced by (Nespoli, Janetti, and Ochs 2013). In this "line source approach", air channels are reduced to 1D domains for HAM transfer and the coupling with HM transfer through airtight porous material is ensured by using heat and moisture surface film coefficients along the air channel. This approach has been successfully compared to experimental data (Janetti 2014). To deal with complex assemblies including air channels and air permeable porous insulating materials, (Janssens 1998) and (Langmans 2013) solve the velocity field in both domains: with Darcy law in the porous medium, and averaged Poiseuille law in the air channel. Similarly to Janetti, HAM transfer between porous and air domains are solved with a two-domain approach, meaning that both porous and air domains are coupled with surface coefficients.

In the present article, we use a newly-developed model called HAM-Lea ("Lea" standing for "Leakage"). It differs from the previous model by considering a one-domain approach - also called conjugate approach - to simulate HAM transfer in porous and air domains. Practically, the same structure of equations is used in both domains, and no surface coefficient is therefore required for the coupling. This model has been built from a HA-model (Belleudy et al. 2014) in which moisture has been subsequently implemented. This paper firstly presents HAM-Lea's governing equations, before applying it to a 2D airtightness defect in infiltration or exfiltration with transient temperature and moisture boundary conditions. In a third part, the impact of airflow on the building assembly is analysed in terms of moisture and energy, and the consequence on the results of a higher external maximal temperature is investigated.

2 NUMERICAL MODEL

2.1 Governing equations

HAM-Lea is formulated using the continuous medium approximation: material properties and local fields are averaged over Representative Elementary Volumes (REV), which enable conservation laws to be written in their local form using Partial Derivative Equations (PDE). HAM-Lea's conservation equations for energy (eq. 1), moisture (eq. 2), mass (eq. 3), and momentum (eq. 4) are given below, with $(T; \varphi; u; P)$ as variables:

$$\frac{\partial H(T;\varphi)}{\partial t} = -\nabla\cdot\left[q_{cond}(T;\varphi) + q_{conv}(T) + q_{lat}(T;\varphi)\right] \tag{1}$$

$$\frac{\partial w(\varphi)}{\partial t} = -\nabla\cdot\left[g_{diff}(T;\varphi) + g_{adv}(T;\varphi) + g_{liq}(\varphi)\right] \tag{2}$$

$$\nabla\cdot u = 0 \tag{3}$$

$$u = -\frac{k_{mat}}{\mu_{air}}\nabla P \tag{4}$$

The simple form of continuity equation (eq. 3) is due to low air velocities encountered in building physics, implying the assumption of incompressible flow. Darcy law (eq. 4) is a simplified form of Navier-Stokes equation in the porous material to describe momentum conservation. Averaged Poseuille law written in the air channel allows to identify an equivalent permeability (Belleudy 2015). Darcy law is valid for low velocities, i.e. a pore Reynolds number of order of unity. Natural convection is said to be of limited importance when indoor-outdoor temperature differences do not exceed 40°C (Langlais, Arquis, and McCaa 1990). As HAM-Lea is firstly dedicated for temperate climates, natural convection is not implemented. This choice also enhances simulation performance because Darcy law can be solved independently, prior to moisture and energy equations. For the energy equation, a one-temperature approach is adopted, assuming therefore thermal equilibrium between air and solid material. This approximation is valid for building insulation materials with low air velocities, as proved by (Buchanan and Sherman 2000). In the energy equation (eq. 1), the enthalpy variation rate of a REV containing porous material is expressed as:

$$\frac{\partial H(T;\varphi)}{\partial t} = [\rho_{mat} c_{mat} + w(\varphi) c_w] \frac{\partial T}{\partial t} \tag{5}$$

A similar term can be written for REV in air channels (Belleudy 2015). The energy variation is driven by three net heat fluxes (q_{lat}, explained at the end of this section, q_{cond} and q_{conv}) as shown in (eq. 1). The heat conduction flux density q_{cond} is described by the well-known Fourier law (eq. 6) with a moisture dependent thermal conductivity $\lambda_{mat}(\varphi)$ in the porous material and a constant one λ_{air} in the air channel:

$$q_{cond}(T;\varphi) = -\lambda_{mat}(\varphi) \nabla T \tag{6}$$

Air is regarded as an ideal mixture between dry air and water vapour following the ideal gas law. The expression of the convective dry air flux density appearing in (eq. 1) is:

$$q_{conv}(T) = \rho_{air}\, c_{p_{air}}\, u\, T \tag{7}$$

Moisture conservation states that the increase rate of moisture content *w* in an REV is equal to the sum of three net moisture fluxes which are vapour diffusion, vapour advection and liquid transport. Vapour diffusion is described by Fick law (eq. 8). Vapour advection refers to the vapour flow carried by airflow (eq. 9). The humidity by volume of air, ρ_{vap}, can be linked with temperature and vapour pressure via the ideal gas law (eq. 11). Liquid transport is driven by capillary suction in water filled pores. This phenomena occurs in smaller pores first, and becomes dominant for high relative humidities ($\varphi > 0.98$). It can be written with φ as potential (eq. 10). g_{liq} is zero in air channels. The definition of relative humidity is needed (eq. 12) in order to add the relationship between φ and vapour pressure p_v, introducing the saturation pressure P_{sat}.

$$g_{diff}(T;\varphi) = -\delta_{mat}(\varphi) \nabla p_v(T;\varphi) \tag{8}$$

$$g_{adv}(T;\varphi) = \rho_{vap}(T;\varphi)\, u \tag{9}$$

$$g_{liq}(\varphi) = -D_w(\varphi) \frac{\partial w(\varphi)}{\partial \varphi} \nabla \varphi \tag{10}$$

$$\rho_{vap}(T;\varphi) = \frac{M_w}{RT}\, p_v(T;\varphi) \tag{11}$$

$$p_v(T;\varphi) = \varphi\, P_{sat}(T) \tag{12}$$

The latent heat flux density q_{lat} describes the latent heat transfer due to moisture sorption and desorption in the porous medium. This flux is the sum of a diffusive and an advective part (eq. 13). The sensible heat carried by liquid and vapour fluxes is commonly neglected toward the latent part in calculations (Künzel 1995).

$$q_{lat}(T;\varphi) = q_{lat-diff}(T;\varphi) + q_{lat-adv}(T;\varphi) = L_v g_{diff}(T;\varphi) + L_v g_{adv}(T;\varphi) \tag{13}$$

HAM-Lea outputs have been successfully compared with 1D numerical benchmarks from HAMSTAD project and with 2D experimental measurements (Belleudy 2015).

2.2 Boundary conditions

For air transfer, relative pressure is imposed at air inlets and a reference pressure at air outlets. The scalar product between air velocity and the outward-pointing normal vector *n* is set to zero on airtight boundaries. Boundary conditions for moisture transfer consist of relative humidity at air inlets and an inward imposed vapour diffusion flux at air outlets. This latter condition also holds for airtight but vapour permeable interfaces.

$$-n \cdot g_{surf} = \beta_{amb}(p_{v,amb} - p_{v,surf}) \quad (14)$$

Vapour tight boundaries are affected with a normal moisture flux equal to zero. Heat transfer boundary conditions consists of an imposed temperature at air inlets, an imposed inward heat flux at air oulet considering a sensible part due to the convective thermal resistance and a latent part due to moisture diffusion. This latter condition also holds for airtight but non-adiabatic interfaces:

$$-n \cdot q_{surf} = h_{amb}(T_{surf} - T_{amb}) + L_v \beta_{amb}(p_{v,surf} - p_{v,amb}) \quad (15)$$

Adiabatic boundary condition is a normal heat flux set to zero.

3 APPLICATION TO AN AIR LEAKAGE CONFIGURATION

3.1 Presentation of the case study

In this section, the numerical model is used to investigate the impact of coupled heat air and moisture transfer on a real-like air leakage geometry (fig. 1). It has been drawn from the previously mentioned measurement campaigns, which aimed to identify locations of wooden frame envelopes often subjected to air leakage.

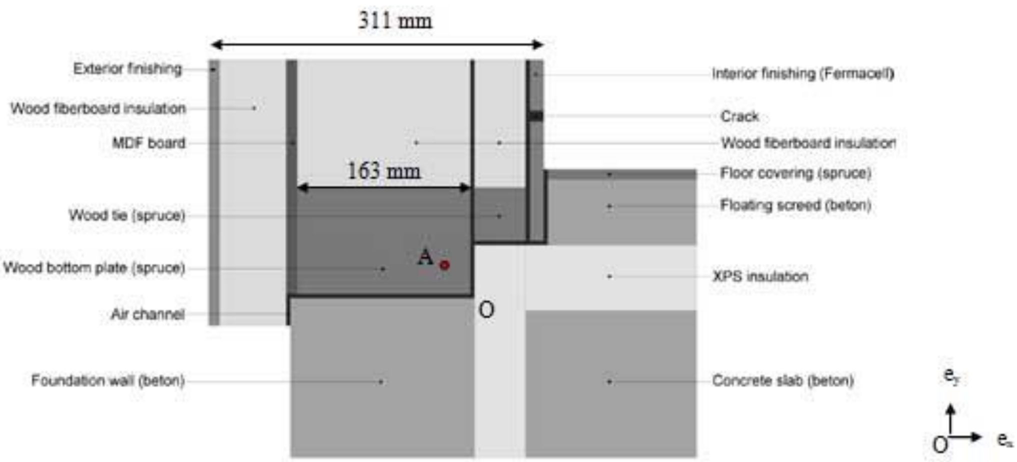


Figure 1: 2D section of the studied airtightness defect.
The point A (-30mm, 30mm) indicated above is used in section 4 to analyse the results.

Consequently to poor workmanship on the junction between wood bottom plate and foundation wall, flexible joints may be overlooked and the vapour barrier not sealed properly, which allows air to flow through the assembly via a thin air channel. To facilitate the modelling, it is assumed that no flexible joint remains, and the vapour barrier is removed to reproduce the effect of strong sealing discontinuities. The air computational domain is constituted by the 2mm air channel, the first layer of wood fiberboard insulation ($k_{mat} = 2.8 \cdot 10^{-10}\ m^2$) located behind the interior finishing, and a 10 mm diameter crack. The second and third layer of wood fiberboard insulation have been considered as airtight as they are not directly subjected to pressure gradient.

Table 1: Summary of the material properties. The readers are referred to the Fraunhofer IBP database, available in WUFI software, for a comprehensive description of the sorption isotherms and material properties

	μ factor [-]	λ_{mat} (φ=0.8) [W/(m.K)]	D_w (φ=0.8) [m^2/s]	c_{mat} [J/(kg.K]	ρ_{mat} [kg/m^3]
Beton C12/15	92	1.91	$217\ 10^{-12}$	850	2200
Fermacell	16	0.32	$160\ 10^{-12}$	1200	1153
Wood fiberboard	3	0.045	$200\ 10^{-12}$	2000	155
MDF board	12	0.12	$0.0745\ 10^{-12}$	2000	528
Spruce	4.3	0.28	$53.3\ 10^{-12}$	1500	455
XPS	100	0.03	Not cap. active	1500	40
Ext. finishing	8.1	0.93	$0.130\ 10^{-12}$	850	1360

3.2 Simulation scheme

This building assembly will be subjected to transient boundary conditions in both temperature and humidity. As we are mostly interested in long term behaviour, we use boundary conditions with yearly variations from WUFI database (fig. 2). Indoor temperature oscillates between 20°C in winter and 22°C in summer, whereas outdoor temperature ranges from 0°C in winter to 18°C in the summer. Humidity by volume is higher indoors than outdoors, but this tendency is reversed for relative humidity because of lower outdoor temperature.

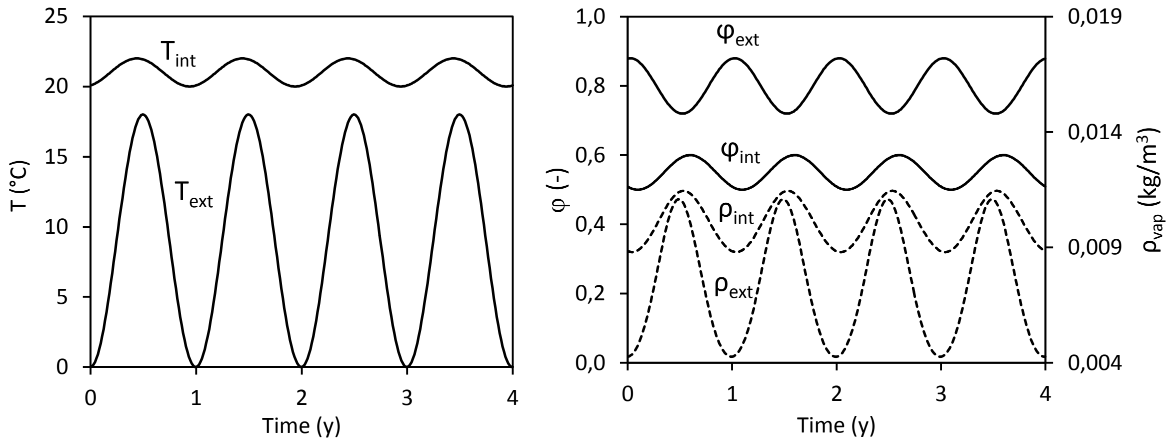


Figure 2: Yearly temperature (left) and humidity profiles (right) used as boundary conditions

In this study, both infiltration and exfiltration scenarii will be tested. For each scenario, two pressure differentials are tested – 0.1 Pa and 1 Pa – generating two airflow rates of 0.04 m^3/h and 0.4 m^3/h, respectively. The simulation scheme consists of one year of HM simulation (without airflow) used for initializing three-year HAM simulations. A 4-year HM simulation is also performed to make a comparison.

4 RESULTS AND DISCUSSION

4.1 Moisture

The impact of airflow on moisture field will be analysed by considering two indicators in a particularly sensible region: the averaged moisture content of the wood bottom plate w_{avg} and the moisture content evaluated at point A, noted w_A (see fig. 1).

It is shown on (fig. 3) that without airflow, the wood bottom plate only needs one year to reach hygric equilibrium, whereas two years are required in case of air leakage (infiltration or exfiltration). Observed tendencies differ in infiltration and exfiltration. When air infiltrates, w_{avg} decreases, indicating that the airflow dries up the assembly (fig. 3, left). As outdoor air is preheated in the assembly, air relative humidity decreases, hence it takes moisture from the assembly. On the contrary, a humidification occurs when air exfiltrates (fig. 3, right). Hot humid air is cooled down in the assembly, which increases its relative humidity and cause possible interstitial condensation if the dew point is reached. It also appears that the humidification process is more marked than the drying one for similar flowrates. This is firstly due to the non-linearity of the sorption curve (the more humid the material, the higher its hygric capacity) – and secondly to the additional liquid capillary flux becoming significant for high relative humidities. By drawing together with mean moisture contents the temperature difference between indoors and outdoors $\Delta T = T_{int} - T_{ext}$ we get information on how fast the hygrothermal field reacts to boundary conditions. Maximum moisture risks are expected for high ΔT. It may be noted that without airflow, the maximum moisture content is attained about three months after the maximum ΔT (20°C). In exfiltration for $\Delta P = 1$ Pa, this period is halved. Similar tendency is also observed in infiltration but to a lesser degree. This can be physically explained because without airflow, moisture is transferred exclusively by diffusion, whereas imposed airflow creates moisture advection which intensifies moisture migration. More local hygrothermal behaviour can be captured considering moisture content at a point A (fig. 4). These plots show that the hygrothermal field of the wood bottom plate is much more impacted in the vicinity of the air channel.

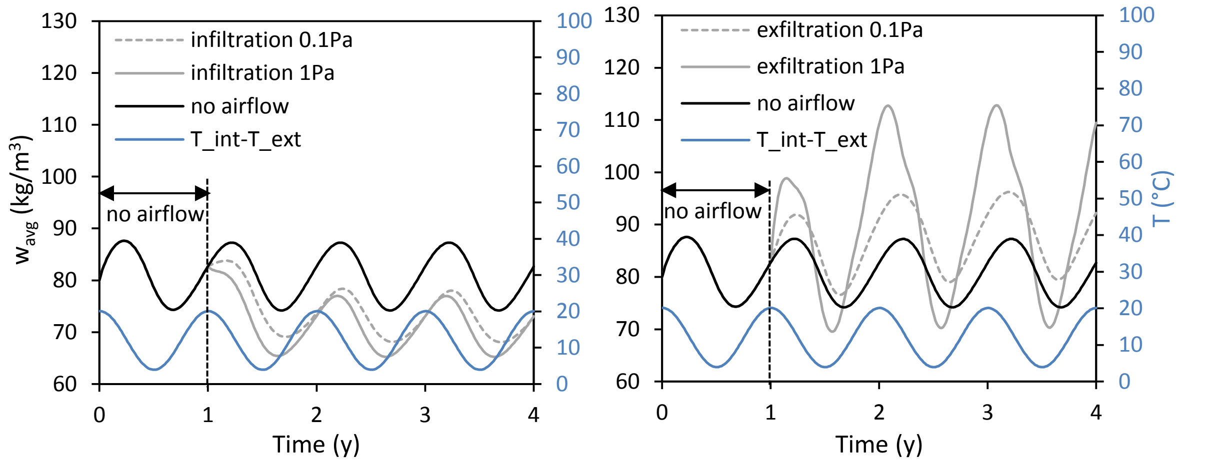


Figure 3 : Averaged moisture content of wood bottom plate for air infiltration (left) and exfiltration (right), over 4 years. The temperature difference between indoor and outdoor spaces is plotted in blue.

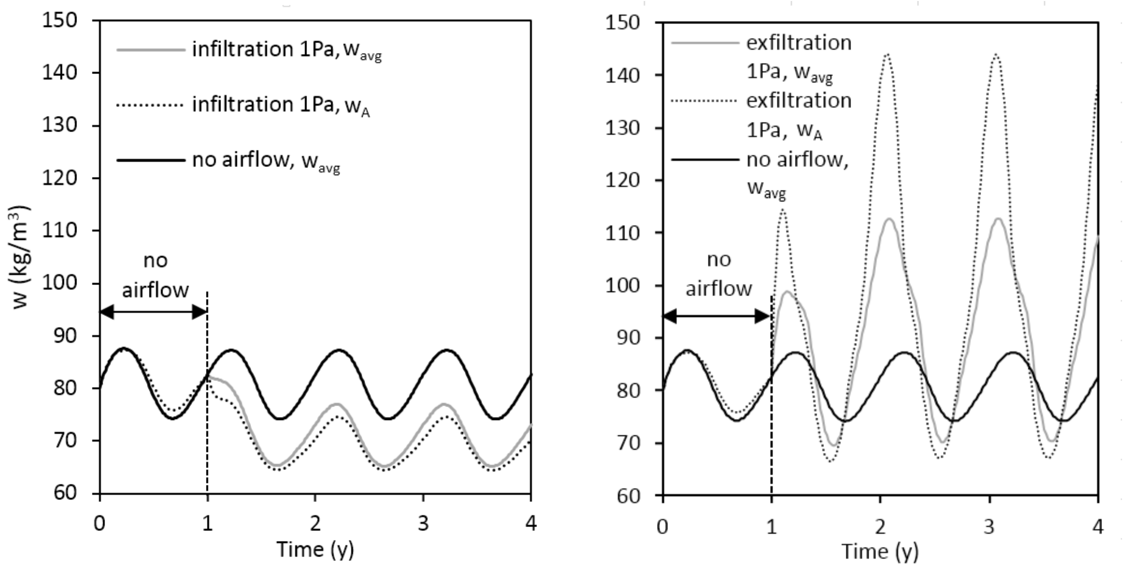


Figure 4: Comparison between averaged moisture content of wood bottom plate and ponctual moisture content evaluated at point A, for infiltrating and exfiltrating scenario.

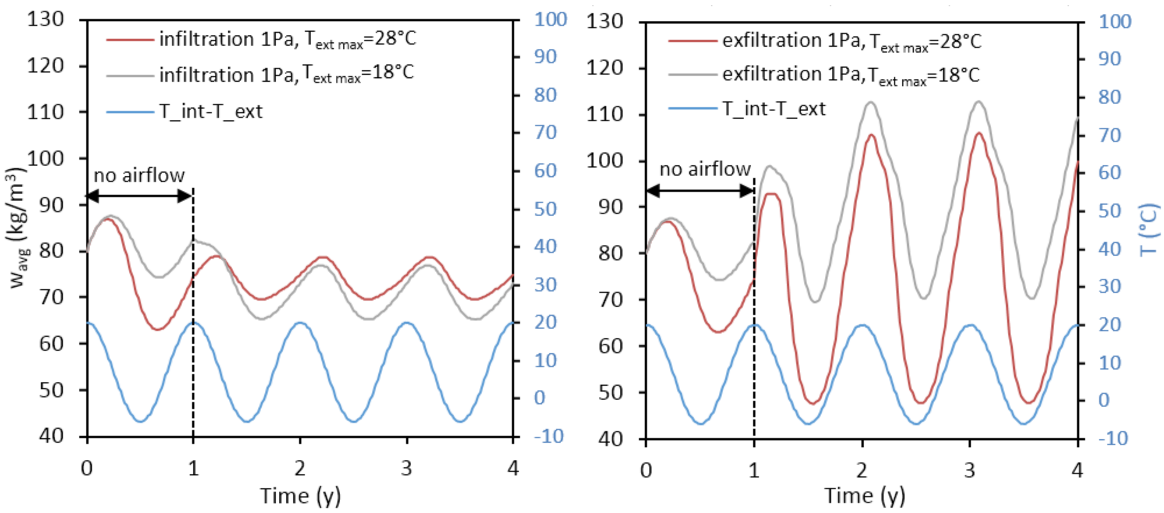


Figure 5: w_{avg} for air infiltration (left) and exfiltration (right), over four years. ΔT with a modified outdoor temperature varying in [0, 28°C] instead of [0, 18°C], is plotted in blue

In order to check the sensitivity of the hygrothermal field to boundary conditions, external temperature variations (fig. 2) have been changed to [0, 28°C] instead of [0, 18°C], the latter designated as the "base case". Variations of w_{avg} in infiltration and exfiltration for $|\Delta P| = 1\ \mathrm{Pa}$ are plotted in (fig. 5). Higher exterior mean temperature together with unchanged exterior relative humidity, result in a more humid exterior air (the exterior vapour pressure has increased). This explains why in infiltration w_{avg} of the modified case is higher than those of the base case. In exfiltration, the assembly is less humidified, as higher temperature level reduces the material relative humidity and thus its moisture content.

4.2 Heat Fluxes

In the following section, the impact of airflow on the hygrothermal field is analysed in term of energy. All heat fluxes included in the energy equation (eq. 1) are evaluated by integrating the respective heat flux densities - q_{cond}, q_{conv}, $q_{lat-diff}$ and $q_{lat-adv}$ - along the interior surface.
Choosing this surface enables to assess the heat transfer between indoor air and the building envelope: the sign of the calculated heat fluxes therefore indicates whether they contribute to heat the interior air (positive flux) or to cool it down (negative flux).
On the 2D section of the defect, the interior surface is composed by five segments (fig. 6). The integration is performed over each segment, what defines a total heat flux:

$$\Phi_{tot} = \int_{1}(q_{cond} + q_{lat-diff})\cdot e_y\,dx + \int_{2}(q_{cond} + q_{conv} + q_{lat-diff} + q_{lat-adv})\cdot e_y\,dx + \int_{3}(q_{cond} + q_{lat-diff})\cdot e_x\,dy + \int_{4}(q_{cond} + q_{conv} + q_{lat-diff} + q_{lat-adv})\cdot e_x\,dy + \int_{5}(q_{cond} + q_{lat-diff})\cdot e_x\,dy \qquad (16)$$

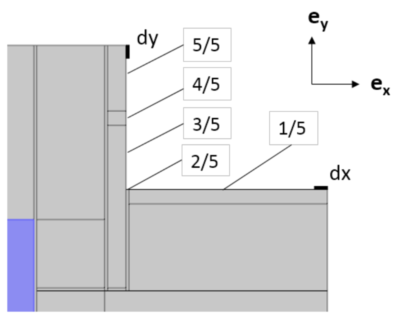


Figure 6: Detail of the five segments composing the interior surface of the assembly

The different heat flux densities have been previously defined in (eqs. 6, 7 and 13). However, the expression of the convective dry air flux and the advective vapour flux will be corrected to the interior conditions, as stated by (eqs. 17 and 18). This does not change the contribution of q_{conv} and $q_{lat-adv}$ in (eq. 1), as only the divergence of these fluxes is needed.

$$q_{conv}(T) = \rho_{air} c_{p_{air}} u\,(T - T_{int}) \tag{17}$$

$$q_{lat-adv}(T;\varphi) = L_v[\rho_{vap}(T;\varphi) - \rho_{vap}(T_{int};\varphi_{int})]u \tag{18}$$

Choosing T_{int} and φ_{int} as reference temperature and relative humidity is consistent from building physics perspective, as they are set values to maintain for ensuring thermal and hygric comfort. Hence, an infiltrated airflow injected at T_{int} in the indoor space results in a zero convective heat flux, whereas an infiltrated airflow injected at T_0 with $T_{ext} = T_0 < T_{int}$ represents a deperditive flux equal to $\rho_{air} c_{p_{air}} u(T_0 - T_{int})$, which must be compensated in order to maintain T_{int} in the building. Keeping the same approach for exfiltration, the convective and the advective heat fluxes are zero, as the temperature and relative humidity of the air inlet (located on the interior surface) are imposed at T_{int} and φ_{int} as boundary conditions. This statement is not astounding, as the associated heat loss advected by airflow is already taken into account on the building zone scale, at air inlet locations. Similarly, if the infiltrated airflow contains less moisture than the interior space, i.e. $\rho_0 < \rho_{int}$, this must be compensated by water vapour generation inside the building zone to maintain the desired humidity, hence a deperditive heat flux equal to $L_v[\rho_{vap}(T_0;\varphi_0) - \rho_{vap}(T_{int};\varphi_{int})]u$.

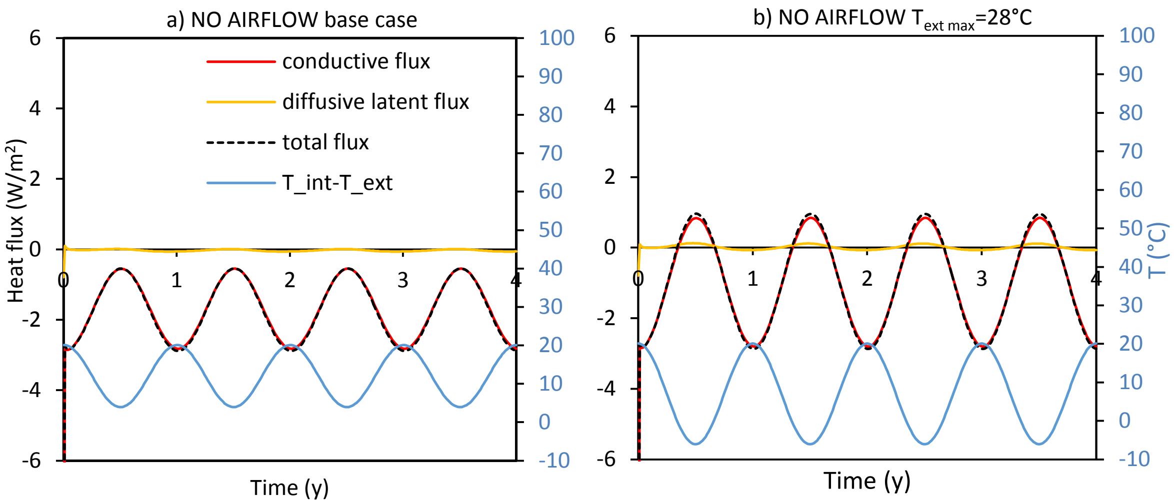


Figure 7: Heat fluxes without airflow over four years, in the base case (a.), in the case of modified $T_{ext\ max}$ (b.)

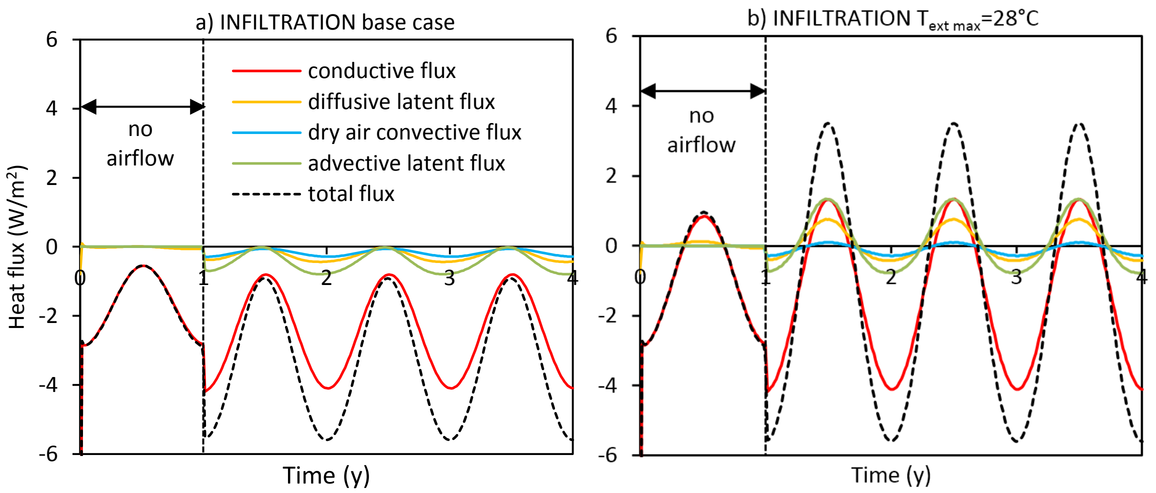


Figure 8: Heat fluxes for the infiltration scenario, in the base case (a.), in the case of modified $T_{ext\ max}$ (b.)

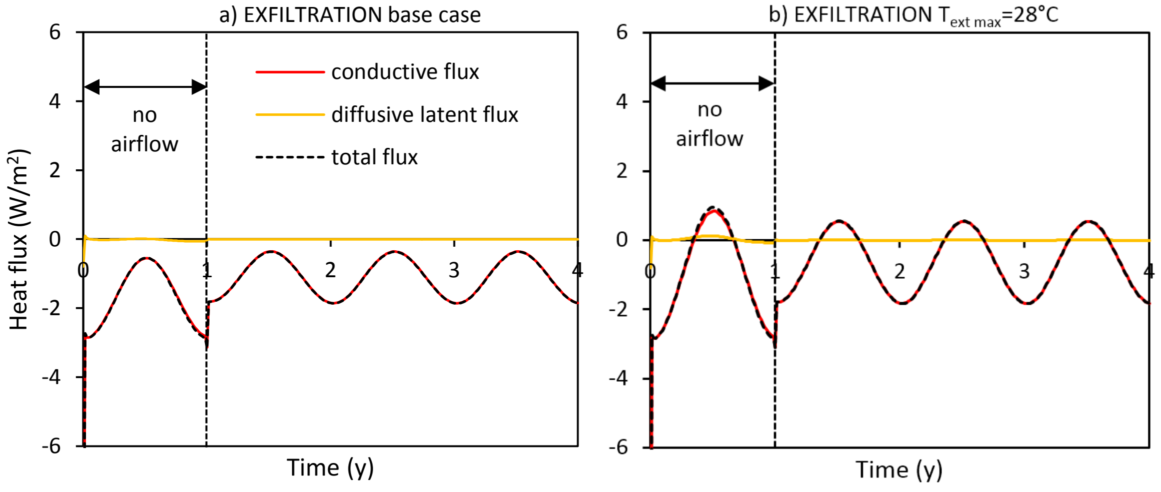


Figure 9: Heat fluxes for the exfiltration scenario, in the base case (a.), in the case of modified $T_{ext\ max}$ (b.)

The heat fluxes are plotted for a 4-year HM simulation without airflow (fig. 7), as well as for infiltration and exfiltration scenarii with $\Delta P = 1$ Pa (figs. 8. and 9. respectively). Contrary to the moisture response, it appears that even subjected to air transfer, the assembly reaches thermal equilibrium in the first year, which illustrates that heat transfer has much smaller time constants than moisture transfer. The following analysis firstly discusses the base case without airflow, then the impact of infiltration and exfiltration (figs. 7a. 8a. and 9a.). Secondly, the impact of external temperature variation is investigated (figs. 7b. 8b. and 9b.).

Without airflow, in the base case, the total heat loss is mainly constituted by the conductive flux, with a mean value of -1.6 W/m (fig. 7a.). The maximum conductive heat loss is logically attained for the maximum ΔT. The diffusive latent flux is negligible, probably because of the high vapour resistance of the whole multi-layered building assembly.

When air infiltrates at $\Delta P = 1$ Pa, the mean conductive heat loss increases to -2.5 W/m (fig. 8a.). This is coherent with the fact that infiltration cools the wall and thus concentrates the temperature gradient in the vicinity of the interior surface, increasing the conductive heat loss. The convective dry air flux is deperditive as well (mean value of -0.17 W/m), as the temperature at the air outlet is lower than T_{int}. Considering latent fluxes, the diffusion flux is still of minor importance (mean value of -0.25 W/m). It is higher than without airflow, as the vapour pressure and temperature gradients are shifted to the interior surface. The advective latent heat flux is a heat loss (mean value of -0.45 W/m), because infiltrated air exiting the interior interface is dryer than ambient air. Overall, the total heat loss in infiltration (mean value of -3.3 W/m) is twice higher than the one without air.

When air exfiltrates, the conductive heat loss decreases to -1.1 W/m (relative to the flux without airflow), as the interior side of the assembly is heated up by the airflow, hence temperature gradient decreases. It is reminded that the convective dry air flux and the advective latent flux are zero according to the choice of T_{int} and φ_{int} as reference temperature and relative humidity. The diffusion latent flux is lower than the one in infiltration, because the temperature and vapour pressure gradient are shifted to the exterior side. With this approach, the total heat loss in exfiltration (mean value of -1.1 W/m) is lower than in infiltration or without airflow.

Changing external temperature variation from [0, 18°C] to [0, 28°C], decreases the mean ΔT, hence it lowers the mean conduction heat loss without airflow (fig. 7b.). When T_{ext} is superior to the interior temperature, the heat flux is reversed, turning the heat conduction flux into a heat gain (positive value).

Similarly in infiltration (fig. 8b.), the advective latent heat flux is positive shortly after ΔT is minimal (meaning that $T_{ext} = 28$ °C and $T_{int} = 22$ °C), which indicates that infiltrated air at the interior surface contains more moisture that the indoor air. The modified T_{ext} leads indeed

to a more humid exterior air. The diffusive latent heat flux is lower compared to the base case, because the vapour pressure gradient between indoor and outdoor is lower.
In exfiltration (fig. 9b.), the effect of the reduced temperature gradient appears clearly, similarly to the case without airflow (fig. 7b.).

5 CONCLUSION

A numerical model for simulating HAM transfer through complex wall assemblies including permeable porous material and thin air channels has been presented. This tool has been used to assess the impact of leaking air on the hygrothermal field in an airtightness defect, subjected to classical boundary conditions. Additional moisture risk is expected in case of air exfiltration, whereas air infiltration has a drying effect on the assembly. Higher external temperature slightly increases moisture storage in infiltration, and reduces it in exfiltration. An approach to calculate sensible and latent heat fluxes through the defect is proposed. To obtain more quantitative results about the energy impact of air leakage, it seems relevant to consider the building zone scale, where airtightness defects, air inlet and outlet can be combined. This could be the aim of further research work.

6 ACKNOWLEDGEMENTS

This work is financially supported by the French Environment and Energy Management Agency (ADEME), the Building Scientific and Technical Centre (CSTB), the French National Agency (ANR) through its Sustainable Cities and Building programme (MOB-AIR project n°ANR-12-VBDU-0009), and the Région Rhône-Alpes.

7 NOMENCLATURE

u [m/s]	air velocity	$\pm_{mat}$ [kg=(s:m:Pa)]	vapour permeability of material
k_{mat} [m^2]	intrinsic permeability of material	¼$_{mat}$ [kg/m^3]	dry density of material
$\dot{H}_{air}$ [Pa.s]	dynamic viscosity of dry air	– [kg/(s.m^2.K)]	moisture surface film coefficient
¼$_{air}$ [kg/m^3]	dry air density	h [W/(m^2.K)]	heat surface film coefficient
$c_{p_{air}}$ [J/(kg.K)]	specific heat of dry air	L_v [J/kg]	vapour latent heat of sorption
c_{mat} [J/(kg.K)]	spec. heat of dry material	D_w [m^2/s]	moisture diffusivity
c_w [J/(kg.K)]	spec. heat of liquid water	R [J/(mol.kg)]	universal gas constant
$\dot{H}$ [˜]	vapour resistance factor of material	M_w [kg/mol]	molar mass of water
H [kg=m^3]	specific enthalpy	P [Pa]	total air pressure

8 REFERENCES

Belleudy, Clément. 2015. “Modélisation Des Transferts D’air et Leur Impact Sur Le Comportement Hygrothermique Des Bâtiments.” Ph.D. Thesis, Univ. Grenoble Alpes.

Belleudy, Clément, Ahmad Kayello, Monika Woloszyn, Hua Ge, Paul Fazio, Marx Chhay, and Daniel Quenard. 2014. “A Heat-Airflow Model for Simulating the Effects of Air Leakage on the Temperature Field in Porous Insulation.” In 10th Nordic Symposium on Building Physics, 79–86. Lund, Sweden.

Buchanan, C. R., and M. H. Sherman. 2000. “A Mathematical Model for Infiltration Heat Recovery.” Lawrence Berkeley National Laboratory.

CETE de Lyon. 2010. “Carnets Prebat Mininfil, Mémento Étanchéité - Construction Ossature Bois - Isolation Thermique Intégrée.”

Janetti, Michele Bianchi. 2014. “Assessment of the Moisture Risk in Constructions Including Convection inside Air Cavities.” In 10th Nordic Symposium on Building Physics, 1038–44. Lund, Sweden.

Janssens, A. 1998. “Reliable Control of Interstitial Condensation in Lightweight Roof Systems: Calculation and Assessement Methods.” Ph.D. Thesis, Departement of Civil Engineering, KU Leuven, Belgium.

Künzel, H.M. 1995. “Simultaneous Heat and Moisture Transport in Building Component.” Fraunhofer IBP.

Langlais, Catherine, Eric Arquis, and Dave J. McCaa. 1990. “A Theoretical and Experimental Study of Convective Effect in Loose-Fill Thermal Insulation.” Insulation Materials: Testing and Applications, ASTM STP 1030, 290–318.

Langmans, Jelle. 2013. “Feasibility of Exterior Air Barriers in Timber Frame Construction.” Ph.D. Thesis, Departement of Civil Engineering, KU Leuven, Belgium.

Nespoli, Lorenzo, M. Bianchi Janetti, and Fabian Ochs. 2013. “Comparing Different Approaches for Moisture Transfer inside Constructions with Air Gaps.” In COMSOL Conference Rotterdam.

CHARACTERIZATION OF SEALANTS AND EXPANDING FOAMS

Filip Van Mieghem

Soudal NV
Everdongenlaan 18-20
B-2300 Turnhout, Belgium
vanmieghem@soudal.com

1. IMPORTANT UNIMPORTANCE

Constructions joints occur everywhere where several construction materials meet. That's many meters that need to be taken care off, respecting the function that the materials or the joint need to fulfill. Typically a joint filled with sealing material has to account for water tightness, or is applied for esthetical reasons. Insulating materials can also be injected for thermal or acoustical insulation. A combination of joint materials often guarantees more advanced functions, fire resistance is a typical example. Of course a combination of functions can also be fulfilled and nearly every time the jointing material needs to take up movement of the surrounding construction elements, which is definitely the case for an expansion joint. These requirements call for specialized quality products. Airtightness certainly adds to this list, putting a new challenge to construction products and jointing materials.
It is quite clear that airleaks often occur where construction elements meet and that they can account for a big part of the total amount of leaks. Air lost in and around windows for instance can easily make up 30 to 40% of all airleaks. Window connection joints are notoriously infamous for being thermal bridges too. No wonder Soudal primarily concentrated on these area's offering solutions based on product combinations that are adapted to local building practice and tested accordingly.

2. CHARACTERIZING PRODUCTS

Characterizing building products in terms of airtightness however is not so easy. Basically EN 12114 provides a rough procedure to test airtightness of a building product in laboratory conditions. It obviously doesn't specify how test samples need to be prepared or how the test should be set up because of the vast number of construction materials.

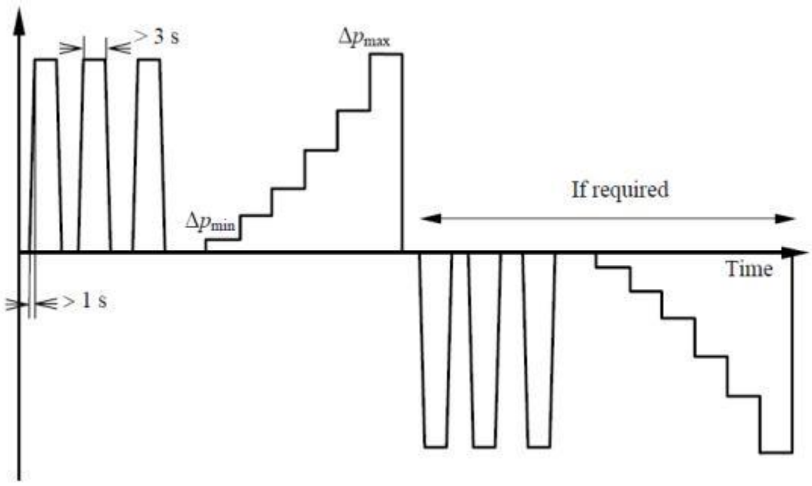


Figure 2: Test procedure provided by EN 12114

A lot of building products are therefore not tested according to this norm as they are for instance accepted to be air tight from their nature, as this the case for sealants for instance. Others rely on measured Sd-values although vapourtightness and airtightness are not the same. A material can be vapour open yet still be airtight. PU-foams are a typical example. That's why quality foams are tested on airtightness according to EN 12114 all the same, providing proof that they can combine thermal and acoustical insulation with airtightness. Newer generation flexible PU-foams can even come with test reports on their elasticity, as is the case for Flexifoam ®. For sealants this essential characteristic has been normed many years ago. EN ISO 11600 provided classes which correspond to movement capacity, ranging from 7,5 to 25. The new CE marking for sealants is to a large extent based on the still existing EN ISO 11600, generalizing the use of these classes and imposing clear communication with regards to the characteristics on the packaging. Movement capacity should be in line with the intended application, so that a sealant remains intact (cohesion) and keeps on adhering to the adjacent materials (adhesion).

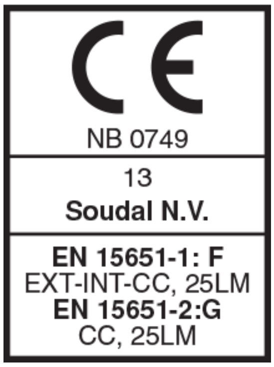


Figure 30: CE marking for sealants: reference for end users

Movement capacity of building materials and jointing materials in particular mean that will perform well for many years, and test reports, technical data sheets and packaging will be of significant help to the (professional) end user. Ultimate proof on how well certain product combinations will perform over the years is almost impossible to organize though, since the number of parameters and boundary conditions mean that there are just too many combinations to be tested. However, the German MO-01/1 directive from ift Rosenheim (Institut für Fenstertechnik) comes close to enabling this. A given product combination for window to wall connection is tested on water-and airtightness before AND after ageing, giving a clear indication on durability of the solution tested. Several combinations within the

Soudal Window System have been granted an OK after being submitted to this thorough and extensive test procedure.

3. RESEARCH

Apart from legal requirements, building contractors and professional end-users look for products that are reliable, user-friendly and time-saving at an acceptable cost. To match these typical requirements with the new demands on airtightness, Soudal invests a lot in R&D, constantly trying to improve products and solutions. A typical innovation along this path is Soudatight LQ, which will later be followed by Soudatight SP (sprayable version). Soudatight LQ is a fiber reinforced polymer dispersion, ready for use – to be applied with a brush, and curing to an elastic, vapour and airtight membrane which can be painted or plastered after drying. It is odourless, solvent free and low-emission (meets the high EC1 Plus standard). It is used to be applied in and around window joints, floor-wall joints, wall-ceiling joints and roof joints. Due to the fibre structure, cracks up to a few millimetres wide can be sealed easily. The sprayable version of course offers the added benefit of treating larger area's in short time. New requirements and user friendliness can go hand in hand!

4. CONCLUSIONS

Solutions and products that enable high levels of airtightness are available. However, due to the many parameters and situations, it is very difficult for the industry to come up with test reports for every imaginable situation, yet decent market players will try to do so whenever possible. Furthermore, for a product to perform, craftsmen or end users should always check the manufacturers advice on how to use it, and check technical data sheets as much as possible.

LABORATORY INVESTIGATION ON THE DURABILITY OF TAPED JOINTS IN EXTERIOR AIR BARRIER APPLICATIONS

Jelle Langmans[*1], Tadiwos Zerihun Desta[2], Lieven Alderweireldt[2] and Staf Roels[1]

[1] *Department of Civil Engineering*
Building Physics Section, University of Leuven
Kasteelpark Arenberg 40 – bus 02447
BE-3001, Heverlee, Belgium
**Corresponding author:*
jelle.langmans@bwk.kuleuven.be

[2] *Redco nv*
Kuiermansstraat 1
B-1880 Kapelle-op-den-Bos, Belgium

ABSTRACT

In timber frame construction in Europe air barrier systems are typically realised at the interior side of the building envelope. Yet in some applications such as renovation projects it can be easier to provide the air barrier layer at the exterior. This way, the air barrier system – typically board materials in which the joints are sealed with tape – is exposed to outdoor weather conditions. The aim of the present article is to investigate the impact severe climatic conditions on the airtightness of typical taped joints. The airtightness of thirty two wood-fibre cement board samples have been investigated. Each specimen has a 2mm wide joint. Two different kinds of commercially available tapes were used to seal the joints. Airtightness of all specimen has been tested before and after accelerated aging procedures that mimics real exterior climate conditions. Three different aging protocols were selected: 1) temperature cycles, 2) temperature, rain and frost cycles and 3) UV exposure under high humid conditions. For the first two test conditions, the specimens had a size of 0.71m by 0.71m. Due to sample size restrictions of the UV exposure cabin, the specimens of the third test run were smaller (0.35m by 0.35m). All specimens had thickness of 12mm. The present paper will discuss the durability of the two tapes by comparing the air permeability of the specimens before and after the three accelerated aging exposures.

KEYWORDS

durability, airtightness, tapes, exterior air barrier

1 INTRODUCTION

Robust air barrier systems are one of the prerequisites to achieve energy efficient buildings. In light weight construction, the air barrier layer is commonly positioned close to the interior finish. This layer often also incorporates the vapour barrier functionality. Contrary to the popular practice, in some projects (e.g. renovations) it can be easier and faster to place the air barrier layer at the exterior side of the building envelope (Langmans et al. 2010). This implicates that the taped joints are exposed to outside environmental conditions. Especially

during the construction phase when the air barrier layer gets the least protection, these taped joints need to withstand high levels of temperature, humidity and UV radiation.
The available international literature regarding building airtightness durability is very limited. Few studies report variations in the global building airtightness within a time interval of several months or years. Bracke et al. (2013) showed a reasonably stable overall air leakage level for buildings with a high level of airtightness. In contrast, others reported significant deviations from the original value with the building ages (Hansén, 2012), or seasons (Borsboom and de Gids (2012) and Kim et al. (1986)). However, these studies evaluated the global building airtightness and do not provide detailed information regarding the key parameters influencing airtightness durability. To the authors knowledge only two studies have been investigating the durability of tapes used for building airtightness applications. Both Ackermann (2012) and Gross & Maas (2011) developed a method based on the 180° peel test according to BS EN 1939:1997[1]. This standard is used to investigate the peel adhesion of self-adhesive tapes. The prepared specimen are loaded into a tensile testing machine in which the tape is peeled from the substrate in a 180° direction. This test essentially provides information on the adhesion strength of the taped connection.
Gross & Maas (2011) studied the peel force for several tapes and substrate materials (foils, wood and fibre cement boards). Before the peel test, artificial aging of the specimen was established by pre-conditioning the samples at a temperature of 65°C and 80% relative humidity. Their results clearly show that the level of peel force depends on the substrate and the curing time of the adhesive. Moreover it is shown that the stress resistance of the adhesives in combination with a PE foil is significantly lower than with other bonding materials. Gross & Maas (2011) also showed that solvent-free adhesives correspond to higher values in peel force. In addition, the authors remarkably noted that with progressive accelerated aging, the majority of the adhesives analysed show an increased peel force.
The second study in this kind was performed by Ackermann (2007,2012) at FIW in Munich.
Similar to Gross & Maas (2011), Ackermann (2012) conditioned the samples before the peel test at a temperature of 65°C and 80% relative humidity. The impact of 21,40 ,80 and 120 days preconditioning on the adhesive strength was investigated. Instead of a static peel test, he developed a method which allows alternating the loads in order to imitate the dynamic wind conditions. Yet his paper mainly focusses on the development of the test method rather than on the research findings.

In summary it can be stated that the available literature on the durability of air barrier sealing products is very limited. The only information currently available is related to the relation of static artificial aging tests on the peel force of the sealing. The applied methods are based on standardised peel tests, and thus, provide only information on the adhesive strength of the connecting. These methods can be applied to compare the adhesion durability of different kinds of taped joints. Yet the outcome of such tests do not provide information on the impact of artificial aging on the air permeability of the taped connection.
The aim of the present study is to explore the possibilities to investigate the impact of weather conditions on the actual air permeability of taped joints. Samples with a taped joints will be tested before and after artificial aging. The study explores the behaviour of two different types of tapes on a wood fibre cement board substrate. Herein three different artificial aging methods will be applied: a) thermal loads, b) hygrothermal loads and c) combined UV and vapour loads.

[1]BS EN 1939:1997 self-adhesive tapes – Measurements of peel adhesion from stainless steel or from its own backing

2 TEST SETUP AND METHODOLOGY

The air permeability of in total thirty two samples with taped joints are measured before and after conditioning. Two different tapes, two spacing materials and three different sample conditioning schemes are investigated. The following sections describe the applied materials, the conditioning schemes and the air permeability apparatus.

2.1 Test materials and samples

The tests are performed on medium density wood-fibre boards[1] with taped joints. These boards are composed of a coarse core, faced on each side with a fine smooth top layer. They consist of mainly Portland cement, water and wood fibres. The test samples are either 0.71 by 0.71 m² or 0.35 by 0.35 m² depending on the sample conditioning (see section 2.2). Figure 1 illustrates that each specimen consists out of two part which are connected with two spacers leaving a joint of 2 mm in between. Two different spacer materials are used: a) aluminium and b) wood. These two spacer material are chosen because of their different thermal and hygric expansion coefficients. This will induce different loads on the tapes during the sample conditioning (see section 2.2).

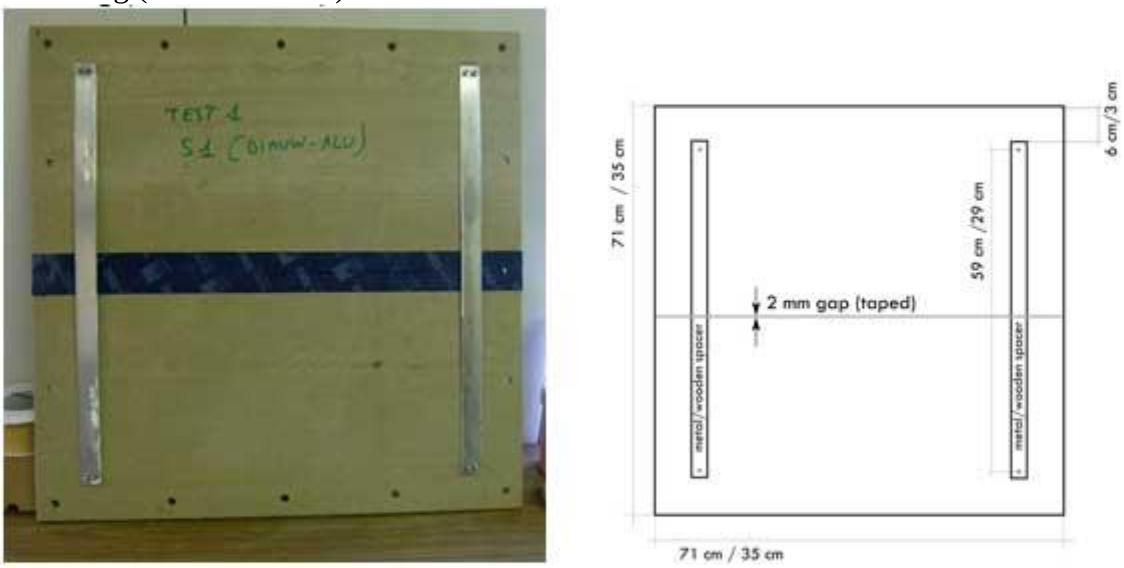


Figure 5: Test samples: left) sample of 0.71 by 0.71 m with aluminium spacer and right) dimensions for the big/small samples.

The tests are performed for two different commercially available airtightness tapes: Tape A) and Tape B). Table 1 summarizes the sample configurations. Next section will outline the conditions to which these samples are exposed.

Table1: Summary of test program.

TEST SERIES	TAPE	Spacer	Number of samples
A	Tape A	Aluminium	8
B	Tape B	Aluminium	8
C	Tape A	Wood	8
D	Tape B	Wood	8

[1]Duripanel of Eternit

2.2 Sample conditioning

This section outlines the three different conditioning schemes to which the samples are exposed (see Table 1).

Test 1: Temperature cycles

The first batch of samples (0.71 m by 0.71 m) are exposed to temperature cycles at ambient relative humidity levels. The aim of this conditioning scheme is to expose the samples to high temperature differences. For two weeks cycles of 24 hours at 70°C are followed by 24 hours at 15°C. Indirectly, in addition to the thermal load, this scheme corresponds to a mechanical load. This is induced by the thermal expansion of the spacer which is only fixed at the end points. For aluminium, which has a thermal expansion coefficient of 23.2 10^{-6} m/m/K this corresponds to a displacement of 0.74 mm. For the wooden spacer with a thermal expansion coefficient of 5 10^{-6} m/m/K this is only 0.2 mm.

Test 2: Temperature, rain and frost cycles

The second conditioning scheme has an increased load in that the samples (0.71 by 0.71) are now also exposed to water and freezing conditions. First the samples are exposed to 40 heat-rain cycles (3 hours temperature increase (70°C), 1 hour rain and 2 hour repose). Thereafter two frost cycles are imposed (8 hours 50°C followed by 16 hours of freezing (-20°C). This scheme corresponds to half of the normal hygrothermal cycles of ETAG004[1].

Test 3: UV-light cycles

During the construction phase exterior air barrier tapes will, in addition to hygrothermal loads, be exposed to solar radiation. This is often combined or alternated with periods of high moisture contents due to rain or condensation on the building envelopes surface. These conditions are simulated in the third conditioning scheme in which samples (0.35 by 0.35 m) are exposed to UV-light alternated with periods of vapour exposure based on ASTM G-154[2]. Herein the samples are exposed to 56 cycles of 8 hours UV-exposure (40°C) followed by 4 hours of vapour exposure (60°C). The UV-light is transmitted by a UVB-313EL[3] lamp corresponding to wavelengths between 280-400nm and an intensity of 0.7 W/m²/nm. All three accelerated climatic exposures are summarised in Table 2.

Table 6: Summary of test conditions.

Test series	Type	Total time	Conditions
TEST 1	Temperature	2 weeks	6 x (24h 70°C and 24h 15°C @30% RH)
TEST 2	Temperature, rain, frost	12 days	40 x (3h 70°C - 1h rain - 2h repose) - 2 x (8h 50°C - 16h -20°C)
TEST 3	UV-exposure, vapour	4 weeks	56 x (8h UV (40°C) and 4h vapour exposure (60°C))

[1]Guideline for European technical approval of external thermal insulation composite systems with rendering.
[2]Operating Fluorescent Light Apparatus for UV Exposure for Nonmetallic Materials
[3]QUV weathering tester (Q-lab)

2.1 Air permeability test-setup

The air permeability of the samples are measured in laboratory conditions (20°C,50%RH) both before and after the sample conditioning. Two similar test-setups have been applied for these measurements. The first test setup was designed to measure the air permeability of the samples with a dimension of 0.71 by 0.71 m. This box is applied for the samples corresponding to TEST 1 and TEST 2 (see table 1). In addition a smaller box with a test frame of 0.35 by 0.35 m is used for the samples which were exposed to the UV-light (TEST 3). Both setups, which are designed according to the EN12114 standard, consists of a metal frame box open at one side to install the specimen to be measured (Figure 2).
To avoid unwanted air leakages through the perimeter joints between specimen and airtight boxes, closed cell EPDM with a thickness of 2 cm on both sides of the specimen have been used to seal the specimen airtight with a metal frame against the airtight box.

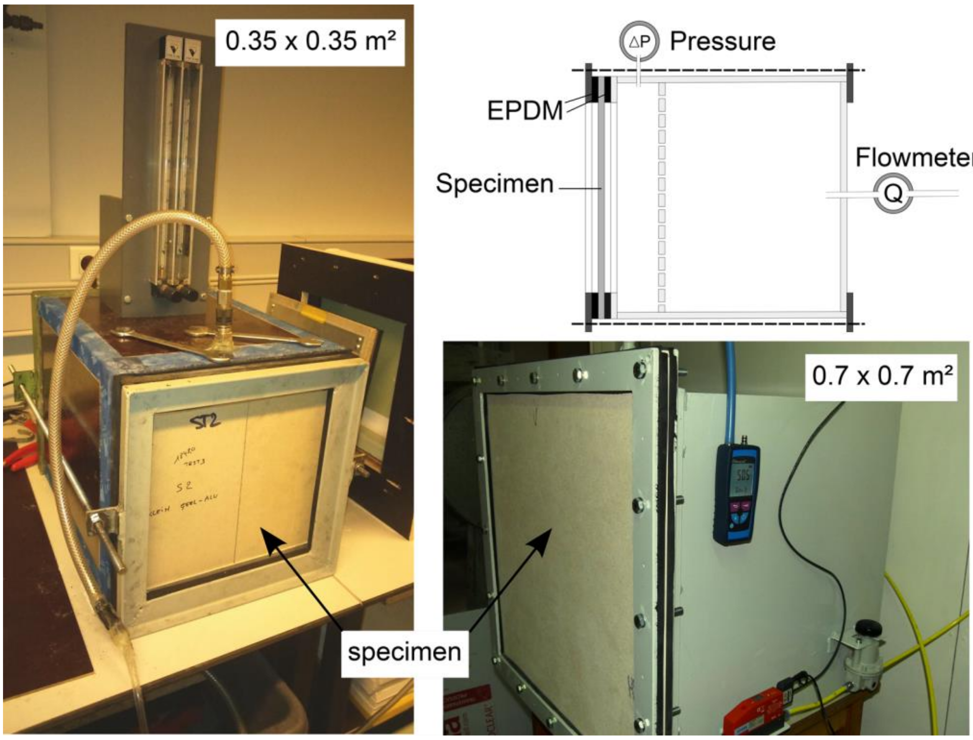


Figure 2: Laboratory test set-up.

After installing the specimen on the airtight box, over-pressure was created in the box. This resulted in an air flow passing through the specimen. By stepwise increasing the pressure difference across the specimen and measuring the corresponding air flow rate and pressure difference across the specimen a data set was obtained. The airflow g_a (m³/m²/h) can be expressed as an air permeance K_a multiplied by the pressure drop across the specimen ΔPa (Hens 2007):

$$g_a = K_a \Delta P_a \tag{1}$$

The construction junctions are assumed to be a parallel circuit of air resistances. Hereby, the air permeance of a joint K_{joint} (m³/m/h/Pa) can be deduced from the measured air permeance of the specimen K_{spec}, given that the air permeance of the material K_{mat} (m³/m²/h/Pa) is known from the small test-setup (Hens, 2006):

$$K_{joint} = \frac{(K_{spec} - K_{mat}).A_{spec}}{l_{joint}} \tag{2}$$

For the small test setup a pressure gauge 4 DG-700, with an accuracy of 1% was applied. The flow rate was determined with a Vögtlin variable area flow meter. In a range from 0.02 m^3/h to 0.900 m^3/h the flow rate could be measured with an accuracy of 2%. The overall leakage of the test setup itself, including the ductwork connections, was estimated to be 0.0035 m^3/h at 50 Pa. For the bigger test-setup a BlueLine S2600 pressure transducer with an accuracy of 1% was used. Here a digital flowmeter (Vögtlin GSM-C) with a measuring range from 0 m^3/h to 0.36 m^3/h and an accuracy of 0.3% was applied. The leakage of the bigger apparatus was 0.0032 m^3/h at 50 Pa.

3 TEST RESULTS

This section discusses the measuring results. First, the air permeability of the board material and the taped joints before the ageing procedure will be outlined. Thereafter the impact of the 3 artificial ageing tests on the specimen's air permeability will be presented.

3.1 Air permeability of the board and taped joints

First the air permeability of the 12 mm wood fibre cement board is measured on specimen without joints. This air permeability of the boards (1.04 10^{-4} $m^3/m^2/h/Pa$) is than subtracted from the air permeability of the specimen with taped joints in the same test-setup before artificial ageing according. The results reveal that the air permeability of the taped joints is very low; 3.1 10^{-6} $m^3/m/h/Pa$ for Tape A and 3.9 10^{-7} $m^3/m/h/Pa$ for Tape B.
In the following section the impact of the artificial ageing on the permeability of the taped joints will be discussed.

3.2 Impact of artificial ageing

The previous section showed that the air permeability of the taped joints is very low before the samples are exposed to artificial ageing for both tapes. This section investigates the impact of the 3 accelerated ageing protocols on the joints air permeability. Figure 4 plots the difference between the air permeability before and after the artificial ageing test. The bars in the figure correspond to the averaged values and the error bars refer to the minimum and maximum value.

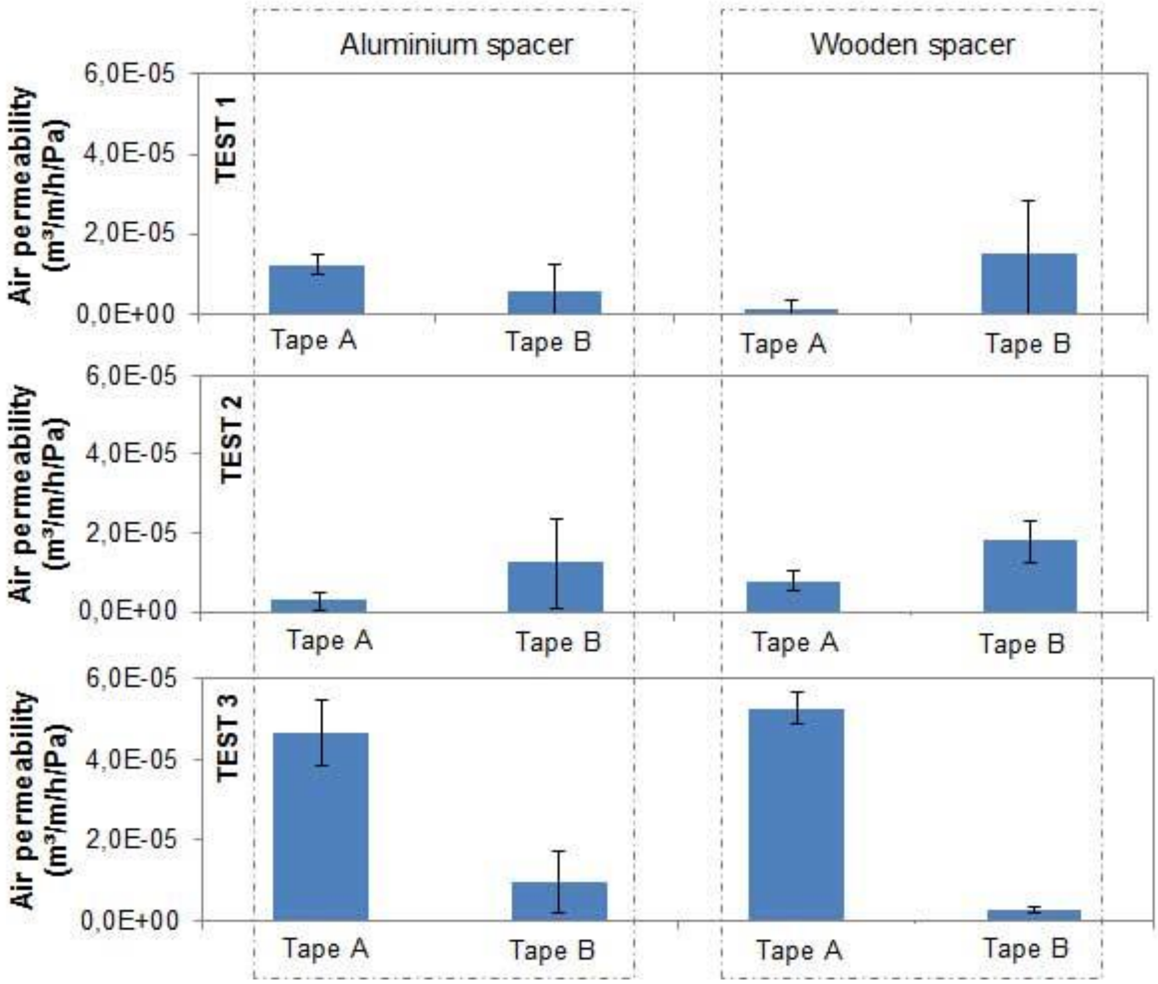


Figure 4: Increase in air permeability of the joints after the artificial ageing of the samples.

The results indicate that the increase in air permeability is limited to 2 10^{-5} m³/m/h/Pa for the first two ageing procedures. Nevertheless the second procedure is much more severe including rain and frost cycles, the impact on the permeability remains in line with the first procedure. In addition no significant differences between the different spacing methods nor the different tapes is noticed. For the third ageing protocol, in contrast, Tape B seems to perform better than Tape A. However, it should be noted that the increase of the air permeability is limited to 4-6 10^{-5} m³/m/h/Pa which is still very low.

4 DISCUSSION AND CONCLUSIONS

Exterior air barrier systems can – in contrast to traditional interior barriers - be exposed to severe weather conditions. These outdoor conditions may have an impact on the durability of sealed connections. The article at hand proposes a methodology to study the durability of taped joints based on air permeability testing and artificial ageing tests. In this study two different commercially available tapes are applied which are exposed to three accelerated ageing protocols ranging from a) temperature cycles, b) temperature, rain, frost cycles up to c)

UV and vapour exposure. The joint's air permeability was measured before and after being exposed to these conditions.
The results reveal that the permeability increase is limited for both tapes tested. For the temperature, rain and frost cycles the increment stays below 2 10^{-5} m³/m/h/Pa. For the UV and vapour cycles a slightly higher impact was noticed for Tape A (4-6 10^{-5} m³/m/h/Pa). Yet it should be stressed that this increase is still very small. To get an idea of the order of magnitude this increase in permeability can be translated to a share of the overall n_{50}-value of a building. Langmans et al. (2010) studied the airtightness of a detached house with an exterior air barrier. The total length of the joints in this exterior layer was 1280 m and the volume of this building was 1083 m³. For this case study an increase of the air permeability of 6 10^{-5} m³/m/h/Pa would correspond to an increase of the n_{50}-value of only 0.003 1/h. This is two orders of magnitude smaller than the threshold value applied in the Passive house standard (0.6 1/h).
The present article is limited to taped joints which are installed in perfect conditions. Further research to investigate the influence of the weather impact on the durability of non-perfectly taped joints is recommended. Interesting parameters to investigate in future research can be the pressure applied on the tape during installation, effects of dusty surfaces and the use of primers.

5 ACKNOWLEDGEMENTS

Research funded by a Postdoctoral Innovation Mandate (grant number 140695) of the Institute for the Promotion of Innovation through Science and Technology in Flanders (IWT-Vlaanderen).

6 REFERENCES

Ackermann, T. (2007). *Untersuchungsergebnisse zur Dauerhaftigkeit von Verbindungen und Anschlüssen bei Luftdichtheitsschichten mittels Klebemassen und Klebebändern*. In Europäisches BlowerDoor-Symposium: Dichte Gebäudehülle, Thermografie und Wohnungslüftung (pp. 109–119).

Ackermann, T. (2012). *Alternating loads – a method for testing the durability of adhesives in air tightness layers*. Proceedings of the AIVC-TightVent international workshop, 28-29 March 2012, Brussels, Belgium. pp. 61-66.

ASTM G-154 : Operating Fluorescent Light Apparatus for UV Exposure for Nonmetallic Materials.

Bracke, W., Laverge, J., Bossche, N. Van Den, & Janssens, A. (2013). *Durability and measurement uncertainty of airtightness in extremely airtight dwellings*. In AIVC Conference Proceedings.

Borsboom, W. and de Gids, W. 2012. "Seasonal variation of facade airtightness: field observations and potential impact in NZEB". Proceedings of the AIVC-TightVent international workshop, 28-29 March 2012, Brussels, Belgium. pp. 77-84.

Etherridge, D. W., & M, S. (1996). *Building Ventilation: Theory and Measurements*. John Wiley & Sons, New Jersey.

ETAG 004 (2009) guideline for European technical approval of External thermal insulation Composite systems with rendering

Gross, D. R., & Maas, A. (2011). *Qualitätssicherung klebemassenbasierter Verbindungs-technik für die Ausbildung der Luftdichtheitsschichten*. In *6th International BUILDAIR*.

Hansén, M. 2012. “Changes in airtightness after 10-20 years”. Proceedings of the AIVC-TightVent international workshop, 28-29 March 2012, Brussels, Belgium. pp. 67-75.

Hens H. (2007) *Building Physics- Heat, Air and Moisture*, Wilhelm Ernst & Sohn, Berlin. Germany, 2007

Hens H. (2006) *The vapor diffusion resistance and air permeance of masonry and roofing systems*, Building and environment 41 :6, 745-755.

Kim, A., Shaw, C., Saum, D., Shaw, J. & Colliver, D. (1986), ‘Seasonal variation in airtightness of two detached houses’, ASTM special technical publication 904, 17–32.

Langmans, J., Klein, R., De Paepe, M., Roels, S. (2010). *Potential of wind barriers to assure airtightness of wood-frame low energy constructions*. Energy and Buildings, 42 (12), 2376-2385.

Laverge J. et al. 2010. Airtightness assessment of newly build single family houses in Belgium. In: Proceedings of Buildair 2010, pp. 8.

Michaux B., Loncour X., Mees C. (2012), *Assessment of the durability of Airtightness and impact on the Conception of building details*, Proceedings of the AIVC-TightVent international workshop, 28-29 March 2012, Brussels, Belgium. pp. 85-91.

Weissmueller A. and Frédéric Delcuve F. (2013) Evaluation of the long term durability of adhesive tapes and its substrates: Requirements and testing. Webinair AIVC TIGHTVENT

ENERGY USE CONSEQUENCES OF VENTILATIVE COOLING IN A ZEB RESIDENTIAL BUILDING

Maria Justo Alonso[*1], Cathrine Kirkøen[2] and Hans Martin Mathisen[2]

1 SINTEF Building and Infrastructure
Alfred Getz v 3
7465 Trondheim, Norway
**Corresponding author: maria.justo.alonso@sintef.no*

2 Norwegian University of Science and Technology,
Kolbjørn Hejes vei 1b,
7491 Trondheim - Norway

Note: the contact addresses may be re-arranged

ABSTRACT

New buildings have to satisfy ever-tightening standards regarding energy efficiency and consumption. This results in higher insulation levels and lower air leakages that reduce heating demands. However, even at moderate outdoor temperatures these buildings are easily warmed up to such a degree that in order to ensure acceptable indoor environment quality, removal of excess heat becomes unavoidable. Use of electric energy related to mechanical cooling is considered incompatible with achieving zero energy buildings (ZEB). The use of ventilative cooling (VC) in combination with mechanical cooling means energy consumption reduction due to lower use of mechanical ventilation and cooling system.

This paper examines the application of ventilative cooling solutions in cold climates through simulations of an existing detached single family house in Norway, the ZEB Living Lab at NTNU/SINTEF. The house has computer controlled motorized windows. This will enable natural ventilation in some part of the year and could then reduce the energy use of fan power. The openable window are placed at the north and south facades and this enables considerably cross ventilation and also stack ventilation as some windows are placed four meters high.

IDA ICE program will be used to calculate the energy consumption of the baseline simulation: demand controlled ventilation with variable air volume and mechanical cooling. By means of using controlled window opening angles in IDA ICE it is possible to calculate the energy consumption while using hybrid mode ventilation.

Results show significant energy savings when using ventilative cooling. Due to the low outdoor temperatures in Norway the use of ventilative cooling remove mechanical cooling demands almost completely. The reference for comparison has been the European standard EN15251 (class II).

Ventilative cooling is proven to be relevant in combination with mechanical ventilation and will be crucial to achieving energy targets for new zero energy buildings while the indoor climate is maintained.

KEYWORDS

Ventilative cooling, energy saving, simulations IDA ICE, motorized window opening

1 INTRODUCTION

Standards for construction and refurbishment of buildings are increasingly demanding regarding energy efficiency and reduced energy consumption. Demands on the insulation level and reduced air leakages have reduced heating demands. However, this high level of insulation becomes an intrinsic heat and solar gain trap even when heating is not needed. Overheating is more often experienced and in such cases to sustain an acceptable indoor climate, removal of excess heat becomes a need. Ventilation has an important role in a building's indoor air quality (IAQ) and comfort (Sundell, 2004). People's trend of spending most of their time indoors (Lück, 2012) makes IAQ requirements become more important. Heating, ventilation and air-conditioning systems (HVAC) are responsible for 39% and 31% of primary energy end-use in residential and commercial sector respectively (Solutions, 2011). Nielsen studied cooling demands in a Danish low energy house and concluded on peaks of up to 60W/m^2, that can be reduced when having appropriate solar shading to 25W/m^2 (Nielsen, 2011). When the building does not have any cooling means, 30% of the time temperatures would be over 26 degrees (Nielsen, 2011). Melcnik (Mlecnik et al., 2012) did some post occupancy questionnaires and concluded that 34% of his respondents experienced eventual high indoor temperatures in the living room and 49% complained about bedroom over temperatures during summer. (Samuelsson, 2009) research revealed dissatisfaction in users of passive houses in Sweden due to over temperatures. In particular, more than 50% of the residents reported that it was too hot in the summer. The same conclusions are obtained by Kleiven for Norway (Kleiven, 2007).

The removal of surplus heat is often done through mechanical cooling (MC). However, energy consumption related to MC is considered incompatible with realizing zero balance. As a response, the use of Ventilative cooling solutions (VC) is settling (Venticool, 2013). VC refers to the use of ventilation air in order to reduce or eliminate the need for mechanical cooling. VC can be applied through both mechanical and natural ventilation strategies, or as a combination of the two strategies. To achieve efficient VC while ensuring an acceptable thermal climate, one should include measures that provide minimization of heat gains. (Oropeza-Perez & Østergaard (2014) studied through validated simulations of a passive house. Using natural ventilation as VC could reduce the number of hours when mechanical ventilation was needed with 90.4%. This resulted in a 37.5% reduction in energy use.

Natural ventilation is considered one of the most effective techniques for cooling whenever outdoor temperatures are lower than indoor temperatures. Also when adaptive comfort criteria can be applied to switch between mechanical and natural ventilation to reduce energy consumption while preserving satisfactory indoor climate(da Graca, Chen, Glicksman, & Norford, 1999).

VC should therefore be perceived as an integrated part of an overall system including solar shadings, minimization of internal heat gains and intelligent use of thermal mass (Venticool, 2013) should be combined with night set back.

This work examines the application of ventilative cooling in cold climates through simulations of an existing residential building (Living lab) in Norway. This building has the ability to be ventilated through mechanical ventilation and hybrid mix mode ventilation with motor controlled windows.

The overall scope of this paper is to evaluate the performance of the Living Lab ventilation solution concerning thermal comfort and energy consumption. Further, it is to compare the results against conventional all-mechanical (no cooling) ventilation systems to evaluate the most energy-efficient solution without compromising the IAQ and comfort.

2 LIVING LAB

The Living Lab is representative of the Norwegian residential building stock regarding typology (detached, single family house) and surface, while integrating state-of-the-art technologies for energy conservation and solar energy exploitation.

The test facility is a single family house with a gross volume of approximately 500 m^3 and a heated surface (floor area) of approximately 100 m^2. The data acquisition system is primarily designed to be able to measure energy demand for heating, ventilation, lighting and appliances, as well as renewable energy harvesting by means of a roof-integrated PV system and of façade-integrated solar thermal panels. Accumulation tanks and indoor environment are also fully monitored.

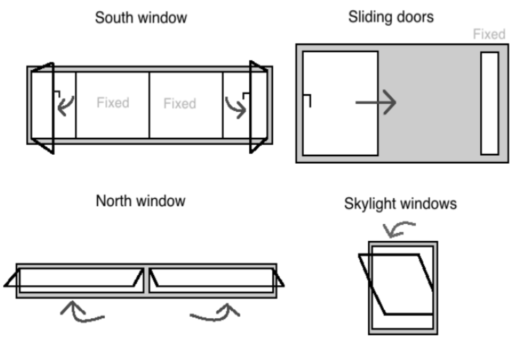
Table 1 - Thermo-physical properties of building envelope components

Thermo physical properties	
U-value wall	0.11 W/m^2K
U-value floor	0.10 W/m^2K
U-value roof	0.10 W/m^2K
U-value windows (south façade)	0.65 / 0.69 (when ventilated) W/m^2K
U-value windows (north façade)	0.97 W/m^2K
U-value windows (east-west façade)	0.80 W/m^2K
U-value skylight	1.0 W/m2K
g-value for windows	0.5-
Air tightness, n_{50}	0.5 ach

The ventilation is designed as a mixed-mode hybrid system with mechanical balanced ventilation. Supply air terminals are located in the living room and in the bedrooms; extract in the bathroom and kitchen. A heat wheel unit with efficiency of 85% at nominal value and a hydronic heating coil capable of warming up the inlet air up to 40^{o}C are installed (Finocchiaro, Goia, Grynning, & Gustavsen, 2014). The dwelling has operable windows on every facade in order to profit from both stack and cross flow ventilation through mechanical opening of windows.

On the North side an elongated window is implemented. It is constructed with hinges at the top, and opens towards the interior to a maximum angle of 39°. On the West and East side there are glass sliding doors. Two sets of rooftop skylight tripled glazed windows facing north have been implemented. They open horizontally to a maximum angle of 30°. The South windows open a maximum of 37°.See Figure 1.

Figure 1:(a)Placing of windows in the Living Lab (b) Window opening in the Living Lab



3 SIMULATIONS

3.1 Window control

For calculations, the simulation program IDA ICE was used. The calculation period was the warmest week in Trondheim (Norway) in IDA ICE's climate TMY year. Preliminary simulations to determine the warmest and coldest room have been done so that these rooms are used for comparison. Six different strategies for control of thermal comfort were compared; afterwards the energy consumption was analyzed. The opening of windows combines with mechanical ventilation. In case of too low outdoors temperatures mechanical ventilation ensures hygienic ventilation rates. To analyze thermal comfort, the number of hours with overheating and over cooling have been studied For determination of these temperatures, optimization was based on natural ventilation.
Table 2 shows an overview of the performed simulations to define how to control windows and ventilation system to maximize the effect of ventilative cooling.

Table 2: Overview of simulations performed to determine how to apply ventilative cooling

Study	Tested solutions	Outcome
How to apply window control: Set point temperatures for opening	Various set-point temperatures for the window control systems	Set-points On/off control, Daytime exclusively Set-points On/off control, Night-time exclusively Set-points On/off control, All hours Set-points PI-Control, Daytime exclusively Set-points PI-Control, Night-time exclusively Set-points PI-Control, All hours
How to apply window control: Active window period	On/off control, Daytime only On/off control, All hours PI control Daytime only PI control, All hours	Best of the On/off control solutions Best of the PI control solutions Most influencing factors for window opening
How to apply ventilative cooling	Natural ventilation On/off control Concurrent On/off control Change-over On/off control Natural ventilation PI-control Concurrent PI-control Change-over PI-control	Best ventilative cooling solution

The first challenge to solve is the determination of the set point temperatures to control the window opening. These are chosen based on the type of control. For a more reactive control, three temperatures are defined for the PI control. For on-off control only two temperatures are selected. For both types of control, the goal is to minimize the number of hours with overheating and with overcooling. Table 3 and Table 4 show the allowed hours of opening and the windows working for providing ventilative cooling.

Table 3: On Off control

Type	Hours applied	Windows utilized	Control signal design
VC Day	07.00-23.00	South + Skylight	Tzone < T1 -> closed Tzone > T1 -> open until T2 is reached
VC Night	23.00-07.00	South + Skylight Kitchen only	Tzone < T3 -> closed Tzone > T3 -> open until T4 is reached
VC All hours	All day long	South + Skylight (-23-07 Loft)	Tzone < T5 -> closed Tzone > T5 -> open until T6 is reached

The windows were frequently opened on sunny days regardless of the occupancy, outdoor temperature and presence of wind. For cloudy days, windows were frequently used for all set-point solutions when outdoor temperatures were high and the building was occupied. For cloudy days and low outdoor temperatures, windows were rarely open regardless of the occupancy schedule and wind.

Table 4: PI control

Type	Hours applied	Windows utilized	Control signal design
VC Day	Weekday:07.00-23.00 Weekend:09.00-24.00	South + Skylight + North (-08-17 weekdays)	Tout < 15 ˚C -> Hold T7 15 ˚C < Tout < 20 ˚C -> Hold T8 20 ˚C < Tout -> Hold T9
VC night	Weekday:23.00-07.00 Weekend:24.00-09.00	South + Skylight Kitchen only	Tout < 15 ˚C -> Hold T10 15 ˚C < Tout < 20 ˚C -> Hold T11 20 ˚C < Tout -> Hold T12
VC All hours	All day long	South + Skylight(-23-07 weekday -24-09 weekend Loft) + North(-08-17 weekdays)	Tout < 15 ˚C -> Hold T13 15 ˚C < Tout < 20 ˚C -> Hold T14 20 ˚C < Tout -> Hold T15

The analysis of the optimal window control temperatures was based on results regarding overheating and overcooling. For overheating the number of hours over 24, 26 and 28 °C in the living room were used, and the maximum temperatures in the attic. For overcooling, the number of hours with temperatures below 20 °C in living room and kitchen and their air velocities were considered as these show overcooling and draft. Based on these, Table 6 shows the control temperatures and the general performance of the control.

Table 5: Overview of the thermal comfort achieved for each tested window control solution

Control	Temperatures and corresponding thermal comfort		
On/off daytime exclusively	(25-20)°C, (24-20)°C or (23-20)°C Poor	25-22°C Good	24-22 °C Best
On/off night-time exclusively	(22-20)°C or (20-19)°C Poor		
On/off all hours	(25-20)°C, (24-20)°C or (23-20)°C Poor	25-22 °C Good	24-22 °C Best
PI exclusively daytime	(22-24-26)°C, (21-23-26) °C or (20-22-26)°C Poor	23-23-26°C Good	23-24-26°C Best
PI exclusively night time	(20-20-21°C) or (19-19-20°C) Poor		
PI all hours	(22-24-26) °C,(21-23-26) °C or (20-22-26) °C Poor	23-23-26°C Good	23-24-26°C Best

According to the analysis of the results, the most influencing factors on the need for ventilative cooling at the Living Lab were: solar radiation, outdoor temperature and occupancy. The wind does not appear to have influence on the need for cooling but influences the efficiency of ventilative cooling.

3.2 Interaction between ventilative cooling and mechanical ventilation

The use of concurrent mechanical ventilation and window opening was compared to its use in change-over (meaning either windows opening or mechanical ventilation). For the concurrent systems, the best on/off and PI window control systems were combined with mechanical ventilation so that constant hygienic mechanical ventilation flow rate was provided.
For the change-over systems, hygienic ventilation was provided either by the best performing on/off and PI window control systems or mechanical ventilation. Hence, both systems were decoupled when ventilative cooling was utilized. An additional control system was constructed to turn off mechanical ventilation when windows were open. The central AHU unit in IDA ICE is initially not able to retrieve signals indicating window opening position. Therefore, an on/off mechanical ventilation system had to be implemented for each zone. It was designed so that if one or more windows in the current zone were open, the mechanical ventilation in that particular zone would be turned off. A ventilation system with this design does no longer ensure a balanced mechanical ventilation system.

Ventilative cooling and on-off control

The results show that both solutions resulted in good thermal comfort, the results regarding overheating and overcooling were similar.

Table 6: Results from annual simulations using the on/off window control system.

Energy	Concurrent Mechanical and On/off window control	Change-over Mechanical and On/off window control
Recovery [kWh]	4798	4769
Zone heating [kWh]	5470	5489
AHU heating [kWh]	242	245
Fan energy [kWh]	1095	1052
Lighting [kWh]	1083	1083
Equipment [kWh]	1116	1116
Total energy demand [kWh]	9004	8984

The hours of overheating were the same for the two systems. The change-over system resulted, however, in slightly more hours of overcooling compared to the concurrent zoned system. The change-over system was designed to only turn off the mechanical ventilation units in the rooms that had open windows. This creates an unbalance in the mechanical ventilation and creates a more unpredictable and unstable system (also resulting in a small reduction on the efficiency of the heat wheel).

When comparing energy demand for both solutions, the differences in energy demand of both systems were not significant. The change-over system resulted in only slightly less energy demands than the concurrent system.

Mechanical ventilation was always in operation for the concurrent system. This should have resulted in a bigger increase in energy use for fans compared to the change-over system.

However, since the change-over system was designed as a zoned system, it only turned off the mechanical units in some of the rooms 10.8% of the time. The difference in energy use for fan operation was therefore not substantial. The pure natural ventilation (in change-over mode) was only utilized 13.2% of the time. For the remaining parts of the year there would be no air exchange except from infiltration though this is expected to be low in such a highly insulated building like Living Lab. The concurrent solution is expected to have the best indoor air quality because it operates mechanical ventilation constantly. It is also considered to be less complex than the change-over system. This is because the window control system and the mechanical ventilation system operate independently. Also, the concurrent system does not jeopardize the balancing of the mechanical ventilation system. Since all solutions provided good thermal comfort and the differences in energy demand for the mixed-mode systems were not significant, the concurrent system solution was deemed the best ventilative cooling solution utilizing on/off window control.

Ventilative cooling using PI window control

The results from the whole year simulations of both ventilative cooling solutions utilizing PI window control are presented in Table 7.

The results regarding overheating were similar to those obtained when using on/off window control. There were very few hours of overheating. However, different results regarding overcooled hours were obtained. The results show that the risk of overcooling was significantly increased when utilizing the PI window control system. Since the window control system had a simpler design, applying mechanical ventilation system improved the situation. Use of mechanical ventilation reduced the need for natural ventilation, hence reducing the amount of overcooled hours.

Table 7: Results from whole year simulations using the PI window control system.

Energy	Concurrent Mechanical and PI window control	Change-over Mechanical and PI window control
Recovery [kWh]	4808.50	4137.4
Zone heating [kWh]	5470.4	6038.3
AHU heating [kWh]	243.0	177.9
Fan energy [kWh]	1094.9	943.7
Lighting [kWh]	1083.0	1083.0
Equipment [kWh]	1116.0	1116.0
Total energy demand [kWh]	9007.0	9358.9

The difference in energy demand for both systems was not significant in this case either. However, opposed to the on/off solutions, the change-over system obtained slightly higher energy demand than the concurrent system. In this case, the increase in energy use for the zone heating was bigger than the decrease in energy use for fans. The concurrent system was also deemed the best solution for ventilative cooling utilizing PI window control. Both solutions provided approximately equally thermal comfort and the differences in energy demand between the mixed-mode systems were not significant.

As explained for the on/off solutions, the concurrent system is expected to provide better indoor air quality. It is also less complex than the change-over system.

3 CONCLUSIONS

The results from the simulations implied that there will be a severe risk of overheating in Living Lab if no active or passive cooling techniques are applied. The results showed nonetheless that ventilative cooling can prevent overheating without significantly increasing the energy demand. Due to the uncertainties related to increased air velocities, it was not possible to eliminate the risk of overcooling caused by ventilative cooling completely.

However, the study showed that the amount of hours with overcooling could be held at an acceptable level. The simulations revealed that nighttime ventilative cooling had no positive effect on the thermal environment in Living Lab. This building has low thermal mass and the control was built so that windows were not open during the day when most internal gains happen.

The study found that the best way to apply ventilative cooling in Living Lab would be to implement a concurrent mixed-mode system where the window control system is only active during the day. It should be designed to open the south and skylight windows to maximum opening when indoor air temperatures exceed 24°C and close them when indoor air temperatures drops below 22°C. The results revealed that this system would reduce the number of overheated hours recorded when not utilizing ventilative cooling with 99%. The number of overcooled hours would be kept at a moderate level, 48 hours/year. Utilizing this ventilative cooling system resulted in increased energy demand of 52 kWh/year compared to use of only hygienic mechanical ventilation.

In Trondheim (and by extension in cold climates) overheating in low-energy dwellings can be prevented with ventilative cooling. Ventilative cooling can have a significant positive effect on the thermal environment without having a significant negative effect on the use of energy. In some cases, energy consumption can even be reduced when applying ventilative cooling. Overcooling can be an issue when utilizing ventilative cooling in these cold climates. A careful design process is required for ventilative cooling to have the desired effect. The process should be individual for each building and climate. A more complex natural ventilation system requires a more accurate and careful design process. Also, the acceptable indoor temperatures for mechanically ventilated buildings often have to be adjusted for buildings intended to utilize ventilative cooling. An automatic window control system is often necessary for ventilative cooling to achieve the desired thermal environment.

4 ACKNOWLEDGEMENTS

This work has been supported by the Research Council of Norway and several partners through the SINTEF/NTNU "The research centre of Zero Emissions Buildings" (ZEB). Grant number 193830. And the industrial project Ventilative Cooling with ENOVA.

5 REFERENCES

da Graca, G. C., Chen, Q., Glicksman, L. R., & Norford, L. K. (1999). Simulation of wind driven ventilative cooling in an apartment building in Beijing and Shanghai. Ishvac 99: 3rd International Symposium on HVAC, Vols 1 and 2, 648-658.

Finocchiaro, L., Goia, F., Grynning, S., & Gustavsen, A. (2014). The zeb living lab: a multi-purpose experimental facility. Paper presented at the IEA EBC Annex 58 Expert Meeting, Ghent.

Kleiven, T. (2007). Brukerunderskelse i husby amfi. In S. Byggforsk. (Ed.).

Lück, K. (2012). Energy efficient building services for tempering performance-oriented interior spaces – A literature review. Journal of Cleaner Production, 1-10.

Mlecnik, E., T.Scutze, Jansen, S., de Vries, G., Visscher, H., & van Hal, A. (2012). End-user experiences in nearly zero-energy houses. Energy and Buildings.

Nielsen, L. S. (2011). Heating and cooling challenge in nZEB Experiences from built houses and detailed building simulations calls for a careful and integrated approach when designing heating and cooling systems for nearly zero energy buildings. REHVA Journal.

Oropeza-Perez, I., & Østergaard, P. A. (2014). Potential of natural ventilaton in temperate countries - a case study of Denmark. Energy and Buildings.

Pérez-Lombard, L., Ortiz, J., & Pout, C. (2008). A review on buildings energy consumption information. Energy and Buildings, 40 394-398.

Samuelsson, M. a. L., T. . (2009). Passive houses from a users perspective.
Solutions, C. f. c. a. E. (2011). Building Envelope.

Sundell, J. (2004). On the history of indoor air quality and health. Indoor Air, 14, 51 – 58.
Venticool. (2013). The international platform for ventilative cooling.

VENTILATIVE COOLING STRATEGIES TO REDUCE COOLING AND VENTILATION NEEDS IN SHOPPING CENTRES

Marta Avantaggiato*[1,2], Annamaria Belleri[1], Michele De Carli[2], Roberto Lollini[1]

1 Institute of Renewable Energy
EURAC Research,
Bolzano

2 DII-Department of Industrial Engineering
University of Padua
Padua

**marta.avantaggiato@eurac.edu*

ABSTRACT

Because of the customer need of best possible comfort condition and satisfaction, shopping centers are conditioned by means of basic HVAC systems, often without considering the potential of natural ventilation to contribute to air change rate, and to reduce the cooling demand. Mechanical ventilation systems are also preferred to natural ventilation because more controllable and reliable since they are not affected by the uncertainty of natural forces.
However, atriums or in general common areas within a shopping center can suite the ventilative cooling purpose. In fact, on one hand they present more relaxed ranges of interior conditions respect to retail zones and, on the other hand, they can exploit the airflow driven both by thermal buoyancy and by wind pressure because of the big volumes involved. The use of ventilative cooling strategies can be beneficial in reducing both cooling demand and electricity consumption for ventilation.
The paper investigates the feasibility of ventilative cooling strategies as retrofit opportunities in shopping centers. Ten shopping centers, representing the retail stock among Europe have been analyzed under two aspects. First, we evaluated, according to the climatic cooling potential, the feasibility of natural or hybrid ventilative cooling strategies depending on internal gains; secondly, we identified the most suitable ventilative cooling strategies according to the architectural features of each shopping center. Then combining the results of the two analysis, we identified the most suitable ventilative cooling strategies according to different internal load. As last step, we proved the beneficial effect of the ventilative cooling strategies proposed in one of the reference shopping center. By means of dynamic simulation, we verified their effectiveness in terms of cooling need and ventilation consumption reduction. The results shows a cooling demand reduction between 41% and 49% depending on the lighting gains. The electricity consumption for ventilation is as well, decreased or nullified depending on the ventilative cooling strategies, linked with the lighting internal gains values.

KEYWORDS

Ventilative cooling feasibility, Shopping center, Cooling energy savings, Climate potential,

1 INTRODUCTION

In shopping centers, differently from residential buildings where energy is mainly consumed by heating, cooling, hot water, cooking and appliance (BPIE, 2011), energy is principally used for store lighting and ventilation followed by heating and air conditioning, food

refrigeration and other (Schönberger, 2013).This energy consumption distribution depends, however, on the shopping center typology and on the climate location.
Based on these data, energy retrofit solutions should consider actions able to reduce both specific lighting power installed and the energy used for the air-conditioning and ventilation system (HVAC). Lighting power installed always affects the HVAC consumption. In fact, while high internal gains during winter have a beneficial effect in decreasing the heating demand, in summer they cause an increase of the cooling demand. Generally, shopping centers have common areas managed by a unique referent (e.g. owner, energy manager) which is also the one who makes the decisions during a retrofit process. On the other side, in the "leasing" area, each shop is managed by franchising companies, which rule and direct the shop according to standardized protocols and refer to a distinctive interior and lighting design, restraining the applicability of general retrofit solutions (e.g. installation of a defined lighting technology, centralized HVAC controls). Accordingly, it is easier at a first retrofit step to consider the common area where there is a higher degree of freedom in the solution applicability. Generally, shopping centers are conditioned by means of HVAC systems, without considering the potential of natural ventilation to guarantee the minimum air change rate required by IAQ standards and to reduce cooling demand. Mechanical ventilation systems are also preferred to natural ventilation because more controllable and reliable, since they are not affected by the uncertainty of natural forces. Thereby, within the design process the team never focused neither on opening sizing nor on control strategies definition for ventilative cooling systems (natural or hybrid). So far, shopping mall design has included a small proportion of automated windows, sized for smoke ventilation only. Depending on the external climate conditions, acceptable levels of thermal comfort and IAQ can be reached without or with partial use of the mechanical systems, leading also to operational and maintenance cost savings.
However, atriums or in general common areas within a shopping center can suite the ventilative cooling purpose. In fact, on one hand they present more relaxed ranges of interior conditions respect to selling area and, on the other hand, they can exploit the airflow driven both by thermal buoyancy and by Venturi effect because of the big volumes involved.
Thermal function and role of the atria depend on the ventilation strategy (Moosavi L., 2014): either they can supply outside fresh air or they can drive stagnant air outside, or a combination of both. Atria can be used to enhance stack ventilation taking air in and exhaust air out from vents at the top of the atrium.
The paper investigates the retrofit opportunities to exploit ventilative cooling in shopping centers' common areas (shop galleries and atria) in terms of external climate conditions and architectural features. In particular, the paper analyses the ventilative cooling strategies in ten reference buildings identified within the CommONEnergy EU FP7 project (Bointer,R, 2014) to represent the retail stock among Europe.
Table 1 collects the reference buildings showing the typology they belong to (Bointer,R, 2014), and the climate classification according to the IEA Task 40 definition (Cory S., 2011).

Table 1 List of reference shopping centers

ID	Name of the shopping center	Shopping Center typology	Location	Country	Climate
CS	*City Syd*	Medium Shopping center	Trondheim	NO	HD
ME	*Mercado del Val*	Specialized and Others	Valladolid	ES	H&CD
GE	*Genova Ex- Officine Guglielmetti*	Specialized and Others	Genoa	IT	CD
KA	*Centro Commerciale Katané*	Medium Shopping center/ Hypermarket	Catania	IT	CD
DO	*Donau Zentrum*	Very Large Shopping center	Wien	AT	H&CD
BC	*Brent Cross*	Very Large Shopping center	London	UK	H&CD
ST	*Studlendas*	Small Shopping center	Klaipeda	LT	HD
GB	*Grand Bazar*	Small Shopping center	Sint-Niklaas	BE	H&CD
WA	*Waasland Shopping Center*	Large Shopping center	Antwerp	BE	H&CD
PA	*Pamarys*	Small Shopping center/ Hypermarket	Silute	LT	HD

2 METHODOLOGY

The following paragraphs describe the methodology used to identify the ventilative cooling strategies that better suits each of the ten reference buildings.
The analysis consists of three steps:

1. Ventilative cooling potential analysis used for the identification of the climatic potential in terms of ventilative cooling suitability for a reference gallery/atrium thermal zone;
2. Architectural features analysis with the aim to identify which ventilative solution better suite with the internal layout and space distribution;
3. Ventilative cooling strategies identification; it combines the two previous analysis with the aim of defining the most convenient one for each reference buildings.

2.1 Ventilative cooling potential analysis

Firstly, we evaluated the ventilative cooling potential of each climate location by using the ventilative cooling potential tool (Belleri A., 2015) that is under development by the IEA EBC Annex 62 research project (IEA EBC Annex 62 - Ventilative cooling, 2014-2017).

The tool requires basic information about very simplified building model, its use and the climate. In particular, the method assumes that the heating balance point temperature (T_{0-hbp}) establishes the outdoor air temperature below which heating must be provided to keep the desired indoor temperature at the heating set point (T_{i-hsp}).Therefore, when outdoor dry bulb temperature (T_{o-db}) exceeds the heating balance point temperature, direct ventilation is considered useful to keep indoor conditions within the comfort zone. The comfort zone is determined according to the adaptive thermal comfort model proposed in the EN 15251:2007 standard. When the outdoor dry bulb temperature falls below the heating balance point temperature, ventilative cooling is no longer needed.

The analysis with the tool is based on a single-zone thermal model. The user should provide to the tool the hourly climatic data. Within this analysis, in order to provide for a proper comparison of the ventilative cooling potential of the ten reference buildings, we referred to the same reference zone that is assumed to be representative for a typical atrium or a gallery. Table 2 reports the geometrical specifications.

Table 2 Reference zone geometrical characteristics

Height m	**Length m**	**Width m**	**Floor Area m^2**	**Envelope Area m^2**	**Fenestration Area m^2**	**Opening hour**	**Heating set-point temperature**
5.1	50	10	500	1112	111	9:00-21:00	16°C

We assumed the opening time of the center as 09:00-21:00 (11 hours per day) and set the heating set point temperature at 16°C as recommended by the standard EN 15251: 2007 for building Category II (new buildings and renovation).
The weather files referred to the reference buildings' locations derive from the historical data series (2000-2009) of the Meteonorm database (Meteonorm, 2009).
Approaching a shopping center retrofit, the most effective and easy-fit retrofit solution is the installation of energy efficient lighting system (Haase M., 2015).Taking this into consideration we decide to assess a parametric analysis by varying the internal lighting power density input and then evaluating the ventilative cooling potential. By doing so we can evaluate possible synergies and restrictions in the application of retrofit solution that consider simultaneously lighting, ventilation and cooling retrofit. The two considered levels are:

- lighting power density of 50 W/m², representing the state-of-the art of lighting installation according to a local retail building design firm (Howlett G., 2011);
- lighting power density of 10 W/m², representing the expected level after retrofit (Haase M., 2015).

General electric equipment power density (10 W/m^2) and internal loads due to occupancy (7 W/m^2) are considered the same in both cases.
A threshold value of 4.5±2.5 ACH as maximum airflows is fixed to prevent too high inlet air velocities which might cause discomfort situations. This value comes from a simple calculation; assuming that 25% of the glazed surface is openable for natural ventilation purposes and keeping air velocity of inlet airflow to be lower than 0.2 m/s as prescribed by standard, we derive the threshold airflows value.
An algorithm integrated in the tool splits the total number of hours when the building is occupied into the following groups:

8. **Ventilative Cooling mode [0]**: outdoor temperature is below the heating balance point temperature: no ventilative cooling can be used since heating is needed;
9. **Ventilative Cooling mode [1]**: outdoor temperature exceeds the balance point temperature, yet falls below the lower temperature limit of the (cooling) comfort zone: direct ventilative cooling with minimum required airflow rate for indoor air quality;
10. **Ventilative Cooling mode [2]**: outdoor temperature is within the range of comfort zone temperatures: direct ventilative cooling with increased airflow rate;
11. **Ventilative Cooling mode [3]**: outdoor temperature exceeds the upper temperature limit of the comfort zone: direct ventilative cooling is not useful.

The tool also predicts the hourly airflow rates required to maintain comfort condition when ventilative cooling mode [1] and [2] occur and the number of hours when night ventilative cooling can be activated.
Based on the ventilative cooling potential analysis we defined the feasibility of four possible ventilative cooling strategies:

- Daytime natural ventilative cooling consists in circulating outdoor air through windows and openings during opening hours. It is assumed to be effective when the average value of the air flows during ventilative cooling mode [1] and [2] are lower than the threshold value of 4.5±2.5 ACH;

- Daytime hybrid ventilative cooling consists in circulating outdoor air through windows and opening until the average value of the air flows during ventilative mode [1] and [2] is lower than the threshold value of 4.5±2.5 ACH. When the airflows exceed this value the ventilation is switched to fully mechanical (openings/vents are closed) providing the minimum airflow rates and the mechanical cooling system maintain the temperature into comfort ranges
- Daytime and night-time natural ventilative cooling when the condition for the daytime natural ventilative cooling are satisfied and also the percentage of hours when night ventilative cooling is activated is higher than 5%. Night ventilative cooling consists in circulation of outdoor air through windows and openings during night-time;
- Daytime and night-time hybrid ventilative cooling when the condition for the hybrid ventilative cooling are satisfied and also the night ventilative cooling activation percentage of hours is higher than 5%. Also in this case, night ventilative cooling is meant as driven by natural forces.

Figure 1 shows the decision flow chart with two-option questions to assess the most effective ventilative cooling strategy according to the climate potential. The first question regards the level of internal lighting gains, which determined whether natural ventilation is enough to offset internal gains. High lighting gain correspond to 50 W/m^2 while low refers to 10 W/m^2. The second is about the number of air change rates in ventilative mode [1] and [2], which determines whether cold draught risk might occur using natural ventilation. The last question is about number of hours when night-time ventilative cooling is activated, which determines night cooling feasibility. The flow chart allows defining the best ventilative cooling strategy for each reference shopping center according to the climatic potential.

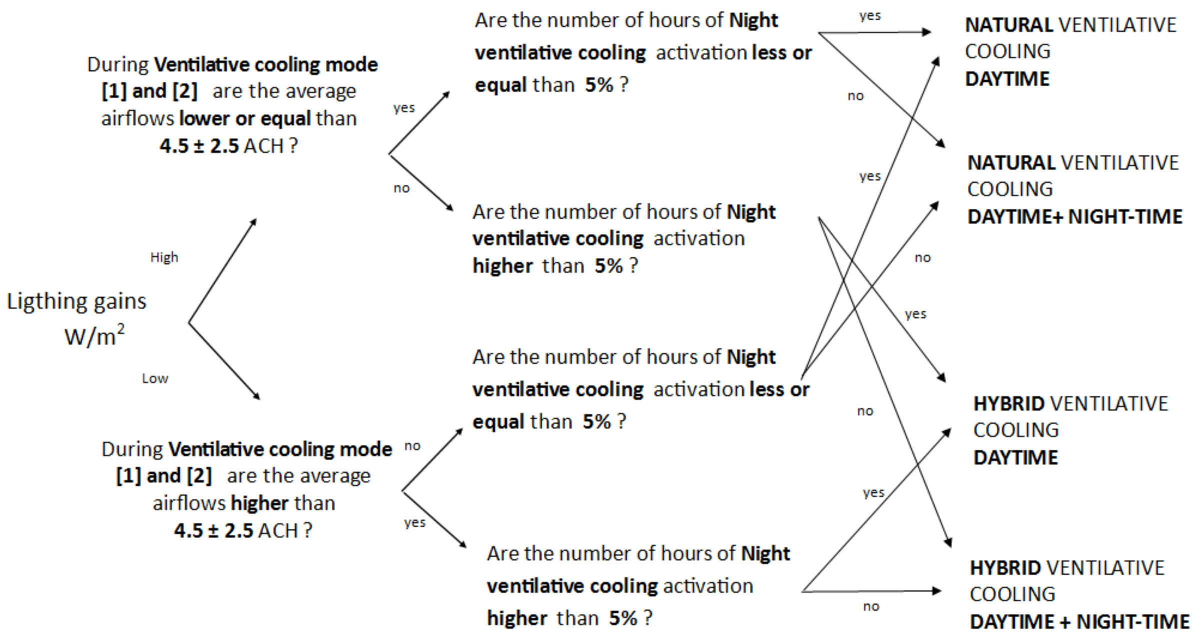


Figure 1 Ventilative cooling strategy decision chart according to climatic potential.

2.2 Architectural features analysis

A ventilative cooling strategy involves the whole building envelope as vents and openings can be located both on façade and roof to exploit buoyancy due to temperature difference between shops and central spaces and along the atrium height. Therefore, it is strictly dependent on building design and indoor spaces layout.

We analyzed the internal layout of each reference building taking into account the following features:

- Interconnected galleries and atria;
- Building shape, number of levels and ceiling height;
- Location of parking areas.

Considering shopping mall features, we identify feasible ventilative cooling systems among the followings:

- *Wind-induced ventilation* uses the wind pressure to force air through the building. Examples of technologies that use this effect are wind scoops, wind catchers and Venturi ventilators;
- *Solar assisted ventilation* exploits the effect of solar radiation to enhance hydrostatic buoyancy of air being warmer than its surroundings. Example of technologies that use this effect are solar chimney, double skin façades and ventilated walls;
- *Wind-buoyancy ventilation* is where air is driven through the building by vertical pressure differences developed by thermal buoyancy;
- *Wind-driven ventilation* exploits the pressure distribution generated by the wind on the building envelope to drive air through the building (i.e. vents and openings located on opposite sides of the building);
- *Fan assisted ventilation* relies on mechanical systems (i.e. fan) to enhance wind and/or buoyancy effects;
- *Mechanically-driven ventilation* uses a system of fans and ducts to circulate air within the building.

Figure 2 shows the ventilative cooling strategy decision chart according to the architectural features. The decision chart is based on three main questions. The first question is about the presence of galleries and atria, which determines the possibility to use ventilative cooling strategies in common areas. The second question investigates the possibility of taking air at roof level depending on its quality and the last one asks for the building shape and in particular the number of levels, which determines the possibility of exploiting air temperature and pressure at different heights (buoyancy effect). Following this chart, we were able to define the best ventilative cooling strategy for the ten reference shopping centers according to the architectural features.
In general, dealing with ventilative cooling as energy retrofit solution, other important design constraints should be considered, such as building regulations and standards (indoor air quality, fire safety), local outdoor environment (air pollution, noise), HVAC system configuration and management and tenants' needs (i.e. food stores with fresh filtered air).
However, the preliminary analyses here reported present at a lower level of detail due to the lack of available information about the design constraints related building energy management and local regulations.

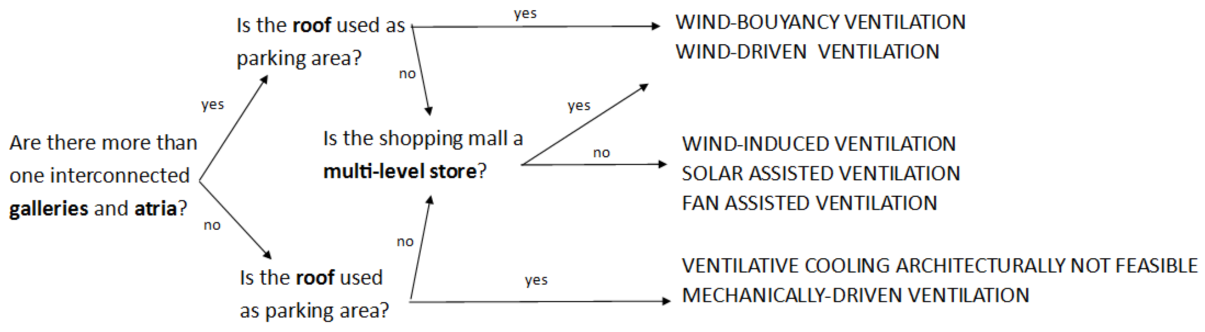


Figure 2 Ventilative cooling strategy decision chart according to architecture

3 VENTILATIVE COOLING FEASIBILITY

The following paragraphs show the results of both the ventilative cooling potential analysis and the architectural features for the ten reference buildings.

3.1 Ventilative cooling potential

The graph in Figure 3 shows the percentage of hours within a year when direct ventilative cooling can potentially assure thermal comfort during opening hours. In general, considering a high level of lighting power density, direct ventilative cooling with minimum and increased airflows is required for a 25% extra time compared to the case with low level of lighting power density. The values of airflows needed to offset the internal gains are also higher than those ones required with low lighting power as shown in Table 3. According to the analysis, when lighting power density is set to 50 W/m², ventilative cooling can be potentially exploited over 80% of the time in Catania and London while for almost all the time in Genoa.

When direct ventilative cooling is not useful during the day (VC mode [3]) because the outdoor temperatures are above the upper temperature limit of the comfort zone, the night-time ventilation potential during the following night is investigated. Table 3 shows for each reference building the average airflows during ventilative cooling mode [1] and [2] and the percentage of hour of night cooling activation. Both values are then used to determine the suitable ventilative cooling strategies according to the decision flow chart in Figure 1.
Night-time ventilation is activated for more than 10% only in Catania, due to the hot climate.

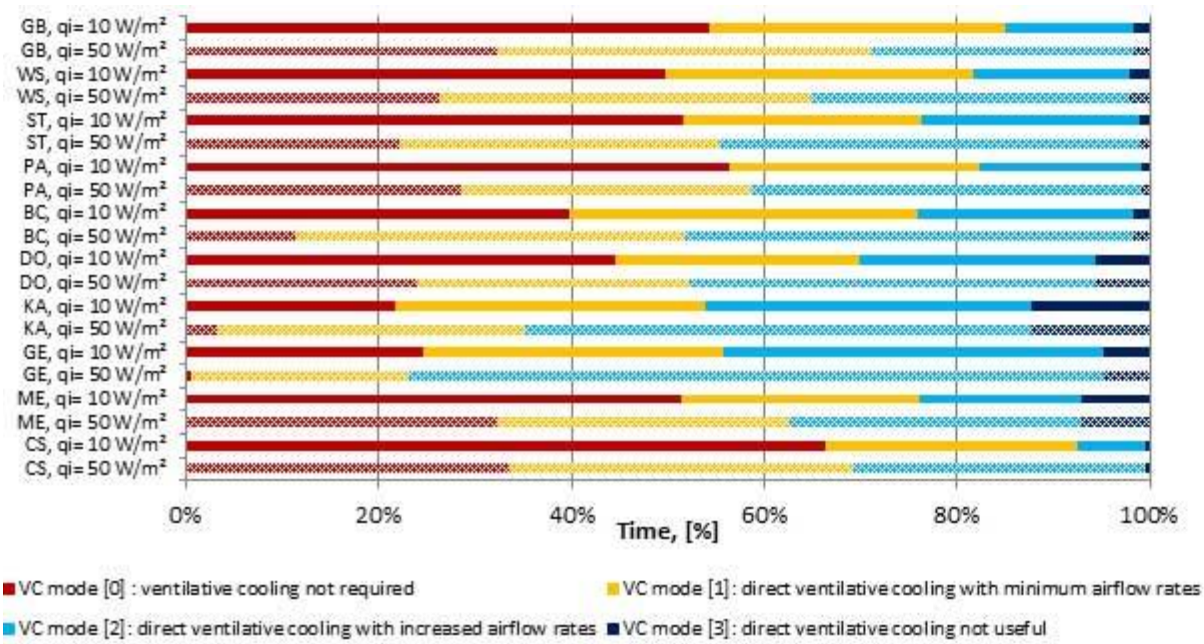


Figure 3 Percentage of hours within a year when direct ventilative cooling is required, useful or not useful in the ten reference building climates considering different values of internal gains.

Table 3 Average airflows during ventilative cooling mode [1] and [2] and % of hours of night ventilative cooling activation

		Reference buildings									
		CS	**ME**	**GE**	**KA**	**DO**	**BC**	**ST**	**GB**	**WA**	**PA**
Average airflow during ventilative cooling mode [1] and [2] [ACH]	q_i= 50 W/m^2	4.2 ± 1.9	6.9 ± 2.9	6.0 ± 3.3	6.8 ± 3.4	6.3 ± 3.2	5.6 ± 2.9	3.8 ± 2.1	6.6 ± 2.9	6.4 ± 2.8	3.7 ± 2
	q_i= 10 W/m^2	2.2 ± 0.9	3.6 ± 1.2	3.3 ± 1.3	3.9 ± 1.6	3.1 ± 1.3	2.2 ± 1	1.7 ± 0.9	3 ± 1	2.8 ± 0.9	2.1 ± 1
% of hours when ventilative cooling is activate during night	q_i= 50 W/m^2	1%	8%	6%	15%	7%	3%	2%	3%	3%	1%
	q_i= 10 W/m^2	0%	6%	6%	14%	7%	2%	2%	2%	1%	1%

3.2 Architectural analysis

The internal layout analysis showed that almost all the shopping centers have interconnected galleries and atria, and with our considerations on the architectural aspects, we found that in all the reference shopping centers there are the conditions to exploit ventilative cooling.
A common strategy is the exploitation of wind-buoyancy ventilation, natural or hybrid, since the shopping centers have usually two or more levels (Table 4).
Pamarys shopping centers is the only single-level store thus a ventilative cooling strategy exploiting air movement by means of buoyancy effect cannot suite for the purpose. In this case, indeed exploitation of wind-induced ventilation by means of devices such as wind scoops and wind catcher or fan-assisted ventilator seems the only feasible solution.
Catania and Genoa have roof portions used as parking area. In that case, because of the multilevel geometry the buoyancy effect can still be applied using openings on the lateral façade as airflow guide for the exhausted air.

3.3 Ventilative cooling strategies

Figure 4 merges the outcomes of the ventilative cooling potential and the ventilative cooling strategies. From the climatic potential point of view, we observed that:

- In heating dominated (HD) climates (Trondheim, Silute and Klaipeda) natural ventilative cooling is enough to offset the internal gains, both low and high level;
- In cooling dominated (CD) and heating&cooling dominated (H&CD) climates, with high lighting gains hybrid ventilative cooling is the selected solution, while with low internal lighting gain natural ventilative cooling is sufficient to guarantee an acceptable comfort level.

Night-time ventilative cooling is useful for the climates of Valladolid, Genoa, Catania and Wien. Because of the very common multiple level layout among the reference buildings that helps in creating temperature stratification a ventilation approach that relies on buoyancy effect seems to be the best solution.
The results suggest that, when facing a retrofit, the ventilative strategies chosen have to be in line with the level of lighting power density retrofitted in order to avoid an over-design of the airflows. Overlooking this interdependency could lead to discomfort condition due to the high

air volume inside the building and to extra investment costs for windows´ and openings´ actuators, which might not be necessary.

4. POTENTIAL ENERGY SAVINGS

Among the ten reference buildings, we analysed the Donau Zentrum to verify the effectiveness of the ventilative cooling solutions proposed in terms of cooling need reduction. According to the climatic potential outcomes we tested a hybrid ventilative cooling strategy when internal lighting power density is equal to 50 W/m^2 while a natural one when lighting power density decrease down to the value of 10 W/m^2.

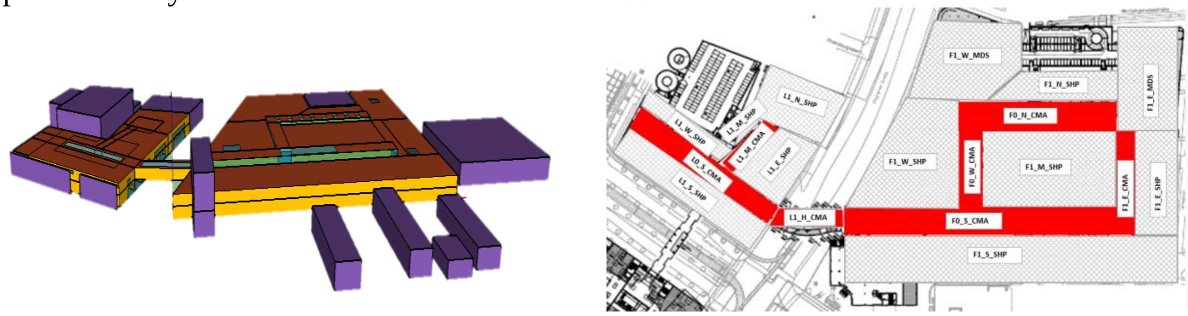


Figure 4 Donau Zentrum sketchup model south view (left) and galleries and atria highlighted in red (right), common area of the mall

We modelled the building within the Trnsys environment applying the ventilative cooling strategies to the galleries and atria highlighted in Figure 4**).**

The baseline is ventilated by a fully mechanical system that supply minimum airflows of 2 ach during opening hour with heat recovery (60% efficiency). The effectiveness of ventilative cooling is tested by setting the airflow rates equal to ones calculated by the ventilative cooling potential tool. When the conditions for ventilative cooling mode [3] occur during the opening hours, the mechanical ventilation (or the natural ventilation) provide the minimum airflow and the cooling system is switched to maintain the indoor temperature within the comfort zone.

The exploitation of the ventilative cooling lead to a decrease of cooling demand and electricity consumption for ventilation (calculated assuming a Specific Fan power (SFP) of 1 Wh/m^2) in both lighting power density cases (Table 5). However, the use of ventilative cooling seems to be not completely beneficial since heating demand increases its values by 30% in case of high lighting power density.

Table 4 Matrix of ventilation approach and ventilative cooling solutions based on different specific lighting internal gains for reference building

ID	Architectural archetype	Internal layout	Lighting power density	Ventilative solution	
CS	compact, purpose built, 2 floors,	Circular gallery in the middle connected with outside	qi= 50 W/m^2	Daytime Natural	Wind-buoyancy ventilation Wind-driven ventilation
			qi= 10 W/m^2	Daytime Natural	
ME	Steel and glass structure, open space with market stands, reconceptualised	Central gallery, big volume over two floor	qi= 50 W/m^2	Daytime and Night-time Natural	
			qi= 10 W/m^2	Daytime and Night-time Natural	
GE	1 floor on extended area, reconceptualised	One semi-circular gallery connected with three entrance from the outside. Part of the roof is used as parking area	qi= 50 W/m^2	Daytime and Night-time Hybrid	
			qi= 10 W/m^2	Daytime and Night-time Natural	
KA	2 floors, purpose built	Main big gallery which extremity end into two atria equipped with glazed surfaces. Part of the roof is used as parking area	qi= 50 W/m^2	Daytime and Night-time Hybrid	
			qi= 10 W/m^2	Daytime and Night-time Natural	
DO	2 floors, compact, purpose built	In total six galleries interconnected; two big atria	qi= 50 W/m^2	Daytime and Night-time Hybrid	
			qi= 10 W/m^2	Daytime and Night-time Natural	
BC	2 floors, compact, purpose built, construction on several years	Main gallery between the shops connected at six small galleries. The gallery roof is glazed	qi= 50 W/m^2	Daytime Hybrid	
			qi= 10 W/m^2	Daytime Natural	
ST	2 floors, compact	Main atrium extended on two floor with escalator in the middle; small galleries between shops connected with the atrium. Glazed surface on both atrium and galleries(at 2nd floor)	qi= 50 W/m^2	Daytime Natural	
			qi= 10 W/m^2	Daytime Natural	
GB	Multilevel, compact	Two circular atria in the middle of the mall. The atria roof is equipped with glazed modules	qi= 50 W/m^2	Daytime Hybrid	
			qi= 10 W/m^2	Daytime Natural	
WS	2 floors, extended	Two big galleries interconnected in three-point Circular glazed atrium with possible operable windows. Glazed surfaces on the 2nd floor	qi= 50 W/m^2	Daytime Hybrid	
			qi= 10 W/m^2	Daytime Natural	
PA	1 floor, extended, purpose built	Simple layout with one gallery facing shop one side and the outside on the other	qi= 50 W/m^2	Daytime Natural	Wind-driven ventilation
			qi= 10 W/m^2	Daytime Natural	

This percentage is restrained to only 5% with low lighting gain. In this last case, because of the consistent energy saving in cooling demand (about 50%), the heating demand increase can be tolerated. The heating demand increase is caused by a drop of the indoor temperature during mid-season because of too high airflow. High airflows create a sort of "sub-cooling" of the building, resulting in an increased heating demand. A possible solution to limit the increase of heating demand is to use smart control systems on windows opening, especially during mid-season. Smart control strategies and system can avoid the temperature drop.

Table 5 Potential energy saving because of the ventilative cooling solutions use

	q_i=50 W/m^2			q_i=10 W/m^2		
	Baseline	Hybrid ventilative cooling	Percentage savings %	Baseline	Natural Ventilative cooling	Percentage savings %
Cooling demand [kWh/(m^2 y)]	105	62	-41%	59	30.1	-49%
Electricity consumption for ventilation [kWh/(m^2 y)]]	202	52	-74%	202	0	-100%
Heating demand [kWh/(m^2 y)]]	105	150	+30%	175	184	+5%

5. CONCLUSION

The paper investigates the retrofit opportunities to exploit ventilative cooling in shopping centers' common areas (shop galleries and atria) in terms of external climate conditions and architectural features. Ventilative cooling climatic potential has been assessed for ten reference buildings considering two different levels of lighting power density. The results shows a linear dependence between the level of lighting power density and the percentage of hours of direct ventilative cooling use. This suggest that when facing a retrofit this synergy cannot be overlooked otherwise it could cause discomfort situations inside the building and to extra non-necessary investment costs for windows´ and openings´ actuators. With low internal gains, natural ventilative cooling is able guarantee thermal comfort condition for all the reference climates. On the other hand, with high lighting gains, hybrid ventilative cooling seems the most suitable strategy except for cooling dominated climates (Trondheim, Silute and Klaipeda) where the natural one is once again able to assure acceptable comfort conditions. The night ventilative cooling is predicted to be useful in the climates of Valladolid, Genoa, Catania and Wien.

In terms of architectural features, almost all the reference shopping centers are multilevel, they have galleries and atrium and only for two of them part of the roof is used as parking areas. All the features make feasible the exploitation of buoyancy ventilation by means of dedicated openings and windows located at different height.

The validation of the effectiveness of the ventilative cooling solution proposed for the Donau Zentrum shopping center, has shown a cooling demand reduction between 41% and 49% depending on the lighting gains. The electricity consumption for ventilation is as well, decreased or nullified depending on the ventilative cooling strategies, linked with the lighting internal gains values.

The analysis presented in this paper can be considered as a first step towards the detailed definition of ventilative cooling retrofit solutions, since we did not considering important design constraints (e.g building regulation and standards, HVAC system configuration and management and tenants' needs) due to the lack of available information,. Starting from this preliminary analysis, in further developments we will investigate computational fluid dynamics simulation (CFD) methodologies, able to predict the detailed air movement inside and outside the building, and to provide detailed thermal comfort analysis.

6. ACKNOWLEDGEMENTS

The research leading to these results has received funding from the European Community Seventh Framework Programme (FP7/2007-2013) under grant agreement n. 608678.

7. REFERENCES

Belleri A., P. T. (2015). Evaluation tool of climate potential for ventilative cooling. 36th AIVC- 5th TightVent- 3rd venticool. Madrid.

Bointer,R. (2014). D21,Shopping mall features in EU-28 + Norway,CommONEnergy project (FP7-2013 grant agreement no 608678). Obtido de http://www.commonenergyproject.eu/
BPIE. (2011). Europe´s buildings under the microscope, A country-by-country review of the energy perfomance of buildings .

Cory S. (2011). Formulating a building climate classification method. roceedings of Building Simulation 2011: 12th Conference of International Building Performance Simulation Association. Sydney.

Emmerich S. J., P. B. (2011). Impact of adaptive thermal comfort on climatic suitability of natural ventilation in office buildings. Energy and Buildings, 43(2101-2107).

Haase M. (2015). D25-Main drivers for deep retrofitting of shopping malls, CommONEnergy project (FP7-2013 grant agreement no 608678).

Hamlyn D., A. P. (2012). Ventilation approaches for shopping malls - An examination of natural and hybrid strategies. ASHRAE transactions vol. 118. Part 1, 575-585.

Howlett G. (2011). Survey of Retail Lighting-Market Trasformation Programme, Developing evidence for Government and business on energy using products. Building Research Establishment.

IEA EBC Annex 62 - Ventilative cooling. (2014-2017). http://venticool.eu/annex-62-home/.
Liddament M. (1996). A guide to energy efficient ventilation. AIVC.

Meteonorm. (2009). Obtido de http://meteonorm.com/

Moosavi L. (2014). Thermal performance of atria: An overview of natural ventilation effective design. Renewable and Sustainable Energy Reviews, 654-570.

Moosavi, L., Mahyudding, N., N, A. G., & M.A., I. (s.d.).

Schönberger, H. (2013). Best Environmental Management Practice in the Retail Trade Sector, European Commmission, Joint Research Centre-Institute for Prospective Technological Studies, Pubblications Office of the European Union, Scientific and Technical Research Series- ISSN 1831. Luxemburg.

OVERHEATING ASSESSMENT OF A PASSIVE HOUSE CASE STUDY IN SPAIN

Juan María Hidalgo*1, Theofanis Psomas2, Carlos García-Gáfaro3, Per Heiselberg2, Jose Antonio Millán1

1 University of the Basque Country UPV/EHU
Plaza Europa, 1.
Donostia-San Sebastián, 20018, Spain
**Corresponding author: juanmaria.hidalgo@ehu.eus*

2 Aalborg University, Dept. Of Civil Engineering
Sofiendalsvej 9-11.
Aalborg, 9200, Denmark.

3 Thermal Area, Lab. Quality Control in Buildings
of the Basque Government
Aguirrelanda, 10.
Vitoria-Gasteiz, 01013, Spain

ABSTRACT

In response to the European Energy Performance Buildings Directive 2010/31/EU and the Energy Efficiency Directive 2012/27/EU, buildings have increasingly become more insulated in order to reduce the heating losses to a minimum. However, this could also lead to the problem of indoor high temperatures during warm and transition seasons. Furthermore, the Intergovernmental Panel on Climate Change (IPCC) warns about increases in temperature of more than 4 ºC by the end of the century. Taking into account the different thermal comfort indices, this research analyses the overheating risk in a single family house built in Spain according to the Passivhaus standard. For the purpose of this research, we selected the following models: the Fanger Predicted Mean Vote (PMV) model defined in the ISO 7730:2005, the adaptive model defined in the EN 15251:2007, the criteria for overheating prevention defined in CIBSE TM52 in 2013 and the PH limitation about warmer temperatures. Moreover, we have analysed the influence of dwelling occupancy and the periods of verification over the results of each methodology.

The studied building has a high level of thermal insulation and air-tight envelope, reducing heat losses until a heating demand of 14 kWh/m2 per year. It is equipped with a convective heating system and a mechanical ventilation system with heat recovery, without any cooling system apart from the bypass configuration of the heat recovery unit and the window openings. The monitoring lasted more than a year, from January of 2013 until April of 2014, and includes both indoor environment and outdoor weather parameters.

The different criteria result in different outputs: According to the ISO 7730 standard, the discomfort caused by warm temperatures represents the 9.8 % of the non-heated season, which reaches up to 13.6 % when taking into account only the day-time rooms; this result is obtained through the weighted average of the temperatures in the kitchen, the living room and the dining room. On the other hand, the adaptive model EN 15251 leads to an outcome of only the 0.2% in the whole house and 1.7% in the weighted average temperature of the day-time rooms. The TM52 criteria for avoiding overheating risk shows that the building is not overheated because it meets the three criteria. The house doesn't meet the Passivhaus requirement, because the period over 25ºC exceeds 11.8 %.

Finally, we have analysed some guidelines about overheating risk assessment and proposed some improvements, such as including transition months of fall and spring, considering full time occupancy instead of specific timetables or splitting the building into different zones to detect local discomfort conditions.

KEYWORDS

Overheating, Passive house, Thermal comfort assessment.

1 INTRODUCTION

As a result of years of research, building energy performance has deeply changed the horizon of construction techniques and traditional parameters for building design. In recent years, many EU countries have developed their building policy for net Zero Energy Building (nZEB) (BPIE, 2015) as Passive buildings and it is necessary to control potential overheating in future warming scenario (IPCC, 2015). In this task, the socio-economic crisis impact in the construction sector has changed recently in the EU-28; evolving from a reduction of 18.6 % between 2008 and 2010 and 9,6 % between 2010 and 2013, to an actual significant increase of 6.1 % in 2013 and 2014 (Eurostat, 2015).

A literature review of the climatic context showed an uncertain future: Despite the fact that the global warming impact is unknown and depends on future actions, the last publication of the World Meteorological Organization (WMO) and the United Nations Environment Programme (UNEP) has already predicted a warming of +4 ºC in Europe by 2100 (IPCC, 2013). In such a case, the energy demand for building cooling will increase even in new buildings with a high level of insulation (McLeod, 2013). In 2010, the Zero Carbon Hub (ZCH) cautioned about overheating (OH), stating that "*Given the prospect of significant warming, well within the expected lifetime of homes, this risk will increase with potentially serious consequences"* (Zero Carbon Hub, 2010). Thus, in our case of intermediate regions with mild weather it could cause many unexpected TC problems because of the recently imposed high level of insulation and their lack of traditional natural cooling systems: windows.

Looking for an OH definition, CIBSE TM52 states: "it implies that building occupants feel uncomfortably hot and that this discomfort is caused by the indoor environment" (CIBSE TM52, 2013). A recent study (McLeod, 2013) shows many high insulated buildings where inhabitants report warm temperatures. Unfortunately, there is no specific regulation to prevent OH in the EU or in Spain, but there are several methodologies to set limits to warm indoor T based on the limits for warm Thermal Comfort (TC), such as the traditional Fanger ISO 7730, the adaptive EN 15251 or the more recent CIBSE TM52 for OH prevention. A recent review of current TC indices (Carlucci, 2012) concludes that it is necessary to set a new TC index for OH risk prevention.

Regarding the residential sector, Passivhaus (PH) is one of the most well-known design standards in Europe, with thousands of buildings developed since 1991. In the words of PH founder: "*A Passive House is a building in which thermal comfort can be guaranteed solely by heating or cooling of the supply air which is required for sufficient indoor air quality without using additional recirculated air*" (Feist, 2007). The PH standard sets a fixed high temperature limit of 25ºC, with less than 10% of the hours per year over this limit. There are many examples of OH prevention around Europe; some results point to user strategies such as ventilation and shading as the key to keep indoor comfort level during hot periods (Mlakar, 2011). They also show that particular attention should be paid to some common problems in the construction process, as these could severely impact the final performance of low energy buildings (Guerra-Santin, 2013). It should be noted that it is possible to address these problems by carrying out timely monitoring activities so as to ensure the performance of buildings' systems. Another study about a multifamily PH recently built in the United Kingdom indicates that 72% of monitored flats failed their design criteria (Sameni, 2015) and points to the user behavior as the most significant factor in increasing or decreasing the risk of OH.

Hence, this study will focus in the warm season, given that the main objective of this research is the assessment of the overheating risk. Besides, different types of rooms will be analysed in the study to verify if the methodologies guarantee the comfort conditions in rooms of different use and orientations.

2 CASE STUDY AND METHODOLOGY

2.1 Description of analysed Passive House

The data was acquired from a detached single family building with PH certification, built in 2012 in the province of Alava, in northern Spain. It was designed according to local town planning regulations, which required a traditional pitched tile roof, thereby increasing room height and heated air volume. The dwelling shape is rectangular and oriented to the south to maximise the solar collection in winter. In the centre of the south facade the main entrance leads to an open foyer that connects with the dining room and the family room; the rest of the house is accessed by a corridor (see Figure 1 below). The building structure of reinforced concrete consists of a foundation slab, pillars and two pitched roof slabs, which result in a considerable thermal mass. The main façade, detailed in Table 1, is made of clay blocks with both internal and external thermal insulation. The building presents a net floor area of 176 m^2 and a heated air volume of 500 m^3; the rooms have pitched or flat ceilings with heights ranging between 2.3 and 3.6 m. Details of the house can be found in (Hidalgo J.M., 2013).

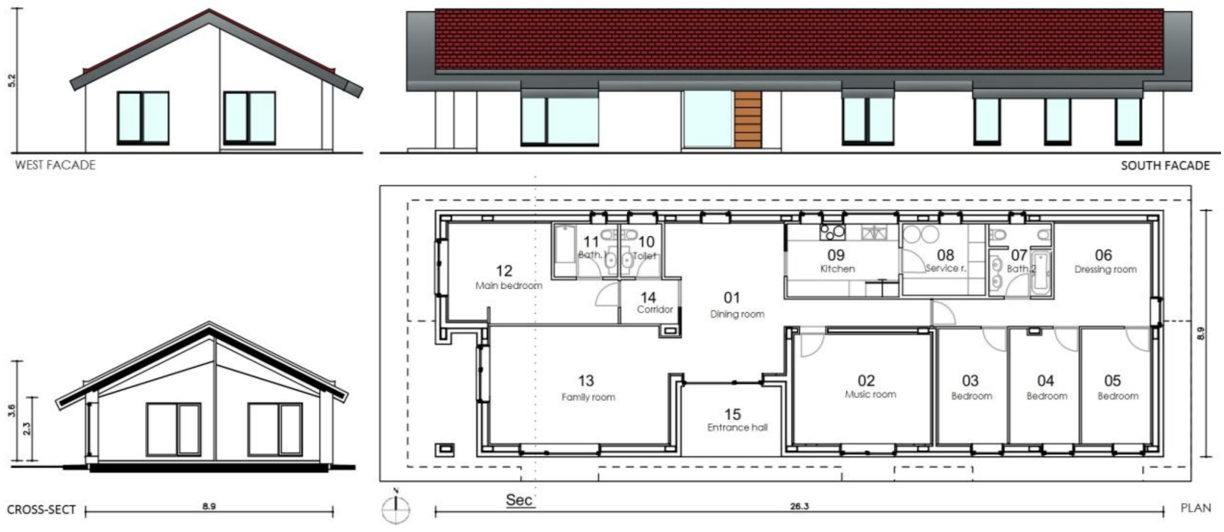


Figure 1: Passivhaus building plan, cross-section and main facades.

Table 1: Thermal envelope main elements and components

MAIN FACADE U = 0,143 W/m²K	**λ [W/(mK)]**	**width [mm]**	**GROUND SLAB U = 0,160 W/m²K**	**λ [W/(mK)]**	**width [mm]**
External plaster	0,870	6	EPS Neopor	0,036	160
EPS Neopor	0,032	160	Reinforced concrete slab	2,400	300
Mortar	1,300	15	EPS Neopor	0,036	50
Arliblock	0,460	200	Mortar levelling	1,300	50
Internal gypsum plaster	0,570	15	Floating wood floor	0,130	15
Rock wool	0,036	50			
Gypsum board	0,250	15			
ROOF U = 111 W/m²K	**λ [W/(mK)]**	**width [mm]**	**WINDOWS**	**U [W/(m²K)]**	**Solar gain [%]**
EPS Neopor	0,032	120	Frame, wood/alum. Mixed	1,00	-
Reinforced concrete slab	2,400	220	Glazing, 3pan, argon/Low E.	0,60	50
EPS Neopor	0,032	160	Window, average unit	0,9	-

The bioclimatic performance of the building was mainly conceived for the summer season, with pitched roof extensions that were designed as a means of protection against direct sun radiation in summer zenith; during winter it allows the absorption of the direct sun radiation. The entrance hall is conceived as a buffer-space and in winter it generates a greenhouse effect thanks to an outside glazing wall. The building heating demand was originally supplied by a biomass stove in the family room; however, it was replaced by two electric heaters located in the family room and in the dressing room. The indoor air quality (IAQ) is mainly provided by a mechanical ventilation system with exhaust air heat recovery (MVHR) with a measured seasonal performance (winter COP) of 86% during the 3 coldest months. During the summer season, the building is free-running and the cooling demand is provided by night-time ventilation due to HR bypass. The air exchange rate was set at 0.7 h^{-1} during all the monitoring period, except for some occasional family celebrations when the air flow was temporary increased. All the windows could be manually opened, and during the non-heated season the users used natural ventilation in combination with MVHR many warm days; it was verified by monitored temperature and relative humidity (RH). The hot water supply is provided by a combination of one solar thermal panel in the roof and a heat pump (HP).

2.2 Description of the Monitoring system

The acquisition system was installed to measure the building thermal response and indoor environment under normal occupancy conditions. The outdoor conditions were also measured in detail by a weather station placed in the plot of the house at a height of 3m. For a summary of all the parameters measured, types of sensors and uncertainties please refer to the Table 2 below. The indoor environment parameters were measured by air temperatures in all the rooms at a height of 2m, in conjunction with RH sensors in the main rooms. Additionally, some sensors were installed to measure the prospective vertical air temperature stratification in the rooms with high ceilings, such as the family room and the dressing room, where the air temperatures were measured at a height of 0.9m, 1.9m and 2.90m. Measurements were also made of some specific surface temperatures, such as walls, pillars, ceilings and floors of the main rooms. Furthermore, two additional sensors were placed to verify other TC parameters, like air velocity and globe temperature, over a period of several weeks throughout the different climatic seasons (spring, summer, fall and winter); thereby it was possible to verify the relation between air dry bulb temperature and operative temperature during different periods of the year.

The monitoring work was also aimed at gathering information for different analysis, which included the heating energy consumption, the window glazing and frame T, the performance of the entrance hall as sun collector, or the characterisation of the thermal envelope performance (with several months of heat flux measurements to calculate the facade transmittance according to ISO-9869-1). Table 2 below summarizes the main features.

Table 2: Main parameters measured by the data acquisition system.

Parameter	Sensor	Units	Num.	Uncertainty
T. air	RTD, PT100 (sheathed)	[ºC]	21	±0,2 ºC
T. surface	RTD, PT100 (encaps.)	[ºC]	57	±0,2 ºC
Relative Humidity	HIH-4000-001	[%]	6	±3,5 %
Heat flux	Ahlborn, Wärmefluss	[W/m^2]	3	±5 %
Electric power	JUMO, dTron 304	[W]	2	±4 %
Global H. Irradiance	Kipp&Zonnen, CMP11	[W/m^2]	1	±3 %
Meteorological Station	VAISALA, WXT520	[ºC], [mbar], [mm], [m/s], [%]	1	-
Data logger	AGILENT, 34980A	-	1	-

2.3 Overheating detection models and methodology

In a conversation with the inhabitants of the case study home, they shared their perceptions regarding the warm environment during some summer days and cold sensations in different parts of the building during winter. As a result, it was proposed to perform an exam of the Indoor Environment (IE), and specifically the Thermal Comfort (TC), in all the rooms of the home. Additionally, the rooms were classified following the criteria of orientation and use: Day-time (family room, dining room, and kitchen), Night-Time (bedrooms) and Services (service room, toilets, dressing room, corridors).

As we have seen before, the risk of OH has been assessed using many standards by calculating the IE warm limits according to an acceptable TC level. In order to perform this OH assessment, the current regulations and the most appropriate standards for the detection of OH risk were selected. First, the TC methodology provided in the current Spanish regulation for residential buildings was selected. The "Codigo Técnico de la Edificación" refers to ISO 7730 categories, also known as PMV-PPD model of Fanger (ISO 7730:2005). These limits are mandatory to every type of residential building, including the building of our case study, which doesn't have any cooling system. Second, the adaptive model EN15251 was selected, because this standard conforms specifically to buildings in free-running mode (without any cooling system during the warm season), which enables natural ventilation in every room (Olesen, 2012), the same features as in our studied building. This methodology applies specific categories to free-running buildings. This methodology calculates the limits depending on the outdoor running mean temperature (Trm). Third, with a view to measure the OH risk, it was instrumental to apply the most recent European methodology outlined in CIBSE TM52, despite the fact that it was designed bearing in mind Great Britain's weather conditions and British building regulations. Last, taking into account that the building was certified by Passivhaus Institute (PHI), it was interesting to ascertain if it meets the PH criteria to avoid OH, which consist on limiting the hours with average air temperatures over 25 ºC to less than 10 % of the time. All the methodologies used, as well as their temperature limits and periods of verification are summarized in Table 3 below. Given that most of these standards were developed originally for office buildings instead of residential use (CIBSE, 2013) (Carlucci, 2012), it is necessary to assess the impact of their assumptions about occupancy timetables and periods of verification, and to find out if they are valid for OH risk assessment in this residential building.

Table 3: Summary of TC standard limits according to indoor O.T.

TC Standard	Summer limits	Winter limits	Summer definition	Summer analyzed	Extended period
ISO 7730	24.5 ± 1.5	22.0 ± 2.0	(Non heating)	4Jun - 11Nov	15Apr - 30Nov
EN 15251	0.33 Trm +18.8 ±3	22.0 ± 2.0	(N.D.)	1Jul - 30Sep	15Apr - 30Nov
CIBSE TM:52	0.33 Trm +18.8 +3	(N.D.)	1 May - 30 Sep.	1May - 30Sep	-
PH overheating	25.0 (air T.)	25.0 (air T.)	(N.D.)	-	Full year

The period of verification is an important parameter for this assessment and, therefore, all available possibilities were examined. First, we studied the different definitions of the term "summer" according to astronomical, meteorological and temperature concepts (Alpert, 2004). According to the Meteonorm database and air temperatures parameter, the three warmest months in a typical meteorological year are: July, August and September. Second, the Fanger method ISO 7730 distinguishes between heated and non-heated periods. In our case, the heating system was switched off from the 4th of June till the 11th of November. In a similar way, the adaptive EN 15251 recommends the analysis of the whole year or a season, bearing in mind that the focus of our assessment is the summer season together with the transition months. Other more specific OH risk detection standards extend their periods of analysis to the whole year or at least they include part of spring and fall to prevent excessive solar gains during transition weeks after and before winter (Carlucci, 2012). This is precisely the approach advocated by the CIBSE TM52 methodology, with a longer period from May to September, a range probably selected to cover all the warm period in the UK. Given all these options, the study of the summer period has been complemented with an extended period that comprises the transitions of spring and fall: from the 15th of April until the 30th of November. below shows a summary of all these possible periods of verification.

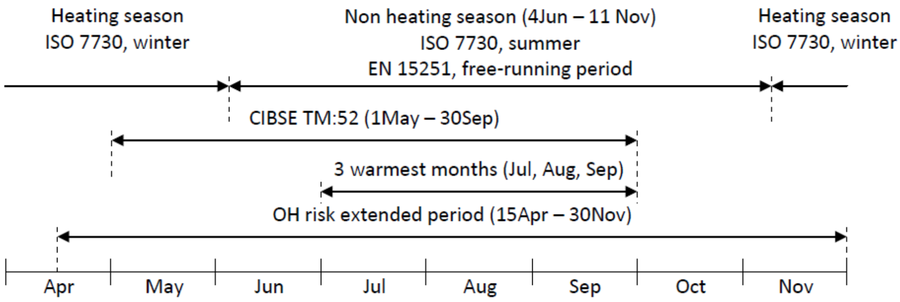


Figure 2: Periods of verification for summer TC conditions and OH detection

Additionally, the occupancy level indicates which hours are considered in the long-term analysis of the TC, as well as in the OH risk assessment. Most of the standards are only applicable whenever there are people at home. However, in our study we have proposed to measure also the unoccupied hours, mainly because future nZEB long time of response. This statement is based on the nZEB definition that assumes low heat losses through envelope and ventilation (BPIE, 2015). Hence, their IE will present fewer fluctuations than in traditional buildings. Taking into account all these issues, in our analysis they were assessed on two levels: full time occupancy and measured occupied hours. The timetable was an average of the real hourly occupancy detected by house monitored data during a representative number of weeks. For the purpose of calculating the average of the occupancy timetable, a week per month was selected during the 5-month period covering the spring and autumn seasons and, during the 3 months of summer, 3 weeks were selected. Figure 2 below shows the resulting timetable. For the purpose of the calculation, some days were excluded in order to account for the periods when the family was on holidays and out of their home for 1 day or more.

Day \ Hours	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Mon	5	5	5	5	5	5	5	5	0	0	0	1	4	4	3	1	0	3	3	3	5	5	5	5
Tue	5	5	5	5	5	5	5	5	0	0	1	1	4	5	4	1	0	1	1	2	4	5	5	5
Wed	5	5	5	5	5	5	5	5	1	0	0	1	4	4	4	1	0	1	3	4	4	5	5	5
Tue	5	5	5	5	5	5	5	5	2	0	0	1	5	5	5	3	1	2	5	4	3	4	4	4
Fri	5	5	5	5	5	5	5	5	0	1	1	0	4	4	4	2	1	2	1	1	2	3	3	4
Sat	5	5	5	5	5	5	5	5	5	5	5	5	5	5	4	4	2	3	3	3	4	3	3	3
Sun	4	4	4	4	4	4	4	4	4	4	4	3	3	3	3	2	2	3	4	4	5	5	5	5

Workweek

Occupancy check by hour along 5 weeks during workweeks within normal school calendar. Occupied if ≥ 3/5

Day \ Hours	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Mon	3	3	3	3	3	3	3	3	3	3	2	3	2	3	3	3	1	0	1	2	3	1	1	2
Tue	3	3	3	3	3	3	3	3	3	3	3	2	2	2	2	0	0	0	0	1	1	2	2	3
Wed	3	3	3	3	3	3	3	3	3	3	3	3	2	3	3	3	2	2	1	1	1	1	2	3
Tue	3	3	3	3	3	3	3	3	3	3	3	3	2	2	3	2	0	1	1	1	1	2	2	2
Fri	2	3	3	3	3	3	3	3	3	3	2	2	1	2	3	1	1	1	1	0	1	2	2	3
Sat	3	3	3	3	3	3	3	3	3	3	3	2	2	3	2	1	1	0	1	0	1	2	3	3
Sun	3	3	3	3	3	3	3	3	3	3	3	3	2	2	3	3	2	1	1	0	0	2	2	3

Summer week

Occupancy check by hour along 3 weeks of summer during school holidays. Occupied if ≥ 2/5

Figure 3: Building occupancy timetables for summer and work week, averaged results.

3 RESULTS

All results are shown following the recommendations of the Strategic Research Centre for ZEB of AAU (Afshari, 2013) and disaggregated by occupancy, building zones and periods of evaluation. Despite the fact that the analysis is focused in high temperatures, considering the relevance of IE, and Table 5 also include low temperature discomfort ranges.

3.1 ISO 7730:2005

Focusing first on the global performance of the house and the occupied hours, 9.8 % of the summer period presents high temperatures, compared to 7.5 % during the extended period. At the same time, if we take into account the full time occupation, the rate increases up to 13.9 % and 9.6 % respectively, showing that some of the hottest hours are reached in the afternoon, when there is nobody at home. In relation to room combinations and orientations, during occupied hours Day-time reaches up to 13.6 % while Night-time remains similar at 7.8 %; however, when taking into account full time occupation, the gap increases till 18.4 % and 11.1 % respectively. This confirms that the areas with higher activity are more exposed to overheating risk. Considering the different orientations, we can say that the East side is fresher and the West side is hotter during all periods, while the hot temperatures in the North and South orientations are slightly below those in the West side. It is unexpected to find low temperatures around 20 % of the time; but this could be explained by both strict temperature limits of the standard (as we have seen in Table 3) and because the inhabitants preferred a low temperature set-point for heating.

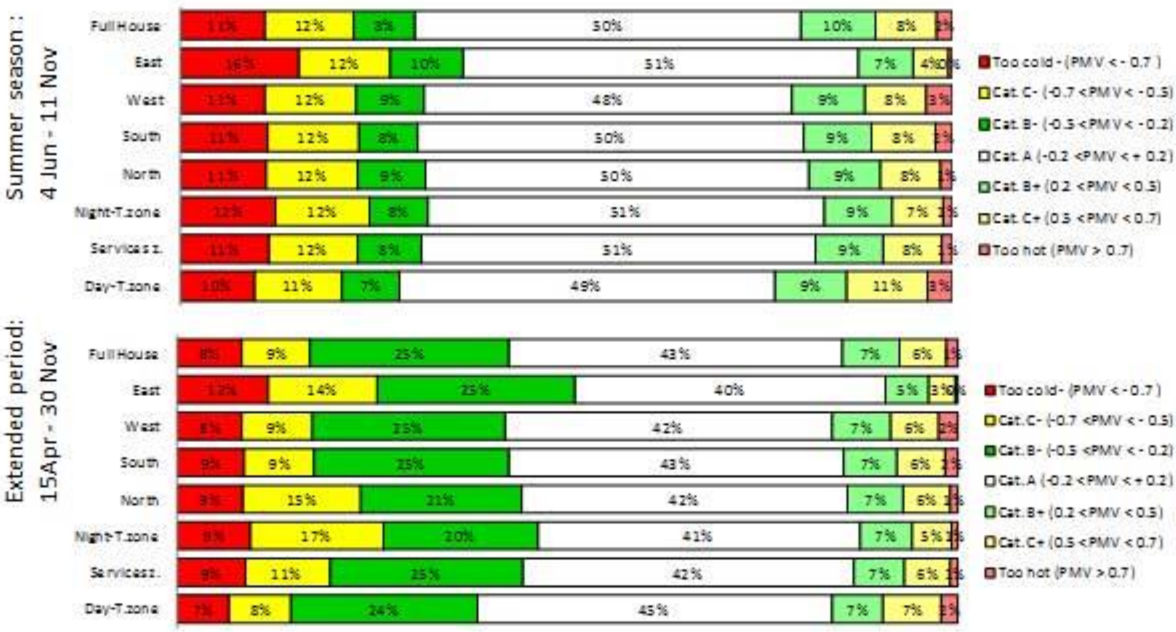


Figure 4: Thermal Comfort by building zones, considering real occupation, in summer and extended period.

Table 4: Discomfort percentages according to ISO 7730, occupied hours

Disconfort and period	Day-time z.	Service z.	Night-time z.	North	South	West	East	Whole House
High T. summer	13,6%	8,9%	7,8%	9,2%	10,4%	11,3%	5,0%	9,8%
Low T. summer	21,0%	23,0%	24,5%	23,0%	23,2%	22,9%	27,2%	22,6%
High T. extended	9,5%	6,8%	5,9%	7,0%	7,9%	8,6%	3,8%	7,5%
Low T. extended	14,7%	19,6%	26,5%	23,6%	17,6%	17,3%	25,7%	17,1%

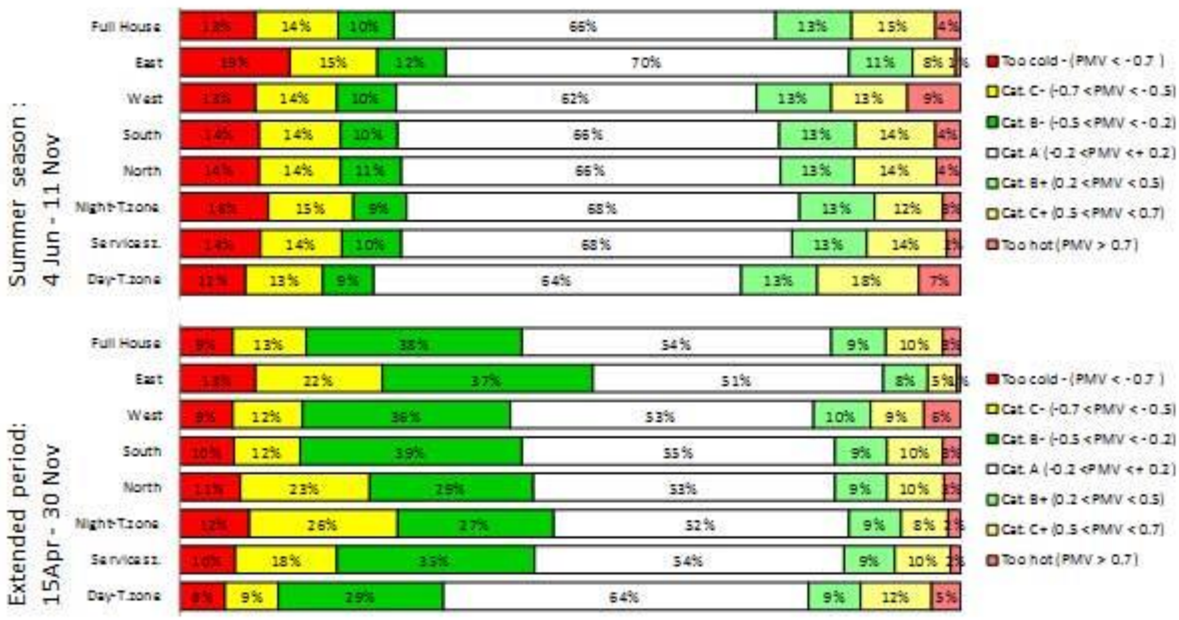


Figure 5: Thermal Comfort by building zones, considering full time occupancy, in summer and extended period.

3.2 EN 15251:2007

Figure 6 below shows the weighted indoor operative temperatures during the extended period, using different markers for non-heated season, spring and fall transitions. The main results according to EN 15251 indicate that discomfort rate by high temperatures stands at 0.2 % for both the summer and the extended period. There is low T discomfort during the transition weeks, which represents the 2.8 % of the extended period. Despite the aforementioned data, we can say that most of the hours fall within the adaptive limits of TC for IE, remaining within Categories I or II for 99.8 % of the summer period and 97.0 % of the extended period. Table 5 shows additional data about the different discomfort ranges taking into account criteria such as room use, orientation, occupied hours versus full time occupancy, and summer (non-heated, free-running period) or the extended period. Moreover, the table shows the relevance of selecting not only the "occupied hours", but also de "full time period", as the whole house discomfort caused by warm temperatures increases up to 0.6%.

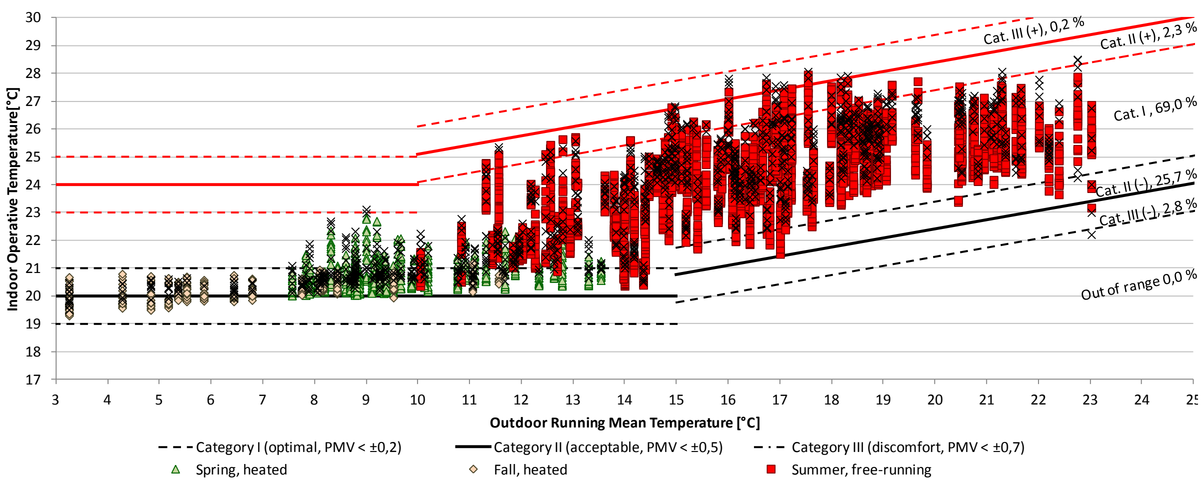


Figure 6: Thermal Comfort by EN 15251, during extended period.

Regarding the results in relation to the different room use and orientations, the Day-time rooms present higher temperatures, with 1.7 and 3.2 % for summer occupied hours and full time occupancy respectively. This is also the case in regards to orientations, with slightly higher temperatures in the West facade on comparison with the rest of the house. The high temperatures reach up to 0.7 % in occupied hours and up to 2,1 % in full time occupancy, showing this side is the most vulnerable part of the house for OH risk. Once again, the discomfort caused by low temperatures is located mainly in the bedrooms.

Table 5: Discomfort percentages according to EN 15251, by occupancy and period of verification.

Discomfort (high/low T), period and occupancy	**Day-time zone**	**Service zone**	**Night-time zone**	**North**	**South**	**West**	**East**	**Whole House**
High T. Summer, Occupied	1,7	0,0	0,0	0,1	0,3	0,7	0,0	0,2
Low T. Summer, Occupied	0,1	0,0	0,2	0,0	0,1	0,0	0,4	0,0
High T. Extended, Occupied	1,2	0,0	0,0	0,1	0,3	0,7	0,0	0,2
Low T. Extended, Occupied	0,1	7,0	14,3	11,9	1,7	2,4	10,0	2,8
High T. Sumer, Full time o.	3,2	0,2	0,2	0,3	0,7	2,1	0,0	0,6
High T. Extended, Full t. o.	2,2	0,2	0,2	0,3	0,7	2,1	0,0	0,6

3.3 CIBSE TM52

The first criterion of this methodology focuses on hours that exceed at least 1K over the EN15251 limits in a rounded value. In our building, despite having some Category III hours, their higher T only exceeded 0.4 K over the limit, so it can't be considered as exceedance values for this criterion and the building meets the criterion. Therefore, also meets the 2nd criterion, which required to check if any day presented a weighted temperature exceedance of 6 K and even the 3rd which limits the peak hour to an exceedance of 4K at any hour. Concluding that according to this methodology the three criteria are fulfilled, thus it could not be considered as OH.

3.4 Passivhaus requirements

Given the specifications of the PH standard, we have taken as reference a typical year of 8760 hours and we have checked all possible hours with Air T higher than 25 ºC in the weighted house temperature. Taking into account the Air T during the extended period, we find that the temperature surpassed 25ºC during 11.8 % of the time, which exceeds the 10 % threshold of the standard.

4 CONCLUSIONS

This case study shows that the Passive house built in the North of Spain cannot be considered as overheated, according to the results obtained in the latest regulation for OH prevention, CIBSE TM52 and EN15251. However, we have found that, with the 11.8 % of the time over 25 ºC, it does not meet the Passivhaus limitation. Similarly, it does not comply with the current Spanish regulation, because the frequent high and low temperature exceeds the ISO 7730 limits.

Every standard has proven to be useful or complementary in OH risk detection. The Spanish regulation methodology, based on ISO 7730:2005, can easily detect which zones of the building have higher risk of high temperatures, but it does not work so well during summer, mainly because in Europe or Spain these kind of residential buildings usually don't have any cooling system. On the other side, the EN 15251:2007 reflects more realistically the user perception, since looking into the days where they used additional natural ventilation they were close to the category III limit; so they felt warm and consequently used natural ventilation. Given that this was the first summer that users spent completely in the house, hopefully they will learn how to ventilate more efficiently in future summers. The CIBSE TM52 has proven a good step forward in OH risk detection, but it is still poorly defined in different building areas and limits rounding. Passivhaus OH prevention was conceived many years ago for cold central European climate, and after many years only a few variations have been introduced for warm climates. Despite the fact that the houses are high insulated and airtight, the IE limits have to be improved according to the outdoor environment influence, as happened with the EN 15251.

Many other problems have been detected in relation to local discomfort conditions depending on different activity areas or orientations. So, we recommend that the OH verification needs to be applied not only to a weighted house temperature but to two or more zones, depending on the size and shape of each case. It could be also necessary to make further research about local discomfort conditions for the most vulnerable rooms.

The obtained results for this highly insulated house show that the main shading elements were designed for summer sunpath and not for transitional months such as September or October, when there are still a lot of hours of radiation, with quite warm temperatures and few cloudy days. These factors in conjunction with the thermal mass of the building after summer could generate unexpected overheating in the end of summer. This issue must be considered in more detail in future building design.

We have seen how important it is to set properly the occupancy and the periods of verification according to each standard and knowing the differences between them. Further research is suggested in future case studies, considering the findings of the present study.

5 ACKNOWLEDGEMENTS

The authors would like to thank especially the Uriarte family for their collaboration and kind help during the monitoring campaign of their home; Construcciones Urrutia and CLIM-Estudio de arquitectura for giving access to building information; the Department of Civil Engineering of the Aalborg University (AAU) for its guidance; Marta Vidal for the proofreading of this study; and the Thermal Area of the Laboratory for the Quality Control in Buildings of the Basque Government for their equipment, facilities, continuous support and help.
This work has been funded by the Researcher Training Program of DEUI of the Basque Government (Spain), as a PhD fellowship.

6 REFERENCES

Eurostat (2015). *Industry and construction statistics - short-term indicators*. Available at: http://ec.europa.eu/eurostat/statistics-explained/

Buildings Performance Institute Europe - BPIE (2015). *Nearly zero energy buildings definitions across Europe*. Available at: http://bpie.eu/

Sameni S, Gaterell M, Montazami A, Ahmed A (2015). *Overheating investigation in UK social housing flats built to the Passivhaus standard*. Building and Environment, 2015, doi: 10.1016/j.buildenv.2015.03.030

Afshari, A., Olesen B.W., Le Dréau, J., Foldbjerg, P., Jensen S.Ø., Larsen, T. (2013). *Definition of the indoor environmental quality used for Net Zero Energy Buildings*. Strategic Research Centre for Zero Energy Buildings, Aalborg.

Guerra-Santin, O., Tweed, C., Jenkins H., Jiang S. (2013). *Monitoring the performance of low energy dwellings: Two UK case studies*. Energy and Buildings, 64, 32 – 40.

McLeod R.S., Hopfe C.J., Kwan A. (2013), *An investigation into future performance and overheating risks in Passivhaus dwellings*, Building and Environment, 70, 189-209.

Hidalgo, J.M., Escudero-Revilla, C., García-Gáfaro, C., Odriozola, M., Millan, J.A., Uriarte, A. (2013). *Monitorización y seguimiento del comportamiento energético de una vivienda con certificación Passivhaus.* Proceedings book of 5ºConferencia Española Passivhaus, Gijón.

Olesen B.W. (2012). *Revision of EN 15251: Indoor Environmental Criteria*. REHVA Journal August 2012, 6-12.

Carlucci, S., Pagliano, L. (2012). *A review of indices for the long-term evaluation of the general thermal comfort conditions in buildings*. Energy and buildings, 53, 194-205.

Mlakar J., Štrancar, J. (2011). *Overheating in residential passive house: Solution strategies revealed and confirmed through data analysis and simulations,* Energy and Buildings, 43, 1443 - 1451.

Zero Carbon Hub – ZCH (2010). *Carbon compliance for tomorrow's new homes, a review of the modelling tool and assumptions, overview of findings and recommendations*. Available at: http://www.zerocarbonhub.org/resourcefiles/.

Feist W. (2007). *Passivhäuser in der Praxis*. Bauphysikkalender 2007. Berlin.

EN 15251:2007. *Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics.*

ISO 7730:2005. *Ergonomics of the thermal environment -- Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria.*

Alpert P., Osetinsky I., Ziv B. and Shafir H. (2004). *A new seasons definition based on classified daily synoptic systems: an example for the eastern mediterranean.* International journal of climatology, 24, 1013–1021.

Intergovernmental Panel on Climate Change – IPCC (2013). *Climate Change 2013: The Physical Science Basis*. IPCC Working Group I Contribution to AR5, Stockholm. ISBN 978-92-9169-138-8.

AUTOMATIC NATURAL VENTILATION IN LARGE SPACES: A PASSIVE VENTILATION TECHNOLOGY FOR PASSIVE BUILDINGS.

Flourentzos Flourentzou1*, Samue Pantet1, Katia Ritz2

1 Estia SA
EPFL Innovation Park
1015 Lausanne, Switzerland www.estia.ch
**flourentzou@estia.ch*

2rk studio bienne+monthey alleestrasse 11
2503 Biel, Switzerland www.rkstudio.ch

ABSTRACT

For zero and low energy buildings, high-energy efficiency ventilation is very often confused with a complex mechanical ventilation system with heat recovery. In school gymnasiums, where large volumes have to be ventilated, and where intermittent occupation is very usual, demand controlled natural ventilation has several advantages, making this technique very attractive. High stack height makes natural ventilation very efficient, limiting the necessary number and dimensions of windows. Large air volume, with high height, combined with intermittent occupation, avoids high pollutant concentrations, especially in the occupied zone, because of air stratification, with fresh air near the floor and pollutants/heat on the ceiling. High stack effect, offers free ventilation all over the year. Natural ventilation is very attractive for architects because of no ducts, no apparent mechanical components, and low cost The article shows the ventilation concept of two gymnasiums, one equipped with demand controlled / summer comfort controlled natural ventilation and the other with a hybrid ventilation system with heat recovery. The monitoring of the naturally ventilated gymnasium results shows the perfect air quality that natural ventilation offers in this type of buildings in winter and summer. Simulations comparing a fully mechanically ventilated hall with a hybrid one and with a purely naturally ventilated hall show the energy performance of the different systems. A life cycle assessment shows that controlled natural ventilation has comparable and even better performances than a heat recovery system. Zero electricity for ventilation all over the year, and no embodied energy for ventilation compensate non-recovered heat.

KEYWORDS

natural ventilation, ventilative cooling, demand-controlled ventilation, passive cooling.

1 INTRODUCTION

Natural ventilation was assimilated in the beginnings of the passive and zero energy trend to high-energy consumption, lack of ventilation control and lack of comfort. For some energy labels, natural ventilation is allowed only under very strict control, automatic window opening in every space for example (like Minergie®). Some energy labels developed in cold countries are extended in southern countries, where intermitted heating and cooling is needed, without reconsidering the ventilation strategies according to the local needs and energy rationality, and they impose mechanical ventilation with heat recovery, just because heat recovery is part of the label (like passivhaus). Energy regulations in southern countries based on EPBD give credits to heat recovery and penalties to natural ventilation (like Cyprus eppd), while recovered primary energy in mild climates is much less than electricity to run a mechanical system (Flourentzou 2013). New challenges for zero energy buildings, with higher cooling needs in highly insulated and air tight buildings are identified by IEA annexe 62, oblige us to reconsider natural ventilation. High ventilation rates, needed to evacuate undesired heat, make natural ventilation an interesting strategy for a global energy performance. But controlled natural ventilation may be an interesting technique with a very positive global environmental balance even in winter (Flourentzou 2013, Flourentzou 2015) because it has no embodied energy and no electricity consumption to move the air. Natural ventilation limits are not energy related because the energy balance is globally positive. It will be shown again in this article in chapter 3. Natural ventilation limits are mainly related to ventilation control and thermal comfort. In chapter 2 we will show how smart design may manage control and comfort risks in a large space (gym in a primary school). The first year monitoring shows that controlled natural ventilation may offer perfect air quality and perfect summer and winter comfort, even for difficult occupation schedules with occupation varying between 30 and 200 children. In chapter 3 we compare a life cycle assessment of purely natural ventilation with hybrid and purely mechanical system.

2 DESIGN OF A NATURALLY VENTILATED GYM

2.1 Architectural constraints and characteristics.

Figure 1. Two joint volumes compose the building. On the left of the picture we can see the gym and on the right the classrooms. The pure line of the building is an architectural wish in the macro scale, to guarantee the wished "monolithic" form in dialogue with the alpine environment and the traditional village urban surroundings.

"Commune de Savièse" organised an architectural contest to build a small 10-classroom building with a gym. rk studio proposal got the first price, with a particular unified volume, as a response to the need to integrate a contemporary modern building in a traditional preserved Wallis mountain village on the Alps. The pure line of the form impose several serious constrains on natural ventilation design. The building was first designed as a typical passive building, intended to get a Minergie® label, with two distinct bidirectional ventilation systems, with heat recovery, one for the classrooms and one for the gym. The gym was designed as a closed mechanically ventilated volume and the classrooms had some windows that could open. From the outside the building is in dialogue with its alpine environment, through the pure line of its form, but also from the inside, through the pure form of glazed openings, framing unique fragments of landscape.

Figure 2. The pure line is a desired characteristic also in the micro scale, with construction details offering pure openings to frame landscape from the inside, without obstructions.

The architectural language of the prized project was in coherence with a closed mechanically ventilated building of the initial ventilation strategy. Windows did not have any functional use. However, very soon in the design phase, technical installation high cost became a limit to the project, and the necessary ducts, to bring 6000 m3/h of air in the gym space, necessary to ventilate up to 200 children in a school cantina, integrated to this space, was a cumbersome element, in opposition to the building's pure line. Cost analysis of different solutions and architectural advantages, like duct integration and room savings for installations, motivated the design team to choose purely natural ventilation for the gym and mechanical ventilation only for the classrooms and offices. The cost of the mechanical ventilation system of 2000 m3/h was around 100'000 CHF, and the duct diameter was 2 X 75 cm for the fresh air and 2 X 75 for the exhaust. Natural ventilation does not represent any extra cost, because the 4 m2 of necessary openings on the top and bottom of the space are required by fire protection regulations, related to smoke evacuation. The difficulty of natural ventilation is to find a way to integrate architecturally large openings, necessary to create a stack effect of 6000 m3/h.

Figure 3. The design trick, to respect the architect wish for pure glazed openings was to dissociate air path from light path. On the picture we may see an opaque opening on the top of the space, guaranteeing evacuation of hot air on the top of the space and avoiding the creation of a hot buffer in the roof triangle. Glazing on the bottom offers only light and view.

Figure 4. On the left picture, we can see the building as it was on the architectural contest poster. On the right as it was realised. The only difference is the opaque windows that we may guess on the top of every triangular roof of the gym on the right picture.

2.2 Window position and dimensioning of air path and flow rate

During summer ventilation, fresh air enters from the basement as it is shown on figure 5 and leaves the building from the top openings as it is shown in figure 6. Summer ventilation is controlled according to inside and outside temperature. When inside temperature is higher than 18°C and

outside temperature is smaller than inside temperature, bottom and top openings open. Ventilation stops when inside temperature falls < 18°C or when there is strong rain or strong wind.
During winter, air enters from the lateral top openings, which are positioned lower than the top front opening in order to avoid cold draughts. When CO2 concentration > 1000 ppm, top openings open 10%. Openings are closed when CO2 < 600 ppm.

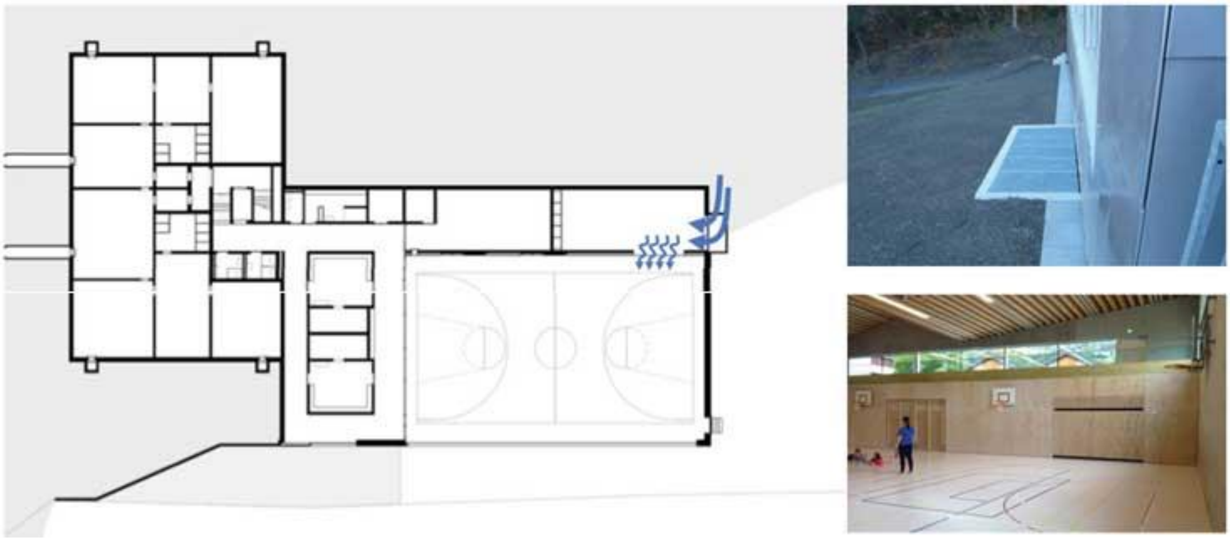


Figure 5. A bottom 4 m2 opening brings fresh air in the basement from a light/ventilation well in the sport equipment storing room. Air enters through two automatic windows and passes through the perforated door that we can see on the bottom right picture.

Figure 6. A top 4 m2 opening exhausts air on the top of the cantina space on the gym situated on the gym balcony (as it is seen on the right picture) and 2 additional 1m2 openings as they are seen on figure 3 allow additional better distribution of air evacuation. On this figure we can see the night ventilation summer strategy with fresh air entering on the basement and going out from the top openings.

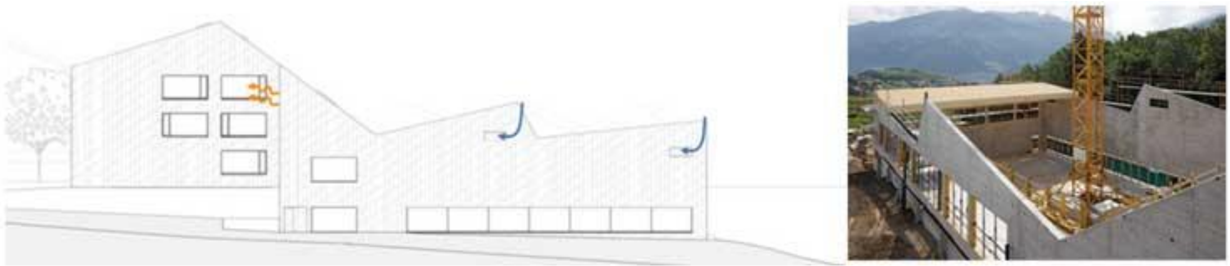


Figure 7. For winter natural ventilation strategy, air enters from the top lateral openings and goes out from the top front opening in order to avoid cold draughts. Cold air entering at 8 m height, is mixed with hot air until it falls to the floor. Users have not reported any cold draughts in the first year use of the gym.

We have simulated natural ventilation airflow with DIAL+ software (Paule 2004) to dimension the openings in order to guarantee enough night ventilation for free cooling of the building. The position of the openings is intended. The bottom opening position in the storing room activates the concrete thermal mass of this extra space and stores coolness during night. It avoids also cold

draughts because event in summer there are fresh days that might create cold draughts. The passage of the air through a perforated wooden door makes airflow
laminar and well distributed. Positioning of the automatic window in the storing room
prevents children to play with it. Extra protection grid for the inside prevents children to access the automatic openings and from outside against rain.
Top openings assure that there is not creation of hot air buffer in the roof triangles. The big exhaust opening position on the opposite side of the air inlet assures swiping the whole space
with fresh air. It is positioned on the top of the cantina space to allow direct pollution evacuation where the higher human concentration takes place.

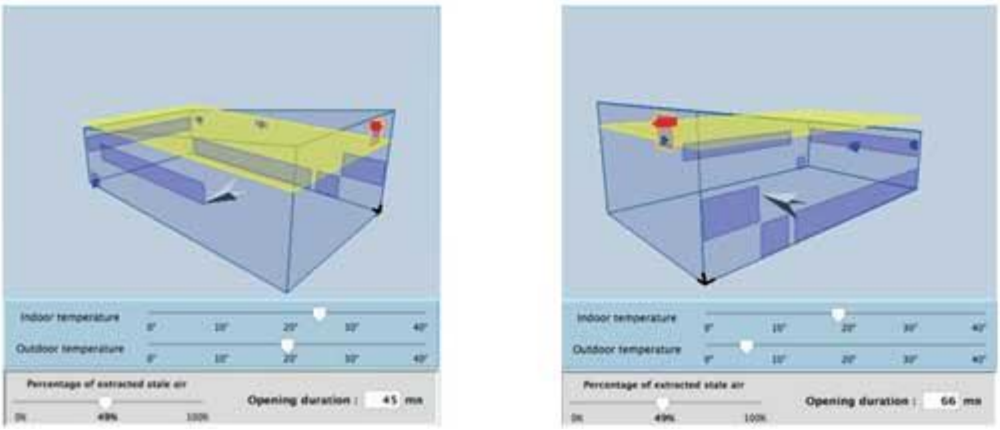


Figure 8. DIAL+ simulations to dimension window openings. In the left image we can see summer ventilation with 7’278 m3/h entering from the bottom opening and 2 lower top openings under the neutral level at 9.34 m when ΔTin-out = 5°C. On the right picture, we can see winter ventilation strategy with the neutral level in the middle of the top opening at 10.4m with 2094 m3/h exhaust air and 422 m3/h inlet air from the same opening and 1077m3/h + 1406 m3/h from the lateral top openings under the neutral level.

As we can see from figure 8, control of the neutral level is essential to elaborate a ventilation strategy, because it determines where fresh air enters and where exhausted air is evacuated from the building. We see from the DIAL+ images than in summer, the main mass of inlet air enters from the bottom opening while during the winter strategy it enters from the 3 top openings. Fixing the in and out temperature in the software we can have the orders of magnitudes of the airflow rate, but detailed dynamic simulations with a coupled air/heat model determine if the airflow rate is enough, especially for the cooling strategy.
The initial intentions were to avoid solar control and there was no night ventilation strategy. Dynamic simulations showed (figure 8) showed that night ventilation and solar control is essential for the building comfort.

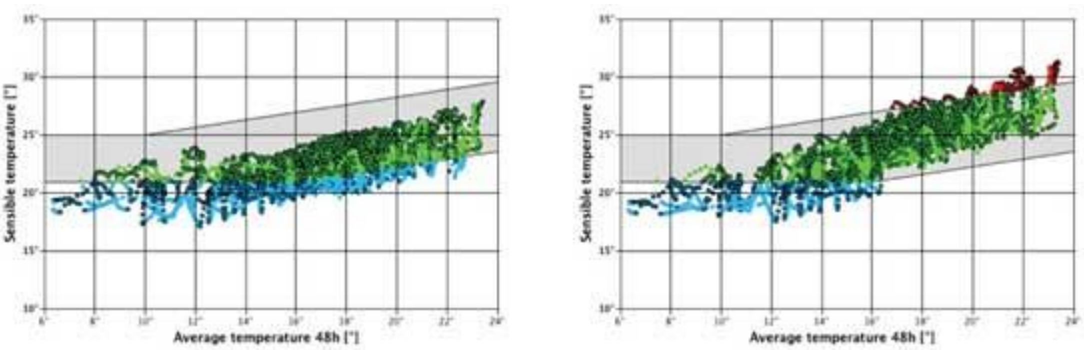


Figure 9. DIAL+ dynamic simulations for summer comfort. In the left image we can see the building behaviour with good solar control and night ventilation strategy 2-3°C under the upper comfort limit of EN 15251 norm. On the right graph, with ventilation only during working hours and no solar control, the building does not comply with the minimum comfort conditions of the norm.

2.3 Results of one year monitoring.

From figure 8 we can see that CO2 concentration never passes over 1100 ppm offering a perfect air quality. This graph was produced during the first 2 months of operation. After that the CO2 threshold for window opening was slightly reduced from 1000 ppm to 900 ppm and an extra opening was planned every working day at 6:00 am for 15 minutes to assure new fresh air every working morning. As we see during 8th and 9th of November, air infiltration assures only half air change day (CO2 concentration falls to 400 ppm after 48 hours).

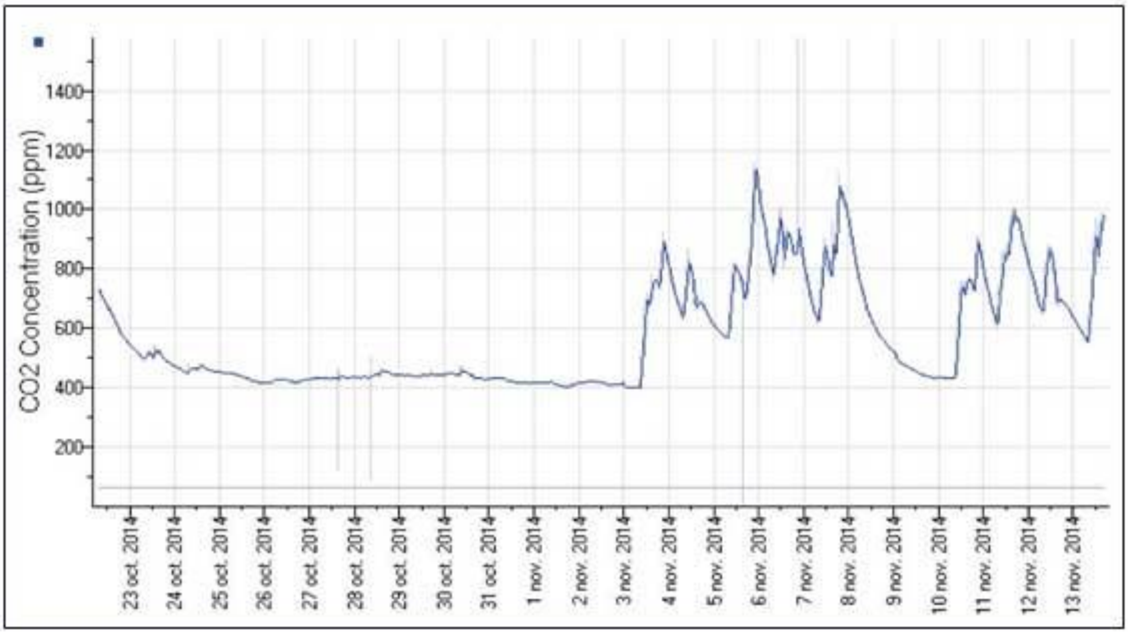


Figure 10. CO2 concentration during the wintertime. From Oct. 23 to Nov. 3 it was a holiday period and 8 & 9 Nov was a weekend.

As we see from the results, a reasonable and controlled infiltration is not "the enemy of energy performance". On the contrary, it offers energy-free controlled ventilation during not working hours of ~1 air change per day. Demand control ventilation limits airflow to the strict minimum. From the graph we may deduce analysing CO2 decays that during the time where the space is occupied with ~40 children, the room is ventilated with 750-850 m3/h in the month of November and when the space is not occupied with 120-420 m3/h (mean of 240 m3/h) of infiltration. From the CO2 decay analysis, we deduce that real occupation is much less than the nominal norm schedule for dimensionning. Less people occupy the space during less time. A demand control strategy is the best solution to transform this reality into energy savings. In this case, although 240 m3/h of mean infiltration corresponds to n50<0.5, the fact that there is a huge volume as fresh air reservoir, infiltration is almost sufficient to provide good air quality. CO2 concentration rises over 1000 ppm only after 15h:00 and openings are open only for few hours to complete ventilation.

The temperature monitoring for summer comfort covered the period of summer 2015, where high temperatures occurred even in the alpine regions. We tested one week without applying night ventilative cooling (10-17 July) and one week applying ventilative cooling (18-25 July). Although during this period the gym is closed without any occupation, we can see that only with solar gains, and although solar shading is automatic, air temperature in the cantina space at the first floor temperature rises up to 27°C, while during the ventilative cooling week, air temperature never rises more than 26°C although outside air temperatures are between 32 to 34 °C. Temperature results are in accordance with simulation results.

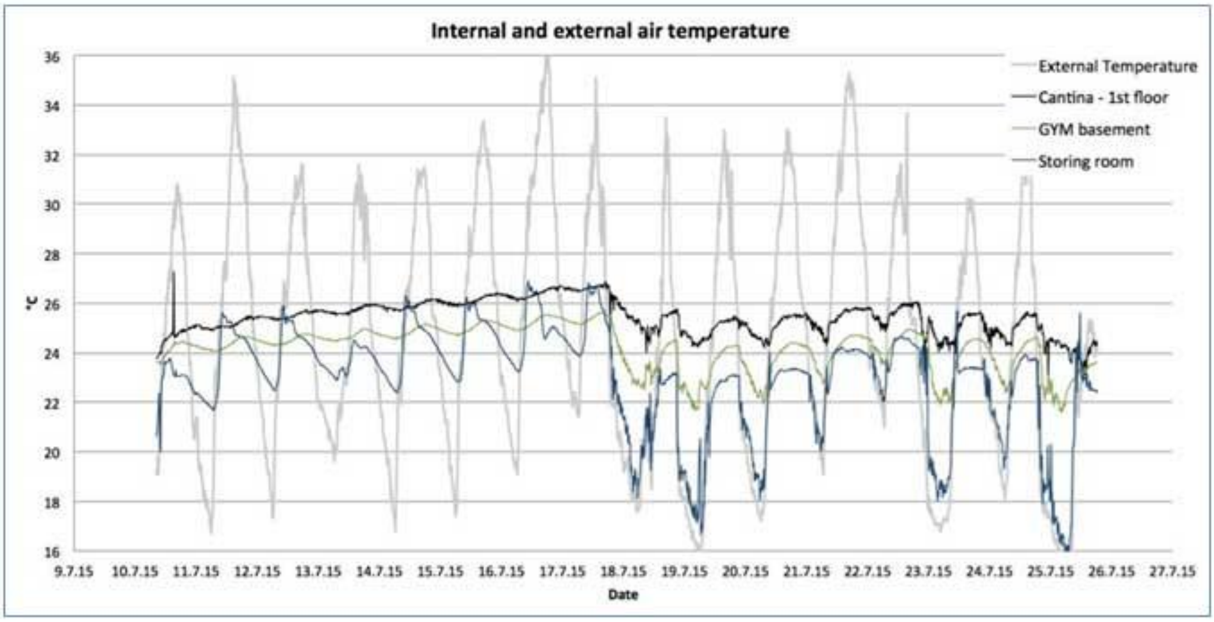


Figure 11. Internal and external temperature during hot summer period. During the first week there is no night ventilation and during the second, night cooling is activated.

3 COMPARATIVE LIFE CYCLE ASSESSMENT FOR 4 ALTERNATIVES

We compare the overall life cycle assessment of heating and electricity consumption, including embodied non-renewable energy, environmental impacts and life cycle costs for 4 ventilation cases. Case 1 is the pure natural ventilation, demand control on a CO2 sensor for air quality and on temperature and climatic conditions for comfort. The system is presented in details in the previous chapters. Case 2, is a hybrid solution, with 2000 m3/h bidirectional ventilation with heat recovery of overall efficiency 70% and specific power input for ventilation 0.50 W/(m3/h), also equipped with demand control based on CO2 concentration,

as for natural ventilation. During high occupation of the cantina, with up to 200 children, (one hour per day) and for night cooling, the system is naturally ventilated. This solution is applied on a similar project, where the building owner is obliged to install a heat recovery system to meet the requirements of a Minergie P® label, imposed by the Cantonal Energy Law. Case 3 is the same as alternative 2 but without demand control. Alternative 4 is the initial scenario of case 2 that was rejected for its high investment cost.

We calculate heat demand for each case with the Swiss software Lesosai applying the Swiss Norm SIA 380/1, (SN 520 380/1 2009) (this Swiss norm is based on the European norm EN ISO 13790). For natural ventilation we considered 0.5 m3/h.m2 during occupation and 0.22 during non-occupation due to infiltration. These values come from a detailed analysis of the CO2 concentrations variations and decays in November. During the natural ventilation analysis, mean temperature was at 8.1 °C, similar to the mean temperature during the whole heating season (8.1°C). For heat recovery units, we considered during, use, 30% of the 2000 m3/h to take into account heat recovery of mean η 70% for mechanical ventilation and 0 out of use hours in addition to 0.22 m3/h.m2 of air infiltration. For the demand control version of

mechanical ventilation, we keep the 0.22 m3/h.m2 of air infiltration and on the remaining 0.28 m3/h.m2 fresh air, we apply a factor of 30% to take into account 70% heat recovery efficiency.

This makes an overall fresh air ventilation of 0.3 m3/h.m2 during use for demand control

mechanical ventilation and 0.5 m3/h.m2 for standard heat recovery mechanical ventilation. The data for environmental impacts for the ventilation systems come from KBOB database

(KBOB 2014). System costs come from EPIQR method cost database and fuel cost from the federal statistics office, mean price 2010-2014. For natural ventilation we considered equivalent embodied energy for 8 m2 of doors doubled with aluminium metal sheet cover. We neglected the embodied energy of 8 chain motors, control cables and control electronic box.

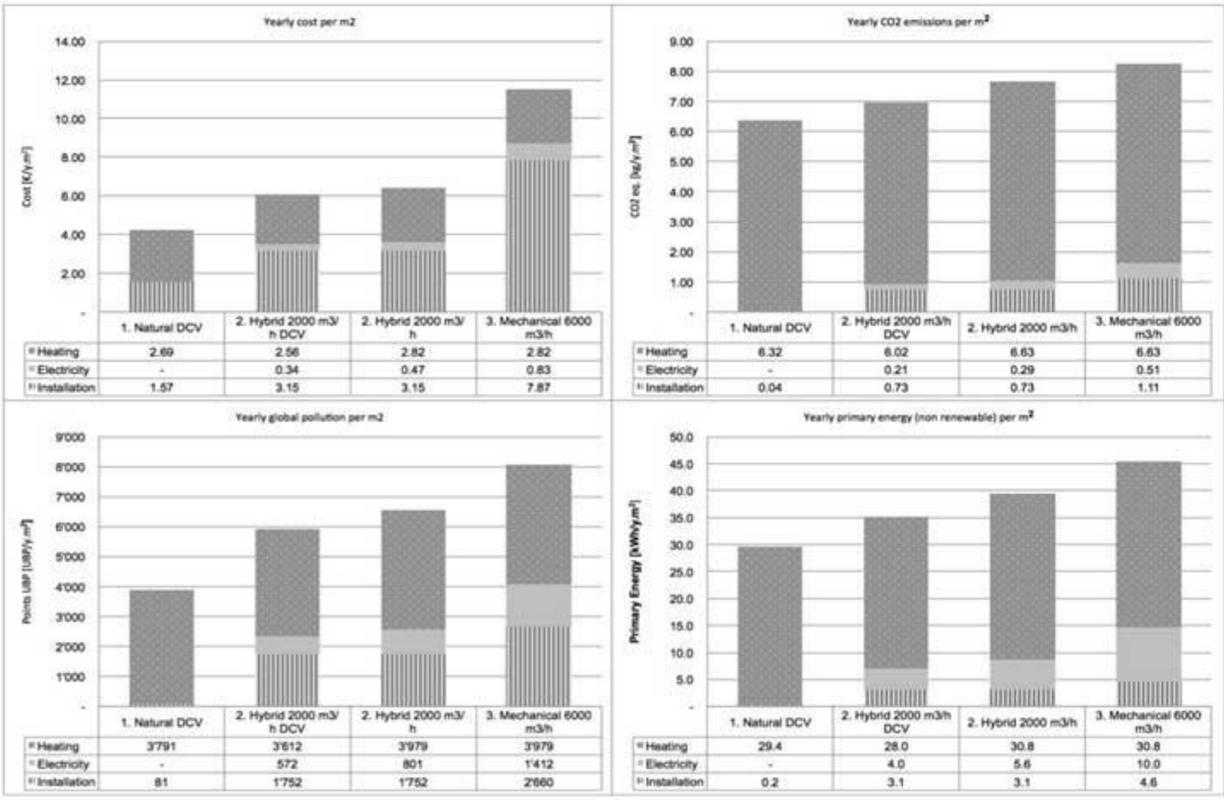


Figure 12. Results of life cycle cost, CO2, global solution and primary energy of 4 ventilation variants, applied on a building, considered to be heated by an oil heating system, η 0.85.

As we see from figure 2, demand controlled natural ventilation of large spaces is largely better solution than non-demand-controlled mechanical ventilation equipped with heat recovery and significantly better even than demand controlled mechanical ventilation with demand control. Although heat recovery reduces thermal losses, it increases electrical consumption for fans and embodied energy and environmental impacts of the system have analogous influence as the higher cost. For large spaces with intermittent occupation, with very high density during short time, a large system for full mechanical ventilation (case 4), increases considerably cost and embodied energy. In case 4 we can see that night ventilation of large space with mechanical fan increases considerably electricity primary energy consumption. These conclusions are valid for a central European climate. We tested if the results are valid for full occupation of the gym during all the day with adults. The difference between the systems is slightly smaller but the order is the same.
Large spaces present 3 significant advantages that make controlled natural ventilation an adequate system assuring best comfort and air quality:
- The height of the building creates a high stack effect making ventilation very efficient.
- The unique large volume creates a large reservoir of fresh air, making possible the exploitation of infiltration for ventilation. In the examined case, infiltration of ~240 m3/h in a volume of ~5000 m3, although it represents infiltration of a high airtight building n50<0.5, it is almost sufficient for adequate ventilation of the whole space, occupied sporadically by 25 –
50 children. CO2 concentration very rarely rises over 1000 ppm to activate window opening. Window opens during winter only after the infiltration potential is completely used. This makes the overall quantity of fresh air of the same order and event better than for a heat recovery system.- Mechanical ventilation for night cooling cost a significant amount of primary energy. Every time that natural ventilation may be used instead of a mechanical system it should be preferred. For

large spaces it is very easy to cool the whole space by controlling only 2-4 openings, reducing cost for the openings and automation. In this case full control is possible with only 4 openings. The air quality - comfort results and the energy and environmental analysis show that demand control natural ventilation for large spaces is the best solution for passive and zero energy buildings.

4 CONCLUSIONS

Natural ventilation for high single volume gyms, and generally for large spaces, is a very simple concept and the results show that it may offer a perfect air quality in winter and perfect temperature comfort during summer. Natural ventilation is a low cost solution with a lot of architectural advantages (pure line, no space consumption for machines, no ducts in the space). Natural ventilation is an ecological solution because it has very low embodied energy and system environmental impacts.

Design may very easily manage the different possible risks (cold draughts, intrusion risk, weather risks, noise risks) for large spaces with automation of few openings. Automatic control for public spaces is essential for good air quality and comfort control.

The life cycle assessment showed that natural ventilation, with demand controlled airflow is also very energy efficient. For Central Europe climates, its global life cycle energy and environmental performance is equivalent or better than bidirectional mechanical ventilation with heat recovery. It is a perfect solution for passive buildings where airflow may be controlled according to occupation.

The architectural advantages of natural ventilation are undeniable. There are no ducts, and openings may be easily dissimulated if they are not desirable to be present in the space. Dissociation of light path from air path is a smart design technic to obtain the optimum air path and exploit the physical properties of the building structure. In this case for example, dissimulation of 4 m2 of opening in the underground, made possible to exploit thermal mass of the concrete storing rooms where the air passes first and avoids direct droughts near the opening. Compliance to fire regulations gives also interesting synergies to minimise architectural, cost and environmental impact of natural ventilation.

Natural ventilation of large spaces is a ventilation system that must be designed by an architect, or by a collaborative team with at leas an architect and a building physicist.

5 REFERENCES

SN 520 380/1(2009) L'énergie thermique dans le bâtiment (thermal energy in buildings), SIA, Zurich. KBOB, 2014, Données des écobilans dans la construction, https://www.kbob.admin.ch, 1-2014.pdf

Paule, B., et al., (2012). DIAL+Suite : a new suite of tools to optimize the global energy performance of room design, Status Seminar, Zurich.

Flourentzou F., Pantet S., 2013, Are heat recovery systems really necessary for nearly zero energy buildings in mild climates? AIVC conference, Athens

Flourentzou F., Pantet S., 2015, Theoretical and real ventilation heat losses and energy performance in low energy buildings, AIVC conference, Madrid.

THEORETICAL AND REAL VENTILATION HEAT LOSSES AND ENERGY PEFRORMANCE IN LOW ENERGY BUILDINGS

Flourentzos Flourentzou[1], Samuel Pantet

1 Estia SA
EPFL Innovation Park
1015 Lausanne, Switzerland
**flourentzou@estia.ch*

ABSTRACT

Ventilation in low energy refurbished buildings is the cause of a big part of energy losses. In order to reduce this impact, some energy regulations prescribe a solution (such as the Swiss energy Law, prescribing heat recovery) and others prescribe a system global performance (such as the EU delegated regulations No 1254 and 1253 / 2004 determining a global energy performance label of the ventilation system). In this article we analyse global theoretical performance of 6 ventilation systems in an apartment residential building and we compare real performance of a demand controlled ventilation system, incompatible with the Swiss prescriptions, with the performance of heat recovery system, compatible to the prescriptions, in two case studies of low energy refurbishment.

Theoretical performance comparison shows that different energy saving strategies may produce similar effects. Very low fan electricity consumption combined with reduced demand-controlled airflow rate may produce energy savings comparable to heat recovery. This phenomenon is accentuated with high performance heat production systems, with low embodied primary energy (such as high performance heat pumps, renewable district heating or wood boilers). In these cases, primary energy for electricity, necessary to recover heat, added to system embodied energy, may be higher than the recovered non renewable primary energy.

The comparison of real energy consumption of the systems in the case studies shows that the small differences between theoretical energy performances between heat recovery and demand-controlled ventilation, is negligible compared to the energy losses due to occupant behaviour (window opening, high internal temperature), while the investment cost of the two systems may vary by > 450% and the life cycle cost by >150%.

This comparative studies show that it is preferable to prescribe an energy performance of the global system, instead of prescribing a specific system or strategy, especially in an energy environment where primary energy for heating and cooling production is continually becoming better and ventilation technology is continuously evolving.

KEYWORDS

Ventilation, heat recovery, nearly zero energy buildings.

1 INTRODUCTION

We are interested on ventilation energy performance and two different prescriptive approaches are concerning it. The first approach, the one of the UE regulations (EU 1254 / 2014) and (EU 1253/2014), are interested on the system performance. They attribute an energy label to ventilation units and prescribe minimum and benchmark performances. The second approach, this of the existing Swiss cantonal regulations (MOPEC 2014), imposes a system. Swiss regulations prescribe

heat recovery for ventilation systems with more than 1000 m3/h and functioning more than 500 hours.
Regulations prescribing a solution deprive liberty of action, considering implicitly that the prescribed solution is the only way to achieve an objective in all the possible cases. However, the overall energy efficiency of heat recovery, prescribed by the Swiss regulations, depends on the amount of extra primary energy, necessary to run the heat recovery system and on the amount of primary energy, necessary to produce energy for ventilation heat or cooling losses. In this article we are interested only on heat losses in residential apartment buildings. Obligation for heat recovery was considered necessary, and "the only solution", in the 90s, when unidirectional ventilation units were consuming electricity, more or less, the half of electricity consumed by a bidirectional heat recovery system and when heat production was essentially provided by oil or gas boilers of low efficiency. Today, technological progress makes affordable and tends to generalise heat pumps of high COP for heat production, wood boilers, and district heating. Recovered heat losses contain much less primary energy than in the 90s. Today in the European market, annual COPs of heat pumps vary between 3 and 4.5. Considering a non-renewable primary energy factor of electricity 2.7, one kWh of recovered heat losses contains only 0.6 to 0.9 kWh of primary energy, compared to 2.7 kWh contained in 1 kWh of electric energy to run the fans. Even fossil energy sources are more efficient than in the 90s with condensation boilers of 8-10% higher efficiencies. Technological progress changes also the situation in regard to the energy necessary to provide controlled ventilation. DC motors for extraction fans offer very high efficiencies with low pressure-drop and specific power input may be less than 0.1 W/(m3/h). Hybrid systems, with boosted stack effect extraction, offer even more interesting energy performances with mean specific power input less than 0.05 W/(m3/h), not functioning for 1000-2000 hours per year when stack effect is sufficient. These specific power inputs must be compared to 0.32 to 0.45 W/(m3/h) of very good bidirectional ventilation units. Demand control options may reduce drastically the mean airflow rate by 30-50% according to occupation (Flourentzou 2011), (CSTB 2013). Some systems, like demand-controlled ventilation may combine all these new advantages, offered by innovative technology: reduced airflow rate, low fan energy consumption. With demand controlled stack natural ventilation, electricity for fan is completely avoided. Overall energy balance and energy performance is required in order to quantify these advantages, using updated information according to the current state of technology. Energy balance, taking into account heating and cooling produced by a standard heat pump, combined with life cycle assessment of the ventilation system, showed that, for office buildings, ventilation heat recovery is energy effective in northern climates (like Oslo) while in southern climates (like Nicosia) is definitely not energy effective (Flourentzou 2013). For central Europe climates, heat recovery effectiveness depends on the heat production energy performance and energy source primary energy factor. This study showed that for office building heated and cooled by mean performance heat pumps (COP 3.5), energy balance of heat recovery bidirectional units is neutral. Extra primary energy consumption for running the fans, defrosting and extra embodied energy for the system, is just compensated by recovered heating and cooling energy. Here we will be concentrated in multifamily residential buildings in a central European medium climate (Geneva).

In the first part of the article we are interested on the global energy performance of different ventilation systems. We will first evaluate their performance according to ecodesign directive. We will also perform a detailed life cycle evaluation, taking into account the primary energy for fan electricity, primary energy for ventilation heat losses, including infiltration, primary energy for defrost, as well as embodied energy for the ventilation system.

In the second part, we will compare the global energy performance of two real high-energy-performance case studies and we evaluate the real energy consumption of the ventilation system. One of the compared buildings is equipped with a heat recovery unit and the other with demand-control hybrid ventilation.

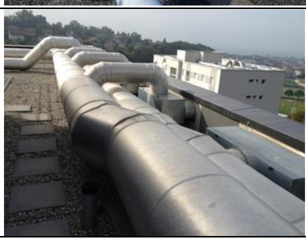
2 GLOBAL ENERGY PERFORMANCE OF VENTILATION SYSTEMS

2.1 Compared ventilation systems

Table 1. The 6 analysed ventilation systems. SPI is the nominal specific power input at reference airflow rate, η is the nominal heat recovery efficiency and CTRL is the control parameter.

		SPI [W/m^{3}h]	η_t	CTRL
1. Nat	Stack natural ventilation – adaptive local demand control	0.00	-	0.65
2. Hyb	Hybrid extraction ventilation – adaptive local demand control	0.05	-	0.65
3. DCV	Extraction unidirectional ventilation unit – adaptive local demand control	0.16	-	0.65
4. old UVU	Old (extraction) unidirectional ventilation unit– high/low velocity	0.08	-	0.95
5. new UVU	Modern (extraction) unidirectional ventilation unit– high/low velocity	0.09	-	0.95
6. HR BVU	Bidirectional ventilation unit with heat recovery	0.44	0.84	0.95

Table 2. Building and ventilation main characteristics

1			Natural ventilation	Renovation 2006
			Heating reference area	4000 m^2
			Number of apartments	62
			Maximum airflow rate	Stack effect
			Control	Local adaptive
			Ventilation power	0 W
			Heating system	Gas
2			Hybrid ventilation	Renovation 2010
			Heating reference area	2481 m^2
			Number of apartments	31
			Maximum airflow rate	6000 m^3/h
			Control	Local adaptive
			Ventilation power	15X16 W at 4500m^3/h
			Heating system	Gas
3			Extraction ventilation	Construction 2008
			Heating reference area	1222 m^2
			Number of apartments	12
			Maximum airflow rate	2070 m^3/h
			Control	Local adaptive
			Ventilation power	2X115W at 2200m^3/h
			Heating system	Wood pellets
4			Extraction ventilation	Construction 1985
			Heating reference area	9’812 m^2
			Number of apartments	31
			Maximum airflow rate	30’400 m^3/h
			Control	H.Speed/L.Speed
			Ventilation power	2’880W at 30400 m^3/h
			Heating system	Gas
5			Extraction ventilation	Renovation 2015
			Heating reference area	9’812 m^2
			Number of apartments	31
			Maximum airflow rate	9’140 m^3/h
			Control	H.Speed/L.Speed
			Ventilation power	759 W at 9140 m^3/h
			Heating system	Gas
6			Heat Recovery	Renovation 2011
			Heating reference area	5’739 m^2 X 2
			Number of apartments	70 X 2
			Maximum airflow rate	5’880 m^3/h
			Control	H.Speed/L.Speed
			Ventilation power	2’580 W at 5880 m^3/h
			Heating system	Oil / wood (in 2016)

The 6 selected systems cover a large spectrum of ventilation systems. Buildings of cases 1, 2 are high-energy standard buildings, recently refurbished (Minergie labeled GE-147 and GE-146). The existing ventilation system, for both, was a stack effect extraction system, with a concrete individual duct bringing fresh air in the WC and kitchen from the basement and extracting in the same rooms to the roof (typical Swiss system in the buildings constructed 1920-1960). The existing system was transformed to adaptive humidity sensitive extraction system. The ducts bringing the air in the WC from the basement were cancelled. New air inlets on the windows, equipped with humidity sensitive devices, adapt the airflow rate between 5 and 35 m^3/h according to the room occupation. In the first case, the ducts on the roof are not changed, and air extraction is purely stack. In the second case, hybrid low-pressure drop coaxial ventilators boost stack effect when is needed.

Case 3 is a recent low energy building (Minergie labelled GE 073), with extraction ventilation – demand controlled with humidity sensible devices in every room (local demand control).

Cases 4 and 5 is actually a single building, constructed in the 80s with a standard ventilation system for its period, but case 5 represents the project for replacement of the current oversized extraction ventilation units ~2.4 $m^3/h.m^2$ with a new system, equipped with low energy ventilators, and air flow re-dimensioned according to the current standards ~1.2 $m^3/h.m^2$.

Case 6 is a recently renovated couple of buildings, according to high-energy standards. The buildings are equipped with bidirectional ventilation system with heat recovery.

Cases 2 and 5 were monitored for one year to understand the difference between projected and real energy consumption (chapter 3).

2.2 Ventilation performance according to EU ecodesign directive

European regulation EU 1254/2014 describes the labelling framework, characterising ventilation unit energy performance. This regulation defines the Specific Energy Consumption indicator (SEC), which is the difference between a negative reference SEC and specific energy consumption for fans, heating and defrosting. Electrical energy consumption is calculated in terms of primary energy consumption with primary energy factor 2.5 and heating energy is considered without correction factor, except of a global heating efficiency of 75%. Regulation EU 1253/2014 indicates the minimum requirements to be applied by the member states until 2018 (class D or better) and fixes as benchmark class A for bidirectional ventilation units and class B for unidirectional residential ventilation units of <1000 m^3/h.

$$SEC = t_a \cdot pef \cdot q_{net} \cdot MISC \cdot CTRL^x \cdot SPI - t_h \cdot \Delta T_h \cdot \eta_h\text{-}1 \cdot C_{air} \left(q_{ref} - q_{net} \cdot CTRL \cdot MISC \cdot (1-\eta_t)\right) + Q_{defr} \quad (1)$$

Equation 1. SEC - Specific Energy Consumption [kWh/m^2.y] according to [EU 1254/2014]

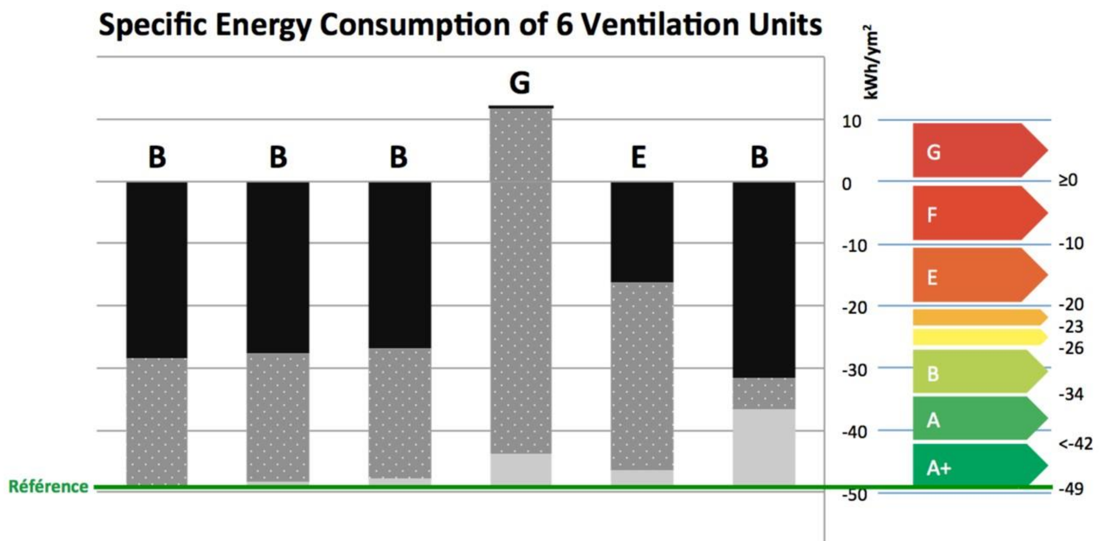
The first term of equation 1 is expressing the specific fan primary energy consumption. T_a is the annual operating hours (8760 h per year), pef is the primary energy factor (2.5), q_{net} is the net ventilation rate demand per m^2 of heated floor area (1.3 $m^3/h.m^2$), MISC is an aggregated general typology factor (1.1 for ducted units and 1.21 for non ducted units), CTRL is a ventilation control factor (1 for manual control, 0.95 for clock control, 0.85 for central demand control and 0.65 for local demand control) and x an exponent taking into account the non linearity between thermal energy and electricity saving (1 for on/off single speed, 1.2 for 2-speed, 1.5 for multi-speed and 2 for variable speed). SPI is the Specific Power Input at the reference airflow rate (kW/(m^3/h)).

The second term of the equation is expressing the difference between a reference heat energy loss due to reference natural ventilation and the effective heat energy loss. T_h are the hours of heating season (5112 h for the medium climate), ΔT_h is the average difference in indoor - 19K- and outdoor temperature over a heating season, minus 3K correction for solar internal gains (9.5 K for average climate), η_h is the average heating efficiency (0.75), C_{air} the specific heat capacity of air (0.00034 kWh/m^3.K), q_{ref} is the reference natural ventilation rate (2.2 m^3/h.m^2) and η_t is the thermal efficiency of heat recovery. The third term, Q_{defr} is equal to zero for unidirectional ventilation units and for regenerative heat exchangers (the only ones used by the systems compared in this article).

All the parameters are fixed by regulations, except of SPI and η_t, which are specified by the ventilation unit providers. q_{net} is fixed to 1.3 m^3/h.m^2 for systems dimensioned according to current regulations but case 4 is adapted for an existing over-dimensioned system with measured q_{net} of 2.4 m^3/h.m^2.

We have calculated SEC for the 6 ventilation systems and we show the results on figure 1. As SEC is a relative indicator, compared to reference SEC due to reference natural ventilation flow rate heat losses, the result is generally negative. Positive SECs are the worst systems of class G. Benchmark SEC according to UE1253/2014, for bidirectional ventilation units, is -42 kWh/y.m^2, i.e. Class A and -27 kWh/y.m^2 for unidirectional ventilation units, i.e. Class B.

As we can see on figure 1, all tested Adaptive Control Ventilation Units (Units 1, 2, 3) are of class B with SEC between 26 and 18 kWh/y.m^2, while the tested bidirectional ventilation unit (SPI 0.45 10^{-3} kW/(m^3/h) heat recovery of nominal value 84%) obtains also class B, with SEC -31.5 kWh/y.m^2. Clock demand, unidirectional ventilation without demand control, even with a good fan of SPI 0.1 kW/(m^3/h) obtains class E, while existing systems are of class G.



	1	2	3	4	5	6
-Fan	0.0	0.7	1.4	5.2	2.5	12.4
-Heat	20.7	20.7	20.7	43.9	30.3	4.8
SEC	-28.3	-27.6	-26.9	12.0	-16.3	-31.7

Figure 1. Results with the SEC and energy class of the 6 ventilation units. On the left of the graph representing the SEC, Fan and Heat energy consumption we can see the ecodesign label thresholds and on the bottom the table with the calculations for each unit..

We applied EU 1254/2014 methodology not for formal purposes. Although some of the examined residential ventilation units are over 1000 m^3/h, not subject of the regulation obligations, calculations give a comparative approach, using a coherent formal methodology. Purely stack effect extraction ventilation, equipped with adaptive control, case 1, is also out of the regulation perimeter, but we have assimilated this system to a unidirectional ventilation unit with a fan consuming 0 energy.

This comparative calculation shows first that existing oversized ventilation systems present a very high energy saving potential. Oversizing in the past was common not only because of cheap energy. Smoking in public spaces and at work, in most European countries, was prohibited between 2006 and 2010. Many of the existing ventilation systems were dimensioned for smoking places and today they are oversized.

In the graph we can also see that adaptive demand control ventilation is of Class B. Low energy consumption for fans, combined to reduced airflow rate, give a good global result. Innovation for better control and promotion of natural or hybrid demand controlled ventilation, with very low or zero fan energy consumption, are good and low cost solutions.

The comparison shows also that heat recovery of high thermal efficiency is not sufficient for high global energy performance. The compared unit has good heat recovery efficiency (84%), but the relatively high Specific Power Input (0.44 $W/(m^3/h)$ degrades the global Specific Energy Performance to class B. For high-rise buildings and for buildings with complex apartment arrangements where verticality is not possible, ducts create high-pressure drop and it is not possible to have sufficiently low fan energy consumption to obtain class A (SPI<0.32 $W/(m^3/h)$.

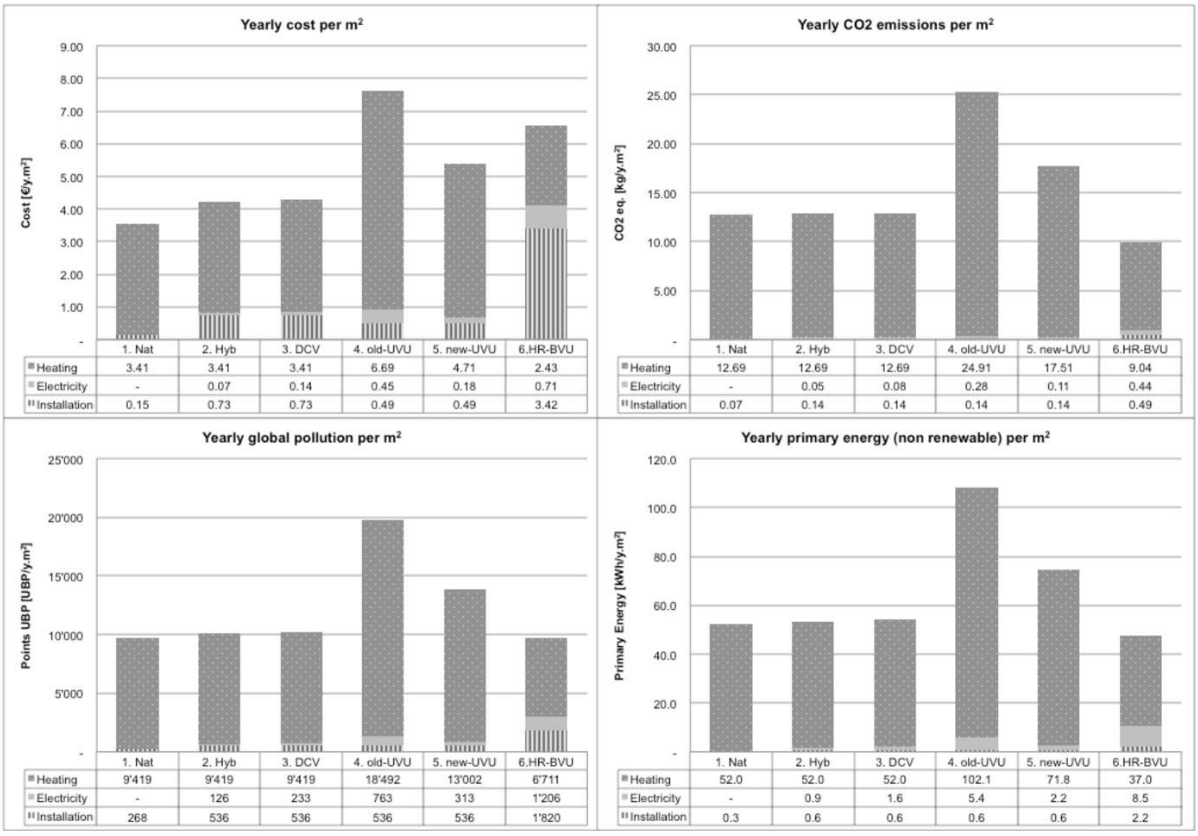
2.3 Ventilation performance according to life cycle assessment.

As EU 1254/2014 regulation uses a lot of default reference values, we try to confirm the conclusions with a life cycle assessment, taking into account design characteristics rather than reference values. We will also examine the effect of heating primary energy effect on ventilation overall performance. Instead of an overall heating efficiency of 75% and electricity primary energy 2.5, we will consider real primary energy of two energy systems: oil boilers of global system η 0.85 and a heat pump of global system COP 3.5.

We compare the 6 ventilation systems of table 1 applied on the renovated building case 6 of table 2. We apply the same dimensioning airflow rate for all systems, i.e 30 m^3/h per bedroom and living room (120 m^3/h in 4 room apartments and 90 m^3/h for 3 room apartments). We calculate heat demand for each case with the Swiss software Lesosai applying the Swiss Norm SIA 380/1, (SN 520 380/1 2009) (this Swiss norm is based on the European norm EN ISO 13790).

Table 3. Values of q_{system} (net airflow rate due to ventilation system), $q_{infiltration}$ (infiltration and parasite airflow rate), AHD - annual heat demand, calculated with Lesosai software, and AEC – annual electricity final energy consumption of fans.

		1. Nat	2. Hyb	3. DCV	4. old UVU	5. new UVU	6.HR-BVU
q_{system}	$m^3/h.m^2$	0.70	0.70	0.70	2.40	1.2	0.32
AHD	$MJ/y.m^2$	130.4	150.1	150.1	256.0	180.0	106.2
AEC	$kWh/y.m^2$	0	0.33	0.61	2.0	0.82	1.73



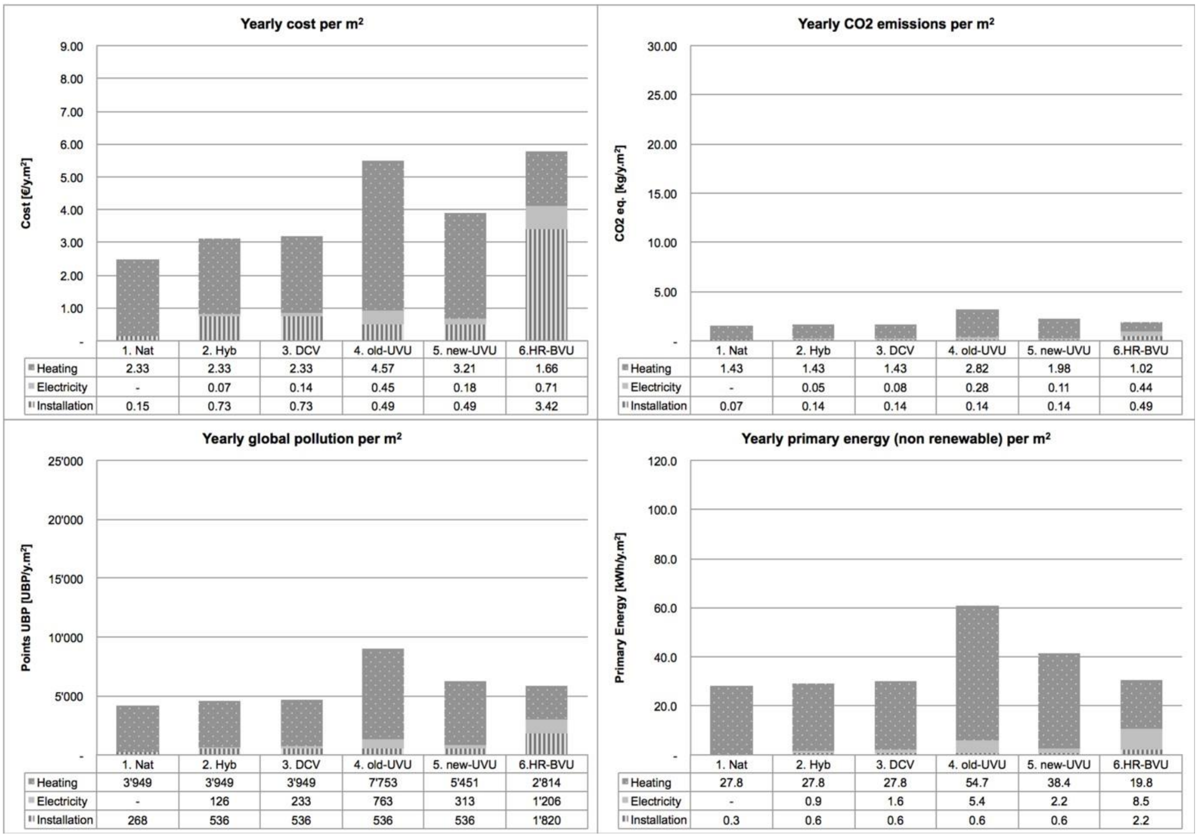
Yearly cost per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	3.41	3.41	3.41	6.69	4.71	2.43
Electricity	-	0.07	0.14	0.45	0.18	0.71
Installation	0.15	0.73	0.73	0.49	0.49	3.42

Yearly CO2 emissions per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	12.69	12.69	12.69	24.91	17.51	9.04
Electricity	-	0.05	0.08	0.28	0.11	0.44
Installation	0.07	0.14	0.14	0.14	0.14	0.49

Yearly global pollution per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	9'419	9'419	9'419	18'492	13'002	6'711
Electricity	-	126	233	763	313	1'206
Installation	268	536	536	536	536	1'820

Yearly primary energy (non renewable) per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	52.0	52.0	52.0	102.1	71.8	37.0
Electricity	-	0.9	1.6	5.4	2.2	8.5
Installation	0.3	0.6	0.6	0.6	0.6	2.2

*Figure 2. Results of life cycle cost, CO2, global solution and primary energy of 6 ventilation systems, applied on a building, considered to be heated by an **oil heating system, η 0.85**.*



Yearly cost per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	2.33	2.33	2.33	4.57	3.21	1.66
Electricity	-	0.07	0.14	0.45	0.18	0.71
Installation	0.15	0.73	0.73	0.49	0.49	3.42

Yearly CO2 emissions per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	1.43	1.43	1.43	2.82	1.98	1.02
Electricity	-	0.05	0.08	0.28	0.11	0.44
Installation	0.07	0.14	0.14	0.14	0.14	0.49

Yearly global pollution per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	3'949	3'949	3'949	7'753	5'451	2'814
Electricity	-	126	233	763	313	1'206
Installation	268	536	536	536	536	1'820

Yearly primary energy (non renewable) per m²	1. Nat	2. Hyb	3. DCV	4. old-UVU	5. new-UVU	6.HR-BVU
Heating	27.8	27.8	27.8	54.7	38.4	19.8
Electricity	-	0.9	1.6	5.4	2.2	8.5
Installation	0.3	0.6	0.6	0.6	0.6	2.2

*Figure 3. Results of life cycle cost, CO2, global solution and primary energy of 6 ventilation systems, applied on a building, considered to be heated by **heat pump COP 3.5**.*

On figures 2 and 3 we can observe several interesting results not coinciding with common belief, part of it adopted by some national regulations.

- Optimisation of ventilation offers a significant energy saving potential. Especially case 5 shows that optimisation of existing oversized ventilation units has much higher environmental impacts due to high energy consumption for heating. Independently of the heating energy source, cases 4 and 5 present, always, higher environmental and primary energy impacts.

- Demand control ventilation is a very good optimisation strategy. It combines low embodied energy, with low electricity consumption and reduced heating energy demand, presenting global performance comparable to bidirectional ventilation units with high performance heat recovery. With high performance energy sources (like heat pumps), it presents a global performance even better than some heat recovery systems.

- The most penalising factor for bidirectional ventilation units is electricity consumption. With large buildings, air transport distances are long and pressure drop high, implying high SPI and electricity consumption. As we see on figure 3, when the recovered heat contains low embodied energy, heat recovery is not the best optimisation strategy.

- Bidirectional ventilation system life cycle cost is always higher, even than a correctly unidirectional ventilation unit without adaptive control. It is cheaper only than an oversized old system, and this only when the building is heated with fossil fuel.

- Even when heat recovery unit is preferred in terms of primary energy, the global environmental impact is always higher than the demand control ventilation. System environmental impact is high, higher even than environmental impact of electricity consumption. Low system environmental impact of demand control natural ventilation, combined with no electricity consumption, makes this technique globally the cheapest and environmentally most friendly ventilation technique for the medium European climate (Central Europe).

- Life cycle analysis is coherent with ecodesign delegate EU regulations 1253/2014 and 1254/2014, especially for fossil fuel heat production. As the perimeter of these regulations is limited to ventilation system, they cannot make the difference between a building heated with oil and another heated with a high performance heat pump, with wood, or with district heating / waste source. But as we can see on the results, heating system source is a capital parameter to take into account in the optimisation strategy.

- Demand controlled natural ventilation present very good environmental performance, but someone must take into account the risks of backflow from the roof in some weather conditions. Ventilation quality is not the same with the other compared systems.

All these comparisons are first of all comparisons of the selected parameters. Selected parameters are never neutral. For these study we have selected systems, which are the most common to the Swiss refurbishment reality in 2015, with parameters prescribed by regulations and confirmed - validated by occasional measurements on the real building.

A key value for the results of figures 2 and 3 is q_{system}. Dimensioning of the nominal airflow rate was done according to the Swiss regulations, prescribing 30 m^3/h per bedroom and living room. In reality the building of case 6, table 2 was functioning before refurbishment with an oversized unidirectional ventilation unit extracting air from kitchen and bathrooms as in case 4. During the refurbishment project a study compared 3 alternative systems, those of case 3-DCV – demand control ventilation, case 5-new-UVU – resized unidirectional ventilation unit, working on high speed during working hours and low speed during night, and case 6-HR-BVU, regenerating heat recovery with bidirectional ventilation unit. All of them dimensioned with maximum airflow rate according to Swiss regulations (30 m^3/h per room – 0.95 m^3/m^2h).

The design team have chosen bidirectional heat recovery ventilation, because of two reasons. The owner wished very high-energy-standard refurbishment. But also, because of regulations, prescribing a heat recovery for ventilation in new buildings and refurbishment projects. The only way to avoid a bidirectional ventilation unit with heat recovery was to install a heat pump on the exhaust air to recover ventilation losses and preheat hot water, but as the building will be connected in the near future to a district heating with wood energy source, preheating of hot water with a heat pump was considered a "double investment".

q_{system} of case 6 (0.31 m^3/m^2h) is calculated by multiplying the dimensioning airflow rate with the mean heating season recovery efficiency (0.7). Measurements on a whole heating system validated exactly this hypothesis. q_{system} of adaptive ventilation – cases 1, 2, 3 (0.70 m^3/m^2h) is considered based on product specifications certified by CSTB (CSTB 2013), taking the most pessimistic value (0.5-0.7 m^3/m^2h). This value is based also on a diploma work that simulated and validated with measurements on the building of case 1 (Ferrini 2008). Case 4 was measured and case 5 is the dimensioning max airflow rate with additional flow rate of 26% infiltration penalty due to lack of control.

The data for environmental impacts for the ventilation systems come from KBOB database (KBOB 2014). System costs come from EPIQR method cost database and fuel cost from the federal statistics office, mean price 2010-2014.

3 THEORETICAL AND REAL PERFORMANCES

A whole year project observed the behaviour of case 2 and case 6 refurbished buildings to compare difference between projected and real energy consumption presented on figure 4.

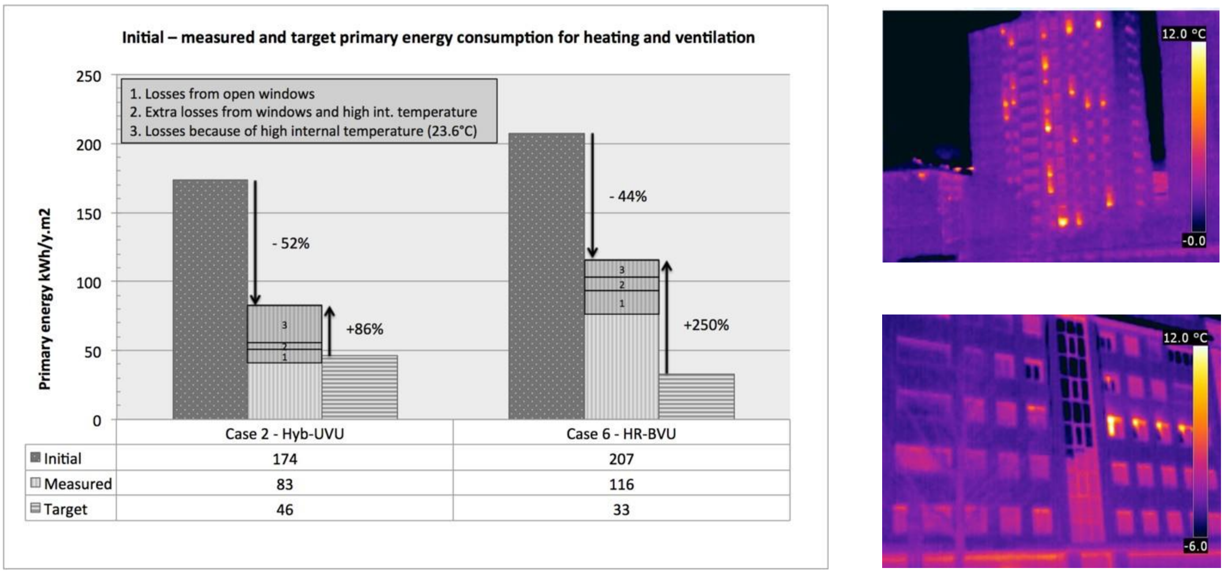


Figure 4. Initial (before refurbishment) primary energy consumption for heating and ventilation compared to measured and target energy consumption. As we see, for case 2 a reduction of 52% does not meet the objectives and energy consumption is 86% higher than the planned target. For case 6, energy consumption is 250% higher than planned. User behaviour (1, 2, 3) explain a big part of extra consumption.

As we see from figure 4, real consumption is much higher than the expected one although there is a significant reduction. The mean internal temperature during heating season was measured to 23.6°C causing 47% extra losses to building 2 and 23% to building 6. Observation of the buildings with a thermocamera showed a mean 10% of windows open for case 2 (extra 0.32 $m^3/h.m^2$ of air flow rate) and 18% of windows open for case 6 (extra 0.62 $m^3/h.m^2$ of air flow rate) provoking respectively 17% and 31% heat losses. If we take into account that apartments are overheated to 23.6°C, heat losses from the windows are even higher. These extra losses due to user behaviour are not depended on the ventilation system, although the extra window ventilation is added to the system airflow. However, the small differences between the global performance of demand control ventilation and heat recovery bidirectional ventilation are insignificant near to the real airflow in the buildings.

Ventilation system quality was almost perfect in both cases, planned and real airflow rates and efficiencies were very near, electricity energy consumption was near to planned and the building envelope insulation was correctly realised. Air quality in both building was controlled during the heating season and was also conform to the objectives with CO2 concentration < 1000 ppm for case 2 and < 1500 ppm for case 6. Performance differences are only due to occupant behaviour and for case 6 due to the heating system problems.

4 CONCLUSIONS

Chapter 2.2 showed that EU regulation is a good tool to choose a ventilation system. The comparison of 6 common ventilation systems in apartment buildings show that it is difficult to get class A, for ventilation systems with specific power input higher than 0.32 W/(m^3.h). Demand control ventilation systems have slightly higher specific energy consumption, but very near to the compared heat recovery system are classified both to class B.

A life cycle environmental assessment in chapter 2.2 give similar results with the EU regulation for buildings heated with standard oil heating systems, but it shows that the optimisation strategy for ventilation depends also from the heating energy source. Heat recovery has better energy performances with oil fuel. However with low primary energy sources, such as a heat pump of COP 3.5, demand control ventilation – DCV benefits (low embodied energy and low electricity consumption) are more environmentally and energy effective than heat recovery benefits. Cost effectiveness of DCV is undeniable in all cases.

Chapter 3 showed that user behaviour with windows, although independent from ventilation system characteristics, has much more effect on building energy consumption than system differences. These losses are not taken into account in several energy regulations.

The conclusions of this analysis show that performance oriented regulations (like the EU regulations) are much more environmentally and energy sound than system oriented regulations (like Swiss regulations), that in sometimes they do not promote the best solution and sometimes prohibit good solutions.

5 REFERENCES

EU 1254 (2014), EU delegate regulation, implementing directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for ventilation units, Brussels

EU 1253 (2014) EU delegate regulation, implementing directive 2009/125/EC of the EU Parliament and of the Council with regard to ecodesign requirements for ventilation units, Brussels, 2014

SN 520 380/1(2009) L’énergie thermique dans le bâtiment (thermal energy in buildings), SIA, Zurich.

CSTB, (2013) Avis technique 14-12-1763, système de ventilation mécanique basse pression hygroréglable, Paris

Berthin S., Savin J.L. and Jardinier M. (2005) Assessment of improvements brought by humidity sensitive and hybrid ventilation / HR-VENT project, 26th AIVC conference

Ferrini Gian-Luca (2008) Caractérisation de la performance énergétique d'un système de ventilation simple-flux hygroréglable dans le label Minergie, EIVD, Yverdon

Flourentzou, F., Irwin, Savin, J.-L., Ferrini G-L. (2011), Insitu Measurements to evaluate the real energy savings of humidity sensitive ventilation in Minergie Buildings, Cisbat conference, Lausanne.

KBOB, 2014, Données des écobilans dans la construction, https://www.kbob.admin.ch, 1-2014.pdf

ENERGY AND IAQ FRIENDLY VARIABLE VENTILATION RATES, ACCORDING WITH THE PROPOSED INDOOR AIR QUALITY REGULATIONS INCLUDED IN THE SPANISH BUILDING CODE.

García, Sonia* [1], Linares, Pilar[1]

1 Eduardo Torroja Institute for construction sciences-CSIC
4, Serrano Galvache St.
Madrid, Spain

**Corresponding author: sgarcia@ietcc.csic.es*

ABSTRACT

The Spanish Building Code (BC) regulates indoor air quality (IAQ) requirements in dwellings by establishing threshold continuous flow rates according to the occupancy, use of the rooms and their usable area. The implementation of this threshold flow allow adequate IAQ.
A revised IAQ requirement have been proposed. These new requirement quantifies the IAQ as a function of CO_2 concentration which means an non continuous flow rates ventilation systems will be able to be used.
In order to reduce the global ventilation rate and save energy related to heating and cooling, research have been conducted to obtain nom continuous flow rates that full fill propose IAQ requirement for some case studies. In addition to this CO2 concentration has been using the continuous ventilation rates including in the current IAQ regulations.
This research has been developed using the CONTAM simulation tool, BD proposed hypothesis and owns hypothesis too. Results show different non continues ventilation rates which are friendly with energy saving and the proposed IAQ levels.

KEYWORDS

Ventilation, IAQ, regulations, energy saving

1 INTRODUCTION

The Spanish Building Code (BC) regulate in the Hygiene, health and the environment Basic Document (acronym in Spanish: DB HS3) the indoor air quality (IAQ) requirements in dwellings. The Building Code sets the minimum continuous flow according to the occupancy, use of the locals and their usability area. The implementation of this threshold flow allow adequate air quality.

The Eduardo Torroja Institute for construction sciences, together with Ministerio de Fomento, has proposed a revised version for the IAQ regulations for dwellings. The proposal quantify the air quality in function of CO_2 concentration and allows non continuous flow.
In order to reduce the ventilation global rate and save climate energy, we simulate and analyse non continuous air flow and the consequence in the IAQ according to the proposal.

2 CURRENT AND PROPOSED IAQ REQUIREMENT

The current IAQ requirement establishes the minimum continuous ventilation rates per habitable rooms. (See Table 1).

Table 1. Building Code minimum ventilation rates

Rooms	**Per person**	**Per usable floor area m^2**	**Per room**
Bedrooms	5 l/s		
Living and dining rooms	3 l/s		
WC and bathrooms			15 l/s
Kitchens		2l/s	

The proposed IAQ requirement uses the CO_2 concentration as an IAQ indicator. The required CO_2 concentration is limited in two ways:

- 900 ppm maximum yearly average;
- 500000 ppm per hour maximum yearly accumulated above 1600 ppm.

Those are design conditions and must be full fill according to other conditions such as occupancy scenarios, CO_2 production rate, yearly average outdoor CO_2 concentration, etc.

3 SELECTION OF CASE STUDIES

The dwelling case studies that have been chosen are the most representative ones in Spain according to Instituto Nacional de Estadística (National Statistics Institute. Population and Housing Census 2011). They are all real dwellings. (See Table 2).

Table 2. Dwellings case study

Dwelling	**Kind and composition of dwelling**	**Number of occupants**
3B	Flat: Living + Kitchen + 3 Bedroom + 2 Bathroom; 71 m^2	4
3C	Flat: Living + Kitchen + 3 Bedrooms + 2 Bathrooms; 80 m^2	4
3D	Flat: Living + Kitchen + 3 Bedrooms + 2 Bathrooms;74 m^2	4

The National Statistics Institute provides population recount and knowledge of its structure. According with variables studied, the dwellings and homes more common in Spain have 3 bedrooms, 1 to 4 occupants, and between 60 and 90 m2. The chosen case study are in this group. The included results correspond to 4 occupants because this cases represent the worse results. Figures 1 to 3 represents the national values for the last census.
Dwelling: Structurally separate and independent premise that due to how it was built, re-built, transformed or adapted, it is conceived to be inhabited by persons or it constitutes the regular residence of a person in the moment the census takes place.. (Total number: 25.208.623)

Home (household): Group of persons (one or more) that reside in the same dwelling when the census takes place. (Total number: 18.083.692)

Figure 1. Dwellings vs rooms per number of occupants.

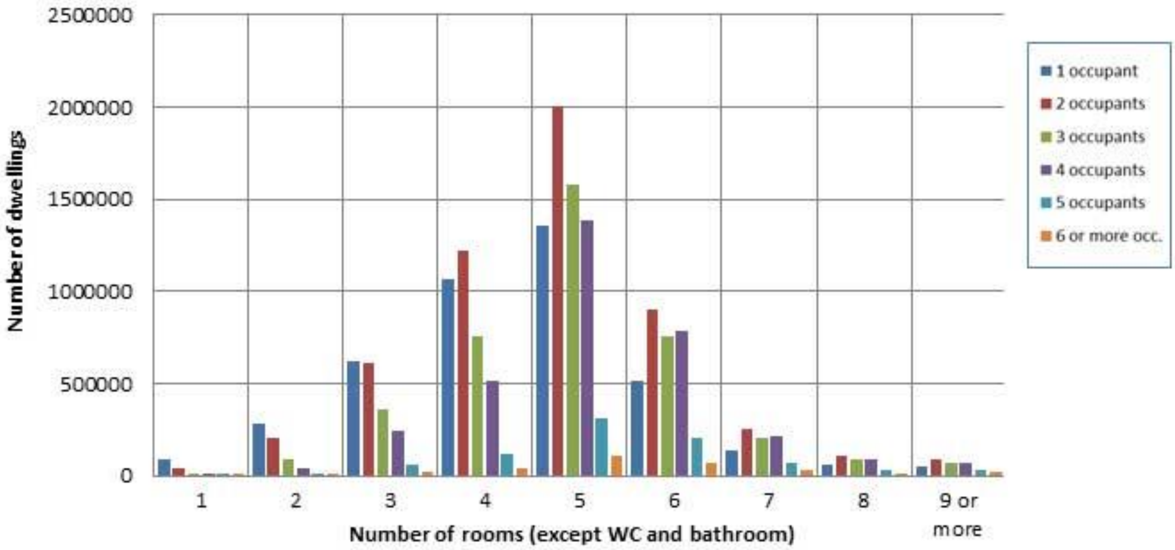


Figure 2. Percentage of dwellings vs usable floor area.

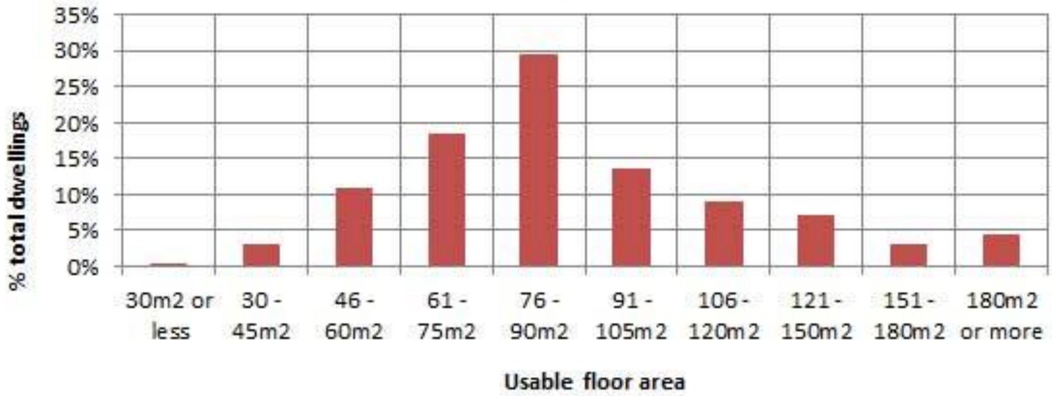
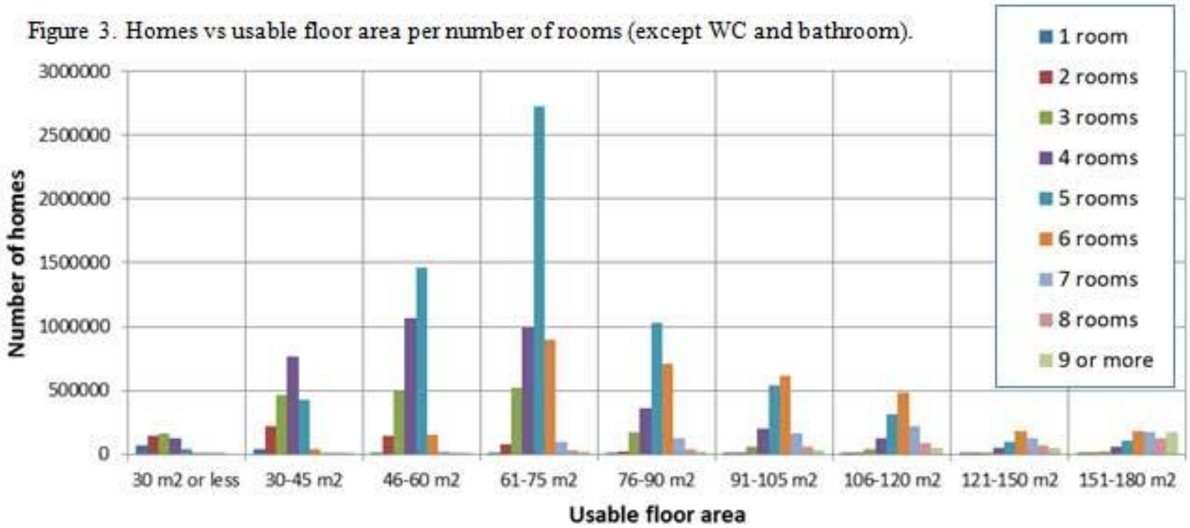


Figure 3. Homes vs usable floor area per number of rooms (except WC and bathroom).



4 PARAMETERS

The most relevant parameters that have been taken in to account are:

- CO_2 generation: 19 l/s (light to moderate intensity activities), 12 l/s (sleeping).
- Occupancy schedule: according to proposed IAQ.
- Occupancy ratio: according with *Population and Housing Census 2011*.
- Dwellings design: according with *Population and Housing Census 2011*.
- Software: CONTAM *Multizone Airflow and Contaminant Transport Analysis Software*.

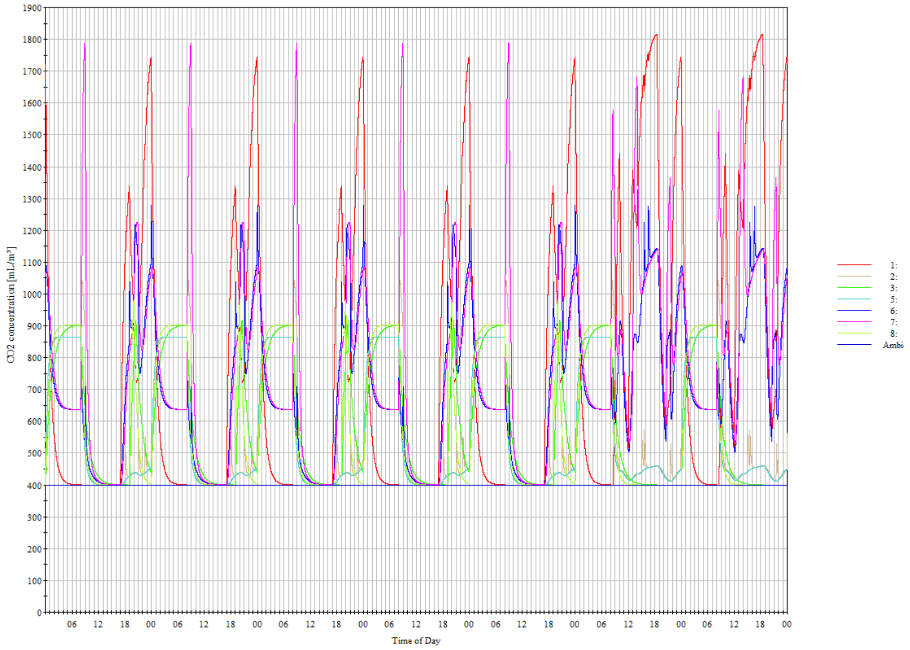
5 CURRENT IAQ REQUIREMENT RESULTS

The current regulation provide good results in general. Only shows a high CO_2 accumulated value when the kitchen usable floor area is too small. In this case, the accumulated above 1600 ppm per year exceed the 1600 ppm. The highest CO_2 concentration are in the in the kitchen, because of the expected high occupants activity (high CO_2 generation) and the expected high number of people present in the kitchen at the same time. Usually living room and kitchen have the high average. In the bedrooms, nigh average of 900ppm become a yearly average of 550-600 ppm.
Table 3 shows the obtained CO_2 results for the case studies with a continuous ventilation rates that provide fulfilment of the current IAQ requirement.

Table 3. Results with continuous ventilation rates that provide fulfilment of the requirement.

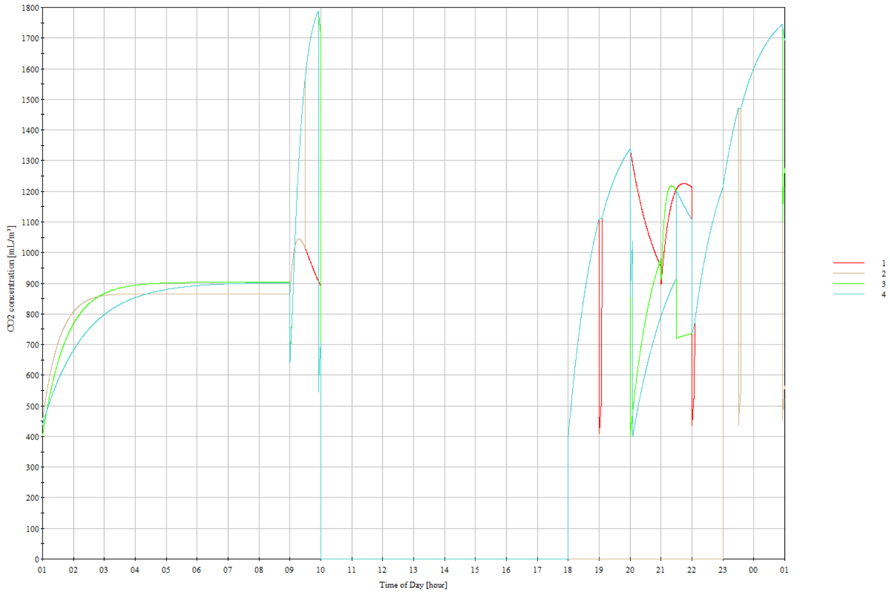
	Occupants	**Kitchen usable floor area**	**WC and bathrooms**	**CO_2 yearly average per room (highest value)**	**Accumulated above 1600 ppm per year**	**Continuous ventilation rate**	
Dwelling	**Number of**	**m^2**	**Number of**	**ppm**	**ppm/year**	**l/s**	**h^{-1}**
3B	4	5.87	2	Kitchen 765	Living 1375332	42	0.7870
3C	4	8.47	2	Kitchen 756	Cocina 60084	47	0.7805
3D	4	10.65	2	Kitchen 660	0	51	0.9180

Figure 4a. Dwelling 3B: CO_2 concentration evolution per room during a week.



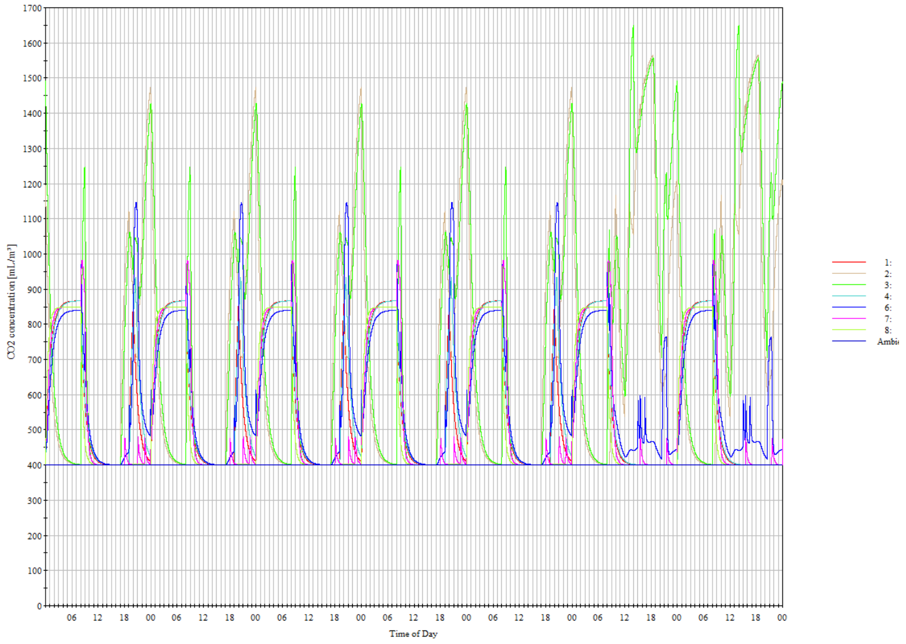
1: Living
2: Bathroom (en-suite)
3:Bedroom 3
5: Double principal bedroom (suite)
6: Bathrrom 2
7: Kithchen
8: Bedroom 2
Last one: Ambient

Figure 4b. Dwelling 3B: Occupants exposure during a working day in the dwelling



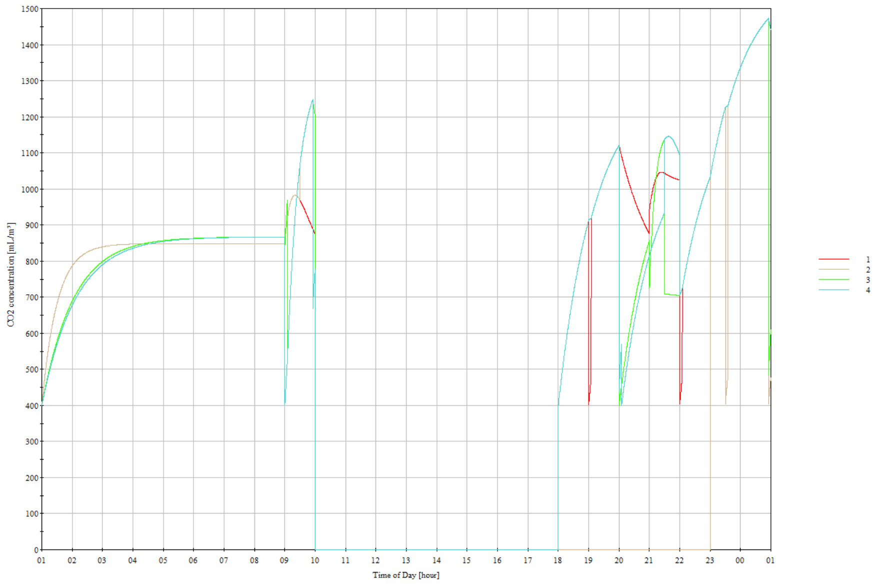
1: occupant 1
2: occupant 2
3: occupant 3
4: occupant 4

Figure 5a. Dwelling 3C: CO_2 concentration evolution per room during a week.



1: Bedroom 2 2: Living 3: Kithchen
4: Bedroom 3 6: Bathrrom 2 7: Bathroom (en-suite)
8: Double bedroom (suite) Last one: Ambient

Figure 5b. Dwelling 3C: Occupants exposure during a working day.



1: occupant 1 2: occupant 2 3: occupant 3 4: occupant 4

6 PROPOSE IAQ REQUIREMENT RESULTS

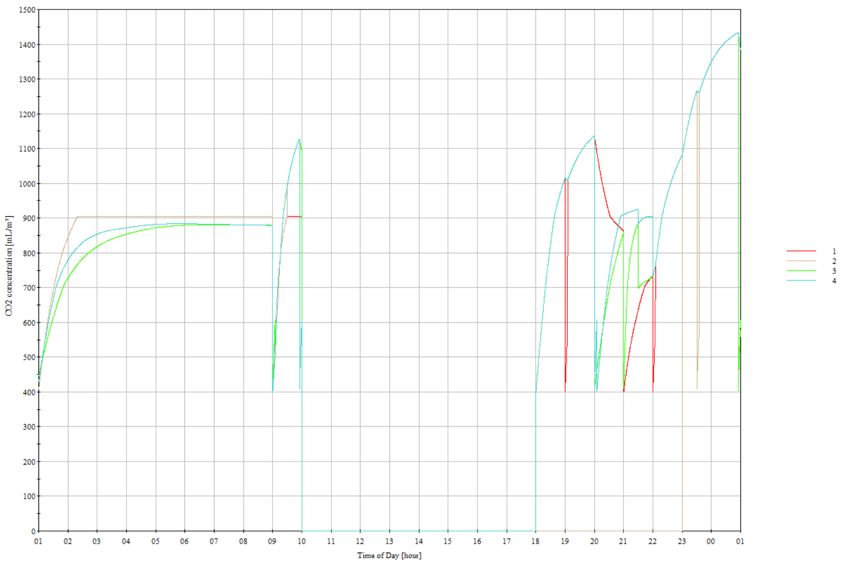
Non continuous ventilation rate allow reduces the total ventilation air flow because a reduction of ventilation during non occupancy periods also increase the ventilation ratio according with occupancy increase. The proposed IAQ open the door to non continuous ventilation flows and smaller global ventilation rates.

The obtained results show similar concentrations and occupant exposure to those ones obtained with the continuous flow establish in the current regulation.
A reduction of flow during non occupancy periods reduces the global rate. However, non ventilation during non occupancy periods is non recommended because there are other contaminants come from furniture and construction materials. There fore the flow during non occupancy periods have been set in a minimum valor. Simulations have been carried out with diferreent minimum valor of ventilation flow rate for the non occupancy periods: such as 20% of the maximum ventilation rates and 2 l/s.
Table 2 and figure 6 shows results for non continuous ventilation rates that provide fulfilment of the proposed requirement, with a 20% of the maximum ventilation flow during non occupancy periods.
It is easy maintain the accumulated above 1600 ppm per year bellow 0 in a simulation tool bases on CO2 concentration. But in a different control system, would be necessary have this item to control the punctual high concentrations.

Table 2. Non continuous ventilation rates that provide fulfilment of the requirement. 20% ventilation rate.

	Occupants	**Kitche n usable floor area**	**WC and bathrooms**	**CO_2 yearly average per room (highest value)**	**Accumulated above 1600 ppm per year**	**Global ventilation rate**	
Dwelling	**Number of**	**m^2**	**Number of**	**ppm**	**ppm/year**	**l/s**	**h^{-1}**
3B	4	5.87	2	Living 697	0	37.8	0.71
3C	4	8.47	2	Living 688	0	38.5	0.64
3D	4	10.65	2	Living 700	0	38.1	0.69

Figure 6. Dwelling 3D: Occupants exposure during a working day.



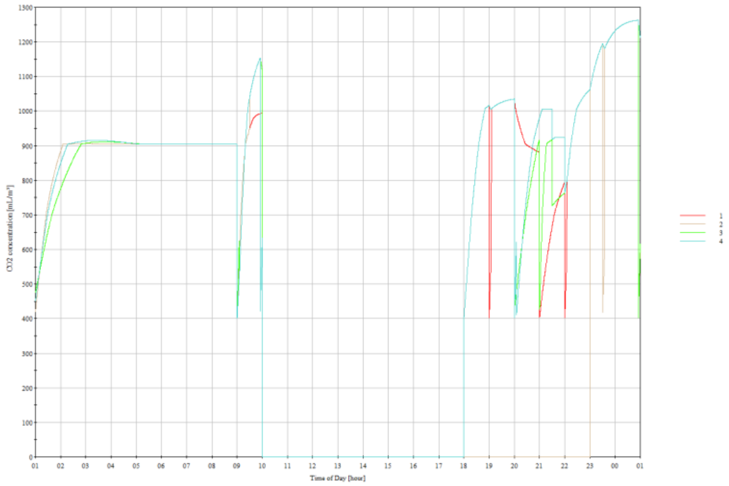
1: occupant 1 2: occupant 2 3: occupant 3 4: occupant 4

Table 3 and figure 7 shows results for non continuous ventilation rates that provide fulfilment of the proposed IAQ requirement too. In this case, the ventilation per room during non occupancy periods are always 2l/s.

Table 3. Other variable ventilation rates possibility: 2l/s not occupancy ventilation flow.

	Occupants	**Kitchen usable floor area**	**WC and bathrooms**	**CO_2 yearly average per room (highest value)**	**Accumulated above 1600 ppm per year**	**Global ventilation rate**	
Dwelling	**Number of**	**m^2**	**Number of**	**ppm**	**ppm/year**	**l/s**	**h^{-1}**
3B	4	5.87	2	Living 683	0	29.64	0.55
3C	4	8.47	2	Living 690	0	30.45	0.51
3D	4	10.65	2	Living 688	0	30.16	0.55

Figure 7. Dwelling 3D. Working day occupants exposure. 2l/s non occupancy ventilation flow.



It is possible optimize the ventilation rates in order to minimize the global rate according to the proposal IAQ by providen the maximum flow when 1400ppm CO_2 concentration is reach. The maximum ventilation when concentration reach 1400pppm is the maximum concentration according to 900ppm average in the whole time. The results show:

- the global ventilation rate is smaller;
- the IAQ is fulfill the DB HS3 requirements;
- the occupant exposure average (time at home) is high (see figure 8), compared with figure 6 and 7, but the locals concentration averages maintain under 900ppm (see figure 9).

Table 4. Non ventilation flow during non occupancy periods

	Occupants	**Kitchen usable floor area**	**WC and bathrooms**	**CO_2 yearly average per room (highest value)**	**Accumulated above 1600 ppm per year**	**Global ventilation rate**	
Dwelling	**Number of**	**m^2**	**Number of**	**ppm**	**ppm/year**	**l/s**	**h^{-1}**
3B	4	5.87	2	Bedroom3 885	0	15.13	**0.28**
3C	4	8.47	2	Bedroom3 893	0	15.37	**0.26**
3D	4	10.65	2	Bedroom2 856	0	15.13	**0.27**

Figure 8. Dwelling 3D. Working day occupants exposure. No ventilation flow during non occupancy periods and focus only in reduce global ventilation rates.

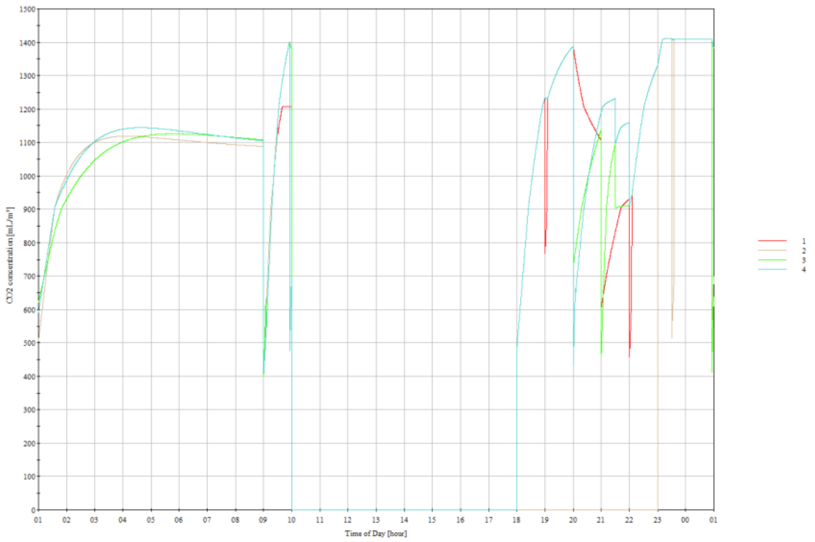
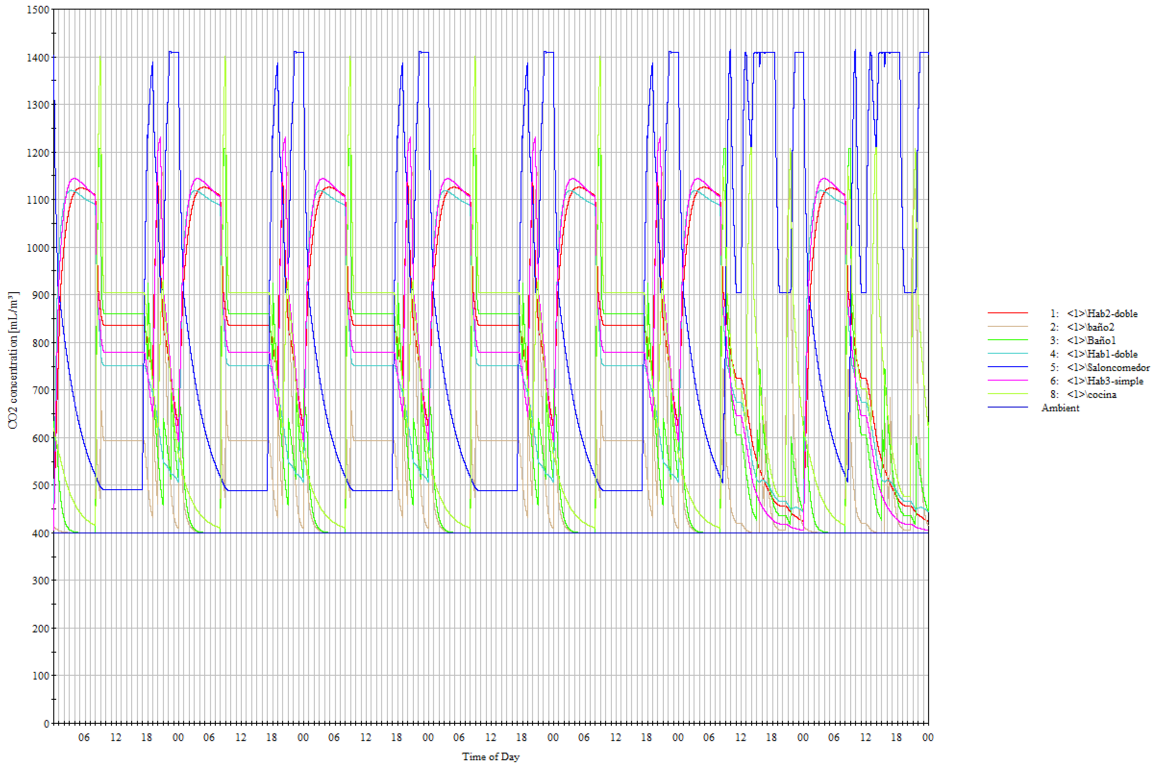


Figure 9. Dwelling 3D CO_2 concentration evolution per room during a week.



1: Bedroom 2   2: Bathroom 2   3: Bathroom 1 (in suite)   4: Double principal bedroom (suite)
5: Living   6: Bedroom 3   8: Kitchen   Last one: Ambient

4 CONCLUSIONS

This study shows how it should be possible, for one of the most common dwellings distribution in Spain, to reduce the ventilation rates. This is possible using non continuous

ventilation flow rates, according with the proposed requerements for the IAQ regulations, saving energy for heating and cooling without impacting air quality.
However, not any ventilation rates provide adecuates IAQ levels. Assessment much be carried out in each case to optimize this rates, taking in to account providing minimum ventilation rates for non occupancy periods is highly recommended.
Some possibilities according with the proposed IAQ regulation decreased the IAQ respect the continuous flow.
The lower ventilation rates with a good IAQ could be obtained if the ventilation system can change the ventilation flow according with the number of occupants and the real needs.

5 REFERENCES

Linares, P., García, S. et al (2014). Proposed change in Spanish regulations relating to indoor air quality with the aim of reducing energy consumption of ventilation systems. PROCEEDINGS of 35th AIVC Conference, 4th TightVent Conference and 2nd venticool Conference: Ventilation and airtightness in transforming the building stock to high performance. Poznań (Poland).

*Código Técnico de la Edificación (*Building Code*). Ministerio de Fomento*. On line.
http://www.codigotecnico.org/web/recursos/documentos/ [Consulted 18/06/2015]

Population and Housing Census 2011. National Statistics Institute. On line.
http://www.ine.es/en/censos2011_datos/cen11_datos_inicio_en.htm [Consulted 11/04/2015]

UNE-CEN/TR 14788:2007 In *Ventilation for buildings - Design and dimensioning of residential ventilation systems.*

CONTAM *Multizone Airflow and Contaminant Transport Analysis Software*. National Institute of Standards and Technology (NIST)
http://www.bfrl.nist.gov/IAQanalysis/index.htm

ENERGY SAVING AS A CONSEQUENCE OF THE PROPOSED CHANGE IN SPANISH REGULATIONS RELATING TO INDOOR AIR QUALITY.

Linares, Pilar[*1], García, Sonia [1], Larrumbide, Enrique [1], Tenorio, José A. [1]

1 Eduardo Torroja Institute for construction sciences-CSIC
4, Serrano Galvache St.
Madrid, Spain

**Corresponding author: plinares@ietcc.csic.es*

ABSTRACT

Recently research at the Eduardo Torroja Institute for construction sciences proposes a new wording for the IAQ regulations for dwellings included in the Spanish Building Code.

The main goal of the earlier research was to adapt required ventilation rates to real needs to achieve a reduction of ventilation rates and energy demand with no negative impact on indoor air quality.

This goal will be achieved by firstly setting maximum values of CO_2 concentration as an indicator of air quality (allowing the use of variable ventilation rate systems), and, secondly, by deriving a table summarizing the minimum continuous ventilation rate threshold values that provide this air quality. As a result, it will be easier to optimize energy demand using either variable flow or lower continuous flow ventilation systems.

For this research, IAQ simulations and energy saving assessment have been performed. IAQ simulations have been developed to optimize the continuous ventilation flows. In this paper the results of the energy saving assessment are presented for different types of dwellings, according to the number of bedrooms and bathrooms and for two distinct Spanish climatic zones. Results show how it should be possible to achieve target CO_2 concentration values in addition to conserving energy.

KEYWORDS

Ventilation, IAQ, regulations, energy saving

1 INTRODUCTION

The current Spanish Building Code came into force in 2006, including IAQ provisions for dwellings. The provision of the code at that time represented a big regulatory step. However, these provisions are not as performance-oriented as was initially anticipated, requiring minimum rates for delivery-to and extraction-from the habitable rooms. These rates are not adjustable, so ventilation systems based on variable ventilation rates are not normally deemed acceptable, unless a comprehensive statement of compliance is provided, justifying the proposed ventilation solution.

In 2010, with the adoption of the recast EPBD, EU Member States faced new challenges. The revised goal is to increase the level of performance - by 2020 - to nearly-zero energy buildings. In order to achieve this goal in Spain, a deep review of the energy requirements was performed in 2013, leading to increased energy efficiency of buildings; though this was deemed insufficient as energy efficiency in buildings is affected as well by ventilation systems.

Therefore the use of variable ventilation systems is desirable, as it would almost certainly produce a reduction of the global ventilation rate and, consequently, a reduction of the heating and cooling energy demand, while maintaining a good level of air quality.

As a consequence, research is now in progress to modify the Spanish regulations to accommodate the use of more efficient systems which are capable of adapting required ventilation rates to real needs. This would mean a reduction in ventilation rates and energy demand but without impact on indoor air quality.

This paper presents the results of the ongoing research that shows how the IAQ requirement can be modified keeping acceptable IAQ values and reducing energy demand.

2 RELEVANCE OF ENERGY DEMAND CAUSED BY VENTILATION

In the European Union it is commonly understood that buildings are responsible for 40% of final energy consumption and 36% of carbon dioxide emissions. For this reason, a reduction of energy consumption and increased use in the building sector of energy from renewable sources can be an important part of measures to reduce energy dependence of the Union and emissions of greenhouse gases. Member States should take adequate measures to ensure that minimum energy performance requirements are set for buildings in order to achieve cost optimal levels.

In Spain, the construction sector has a significant impact on both the country's overall energy consumption and emissions of greenhouse gases. Housing and tertiary buildings represent 26% of global energy consumption, residential sector accounting for 17% of total final consumption. The estimated emissions for each household are more than one ton. This situation is characterized by high external energy dependence, which is close to 80% and well above the European average of 54%.

It is important to highlight the significant technical advances that have occurred in the Spanish building sector since the enforcement in 2006 of the Spanish Building Code (and its energy saving requirements), and more recently in 2013 the approval of the reviewed energy requirements.

Energy demand related to ventilation losses and lack of airtightness is significant. Loads related to air renovation in dwellings can be estimated at 20-40% of total loads produced by heating and cooling. Therefore it is necessary to improve IAQ regulations to reduce the related energy demand.

3 CURRENT IAQ REQUIREMENT

The current IAQ requirement establishes minimum ventilation rates (see Table 1) for delivery-to and extraction-from habitable rooms. These rates have to be provided in a continuous way.

Table 1. Minimum ventilation rates

Rooms	Per person	Per usable floor area m^2	Per room
Bedrooms	5 l/s		
Living and dining rooms	3 l/s		
WC and bathrooms			15 l/s
Kitchens		2 l/s	

4 PROPOSED IAQ REQUIREMENT

IAQ level is usually characterized by a maximum level of pollutants that may affect people´s health and comfort and which could be achieved by different ventilation systems.

However, common pollutants are not easy to assess, so an indicator is commonly taken to represent the state of the rest of the pollutants. Among the possible pollutants that are commonly produced indoors, CO_2 is the most commonplace and closest related to human activity. Despite the fact that CO_2 does not entail any health risk in the commonly encountered concentrations, CO_2 is nevertheless a reliable indicator of ventilation rate, for which reason it is the most common indicator used in regulations and guides.

The required CO_2 concentration is limited in two ways:

- 900 ppm maximum yearly average;

- 500000 ppm per hour maximum yearly accumulated above 1600 ppm. This parameter shows the relationship between the CO_2 concentrations reached above a limit value and their duration over a year. It can be calculated as the sum of the areas (in ppm•h) within the representation of the CO_2 concentration as a time function and the limit value.

These required concentration levels shall be achieved under certain design conditions (such as occupancy scenarios, CO_2 production rate, yearly average outdoor CO_2 concentration, etc.), that should be set in the regulation. That is, it is a “design performance” because it can only be measurable *in situ* under these conditions.

5 PROPOSED VERIFICATION METHOD

Fulfilment of the requirement is achieved through expert methods (such as specialized software), but it is convenient for the regulations to provide a simplified verification method for designers. This simplified method shall be easy to use by non-expert practitioners and will consist of a table with different ventilation rates (continuous) that will provide fulfilment of the requirement for different dwelling types.

The dwelling case studies that have been chosen for the assessment have been classified taking into account their bedroom and bathroom counts (See Table 2). They are real dwellings representative of the ones that have been recently built (the Spanish population and dwelling census has been used).

Table 2. Dwellings case study

Kind and composition of dwelling	Case study
Flat: Living/Kitchen+1 Bedroom+1 Bathroom	1
Flat: Living+Kitchen+2 Bedrooms+2 Bathrooms	2
Flat: Living+Kitchen+3 Bedrooms+2 Bathrooms	3
Flat: Living+Kitchen+4 Bedrooms+2 Bathrooms	4
Terraced house: Living+Kitchen+4 Bedrooms+2 Bathrooms	5

These ventilation rates are obtained from the results of an analysis of CO_2 concentration with pollutants distribution software CONTAM. CONTAM is a multi-zone airflow and contaminant transport analysis software application developed by the National Institute of Standards and Technology (NIST-US).

This analysis consists of simulating these dwellings (with an occupancy scenario) with different ventilation rates in order to optimize them achieving the required IAQ requirements.

Table 3 shows, for the different dwelling case studies, the lowest continuous ventilation rates that fulfil IAQ requirements.

Table 3. Continuous ventilation rates values

Dwelling case study	Continuous ventilation rate (1) (l/s)	Total whole dwelling continuous ventilation rate (l/s)	Yearly average CO_2 concentration (2) (ppm)	Yearly accumulated over 1.600 ppm (2) (ppm·h)
1	6	12	816	0
2	8	24	812	145860
3	11	33	789	150020
4	11	33	848	247000
5	8	24	826	105560

(1) In kitchen and each bathroom.
(2) The highest value per room in each dwelling.

6 ENERGY DEMAND ASSESSMENT

The energy demand assessment has been carried out using *Herramienta Unificada Lider-Calener* (HULC, June 2015). HULC is a whole building energy simulation program that is used to assess energy demand and consumption. It comprises the earlier software tools LIDER and CALENER, allowing both assessment of energy qualification and fulfilment of the energy saving requirements updated to 2013 changes. It is offered by the *Ministerio de Fomento* and the *Ministerio de Industria, Energía y Turismo - Instituto para la diversificación y ahorro de la energía* (IDAE) (Spain), having been developed by *Grupo de Termotecnia de la Asociación de Investigación y Cooperación Industrial de Andalucía*, (AICIA), *Escuela Técnica Superior de Ingenieros* from *Universidad de Sevilla*, with the collaboration of *Unidad de Calidad en la Construcción* from *Instituto Eduardo Torroja de Ciencias de la Construcción* (IETCC-CSIC).

The assessment has been developed for:

- five dwellings (see table 2),
- ventilation rates from the current IAQ regulations (see table 1) and rates obtained from the proposed requirement (see table 3),
- cooling and heating demand,
- two climatic zones,
- two orientations: North and South, to assess the effect of solar gain, which is quite important in Spain.

In the case of the flats, in order to assess the most exposed case, top floor dwellings were considered.

The Spanish Building Code classifies Spain in 24 climatic zones according to several parameters such as temperature, relative humidity, solar radiation…These climatic zones are characterized by two digits: a letter which refers to winter conditions and a number which refers to summer conditions. The chosen climatic zones are A3 (Cádiz) and D3 (Madrid), both with similarly extreme summer climates but different winter climates, *A* being mild and *D* severe.

Table 4 shows the composition for the building envelope. The width of the insulation and the air permeability of windows have been set to fulfil energy saving regulations in each climatic zone. In the case of the more extreme climate, insulation is thicker and permeability is smaller. In D, air permeability is lower than 27 $m^3/h·m^2$, and in A, lower than 50 $m^3/h·m^2$, using in both cases timber windows double glazed 4/12/4 mm.

Table 4. Composition of buildings ‘envelope

	Composition	**Width (m)**	**Conductivity (W/m·K)**
Façade walls	Solid brick wall	0.115	0.991
	Cement mortar render	0.015	0.55
	Not ventilated cavity	0.050	-
	Rockwool insulation board	0.080/0.060 (1)	0.031
	Gypsum board	0.010	0.25
Roof	Gravel layer	0.150	2.00
	XPS insulation board	0.080/0.060 (1)	0.034
	Felt	0.005	0.05
	Waterproofing membrane	0.005	0.230
	Lightweight mortar	0.100	0.410
	Ceramic pot and beam slab	0.300	-
	Unventilated air space	0.010	0.15
	Gypsum board suspended ceiling	0.010	0.25

(1) The first value corresponds to D3 and the second one to A3.

7 RESULTS

Table 5 shows the results of the energy demand assessment.

Table 5. Energy demand

Dwelling case study	Climatic zone	Orientation	Energy demand					
			Current IAQ requirements [Kw·h/m^2·year]		Proposed IAQ requirements [Kw·h/m^2·year]		Reduction (%)	
			Heating	Cooling	Heating	Cooling	Heating	Cooling
1	D3	North	64.66	12.43	30.62	11.26	53	9
		South	39.25	15.25	9.78	14.4	75	6
	A3	North	14.8	12.42	3.95	11.43	73	8
		South	0.93	15.13	0	14.49	100	4
2	D3	North	68.2	10.6	50.95	9.88	25	7
		South	49.49	13.43	33.09	12.82	33	5
	A3	North	18.14	10.59	11.8	9.92	35	6
		South	4.39	13.27	1.37	12.71	69	4
3	D3	North	54.94	12.87	43.39	12.47	21	3
		South	33.73	18.25	23.51	18.01	30	1
	A3	North	11.89	12.86	8.17	12.49	31	3
		South	1.09	18.18	0.15	17.97	86	1
4	D3	North	51.32	11.07	31.77	10.35	38	7
		South	41.87	13.95	23.22	13.33	45	4
	A3	North	11.36	11.08	5.21	10.45	54	6
		South	4.8	13.93	1.07	13.36	78	4
5	D3	North	53.03	13.8	23.6	12.83	55	7
		South	47.2	15.08	19.01	14.18	60	6
	A3	North	7.34	13.57	0.83	12.65	89	7
		South	4.58	14.79	0.13	13.97	97	6

8 CONCLUSIONS

The results of this study show how it should be possible to achieve target IAQ requirements based on lower continuous ventilation rates than the ones that are currently required, thus saving energy for heating and cooling without impacting air quality.

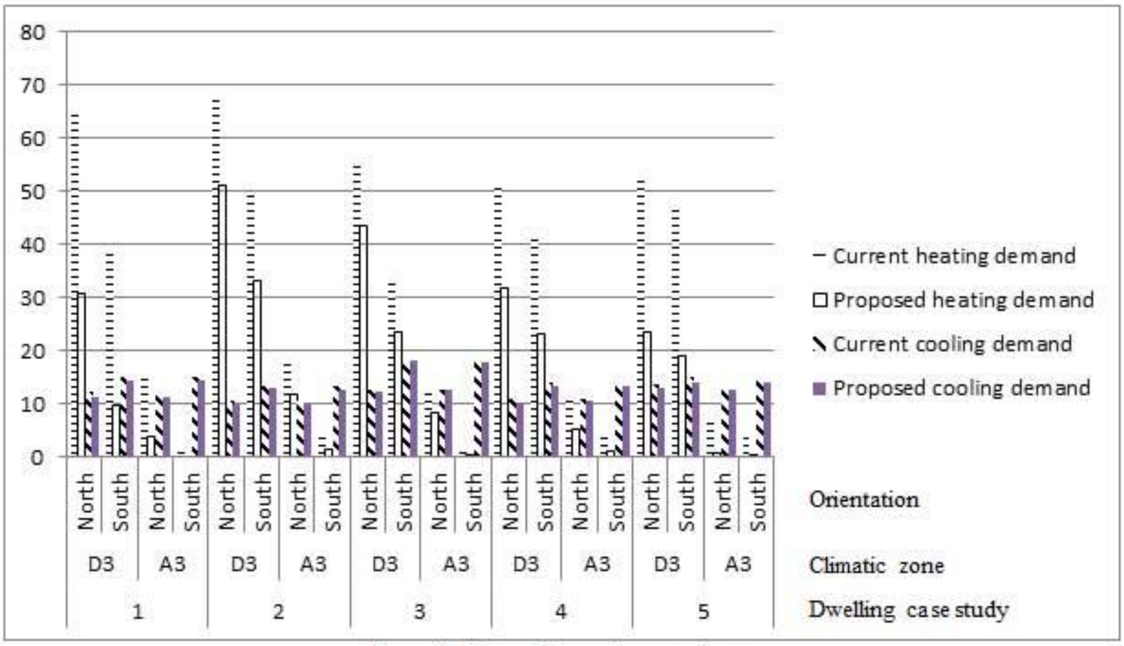


Figure 1. Energy demand comparison

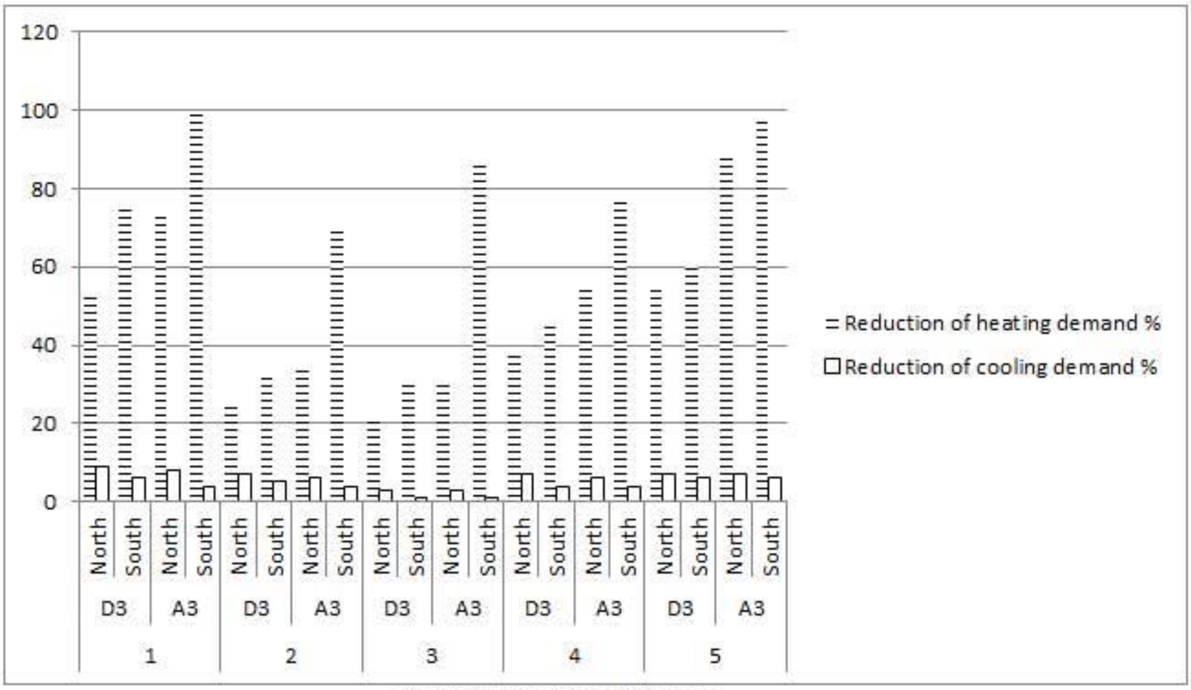


Figure 2. Reduction of demand

In all cases both energy demands get reduced, although the reduction is much bigger for heating ranging from 21 to 100%.

The most sensitive climatic zone and orientation are respectively A3 and South.

9 REFERENCES

Linares, P., García, S. et al (2014). Proposed change in Spanish regulations relating to indoor air quality with the aim of reducing energy consumption of ventilation systems. PROCEEDINGS of 35th AIVC Conference, 4th TightVent Conference and 2nd venticool Conference: Ventilation and airtightness in transforming the building stock to high performance. Poznań (Poland).

*Código Técnico de la Edificación (*Building Code*). Ministerio de Fomento*. On line. http://www.codigotecnico.org/web/recursos/documentos/ [Consulted 18/06/2015]

Gavira, M., Linares, P. et al (2005). *Comportamiento higrotérmico de la envolvente del edificio según el CTE Soluciones alternativas: sistemas de ventilación por caudal variable (*Hygrothermal behaviour of the building envelope according to CTE. Alternative solutions: variable flow ventilation systems*). I Jornadas de investigación, 2*, 739-756.

EPBD. Directive 2010/31/EU of the European parliament and of the council on the energy performance of buildings (recast).

Documento descriptivo climas de referencia (Reference climates descriptive document). *Ministerio de Fomento*. Septiembre 2013. On line. http://www.codigotecnico.org/cte/export/sites/default/web/galerias/archivos/documentosCTE/DB_HE/DOC-DB-HE-0_-_Climas_de_referencia_20130913.pdf [Consulted 18/06/2015]

Plan de Acción de Ahorro y Eficiencia Energética 2011-2020. Technical report. *Instituto para la Diversificación y el Ahorro de Energía* (IDAE).

IMPACT ON THE FORMATION OF MOLD IN THE PERIOD OF SUMMER, THAT INDICATES CHANGE IN EXISTING HOUSING ACH. CLIMATE ZONE D1

Larrumbide, Enrique*[1], Linares, Pilar [1], García, Sonia [1].

1 Eduardo Torroja Institute for construction sciences-CSIC
4, Serrano Galvache St.
Madrid, Spain

**Corresponding author: elarrumbide@ietcc.csic.es*

ABSTRACT

The north of the Iberian Peninsula is characterized by a high level of humidity during the summer. In this climate zone ventilation plays an important role in exercising control over the indoor humidity level of housing and limits the possible formation of surface condensation.

In this article we try to study on a house, the impact that the substitution of a ventilation profile for the summer period set in DB HE 1, 2013 for a constant ventilation profile fulfilling the statutory requirement under the current Hygiene, Health and Environment, Air Quality, DB HS-3, can have on the risk of formation of surface condensations.

The study is set in the climate zone of a coastal province capital with mild summers (zone D1).

KEYWORDS

Ventilation, indoor humidity, condensation.

1 JUSTIFICATION

The Spanish Energy Saving rules, Energy economy and heat retention Document DB HE 2013 climate zone is defined as follows: the area for which the common purpose of calculating the energy demand external stresses are defined. It is identified by a letter, from A to E, corresponding to the severity of winter climate; and a number from 1 to 4, to characterize the severity of summer climate.

The summer period is considered between the months from June to September inclusive; while it is considering winter period, the time between the months from October to May.

In humid climates with mild summers, as may be the area D1, hygrothermal conditions inside the home are highly influenced by external environmental conditions. For these areas, the varying levels of ventilation can cause significant changes in the hygrothermal conditions and therefore, the modification of comfort conditions and the impact on the possible formation of surface condensation.

2 METHODOLOGY

For the study a software tool has been developed. This tool allows scheduling an hourly energy balance, playing indoor temperature and humidity conditions for certain weather conditions.

For outdoor climatic conditions, meteorological climates of the capitals of reference published by the Ministry of Industry, Energy and Tourism of Spain (http://www.minetur.gob.es) have been considered.

To determine the indoor humidity, the criteria set out in the Supporting Document, check limitations of surface and interstitial in the enclosures condensation (DB-HE DA / 2) have been followed.

The climatic zone selected for the study is the area D1, having been regarded as the capital of reference, the city of San Sebastián.

An existing single-family dwelling with unfavorable north orientation has been used as a building type. A value of 0.27 kg / h, has been considered as internal conditions of vapor formation.

For meth metabolic activity of 1.2 and a level of 0.5 clo swaddling, it is considered that the limits of the comfort zone for effective temperatures are between 24.5–19.5º C, while for relative humidity limits are set between 30 and 70%.

In the study two profiles ventilation for the summer period (June-September) are contrasted:

Profile 1: it is the established by the DB HE 2013. It consists in: 4 ACH for the hours between 01:00 h and 08:00 A.M, while a constant ventilation equal to minimum required by the Hygiene, health and the environment, Air Quality Document DB HS-3 of the Building Cod is maintained for the remaining hours of the day.

Profile 2: A constant ventilation level is introduced throughout the day, according to the requirements set in Document DB HS-3.

Once the model has been introduced in the computer tool, the interior temperature and humidity time are obtained and the results are compared.

For the specific case of the building being studied, for verification of surface condensation limiting the number of hours that within the relative humidity is above 80% is taken into account.

3 NORMATIVE CRITERIA

The Building Code establishes, in the current Energy economy and heat retention Document 2013, that the risks related to processes that produce a significant reduction on the thermal performance or elements of the thermal envelope life, have to be limited, such as condensation. While interstitial condensation affect the thermal behavior of the enclosure, surface condensation essentially represent a risk in relation to health, by the formation of mold, and its demand is reflected in the DB HS-3.

The criteria to limit the formation of surface condensation are set out in the Supporting Document to the DB-HE: Limiting testing surface and interstitial condensation in the walls, DA DB-HE / 2:

In that document it is stated that the formation of surface condensation in the walls and interior partitions that make up the thermal envelope of the building, will be limited, so as to avoid the appearance of mold on its inner surface. Therefore, in wall inner surfaces that can absorb water or those susceptible to degradation, especially in thermal bridges thereof, the monthly average relative humidity will be less than 80%.

4 BUILDING DATA

The building considered in the study is an existing single-family building with a net surface of 79.5 m^2, total height from flooring to ceiling is 2.7 m. The volume of the building is a parallelepiped, facing north.

The building has two double bedrooms, one single bedroom, a living room, a kitchen and two bathrooms.

The ventilation level obtained according to the criteria of the current Document DB HS-3 is 0.8 ACH (47 l/s).

The construction characteristics of the building are those defined in Table 1:

Table 1. Composition of building's envelope

	Composition	Width (m)	Conductivity (W/m·K)
Façade walls	Solid brick wall	0.115	0.991
	Cement mortar render	0.015	0.55
	Not ventilated cavity	0.05	-
	Rockwool insulation board	0.06	0.031
	Gypsum board	0.01	0.25
Roof	Gravel layer	0.15	2.00
	XPS insulation board	0.06	0.034
	Felt	0.005	0.05
	Waterproofing membrane	0.005	0.23
	Lightweight mortar	0.10	0.41
	Ceramic pot and beam slab	0.30	-
	Unventilated air space	0.01	0.15
	Gypsum board suspended ceiling	0,01	0.25
Suelo	Ceramic tiles	0.02	1.00
	Cement mortar render	0.03	0.55
	EPS insulation board	0.04	0.029
	Reinforced concrete	0.20	2.30
	Gravel layer	0.20	2.00

5 RESULTS

The results that were obtained from temperature, indoor humidity and increased vapor pressure, for one month, in this case August, are the following:

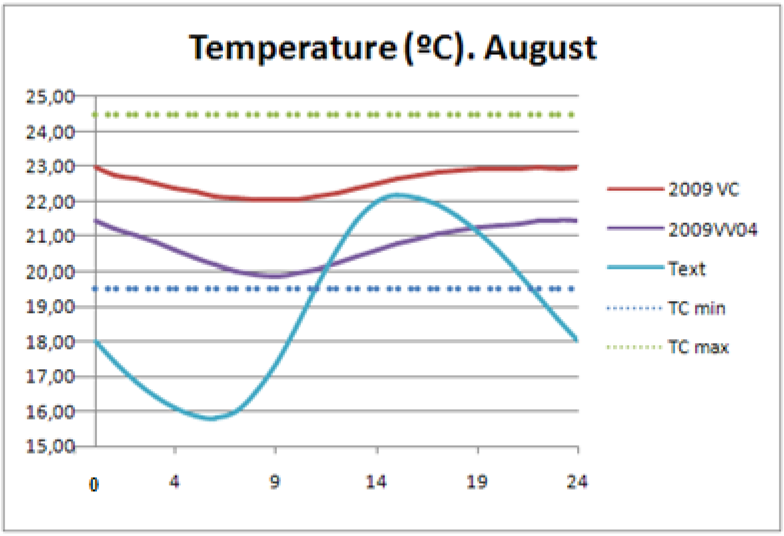


Figure 1.1. Indoor and outdoor temperature conditions.

In the graph of Figure 1.1, it can be observed that for the case in which a constant ventilation profile 2, (2009 VC, red line), is used, the temperature inside the module housing is higher. In both cases they are within the comfort range.

Where,

2009 VC, red line;	Indoor temperature conditions corresponding to profile 2.
2009 VV04, purple line;	Indoor temperature conditions corresponding to profile 1.
Text, blue line;	Outdoor temperature.
TC min, dot line green;	Minimum comfort temperature.
TC max, dot line blue ;	Maximun comfort temperature.

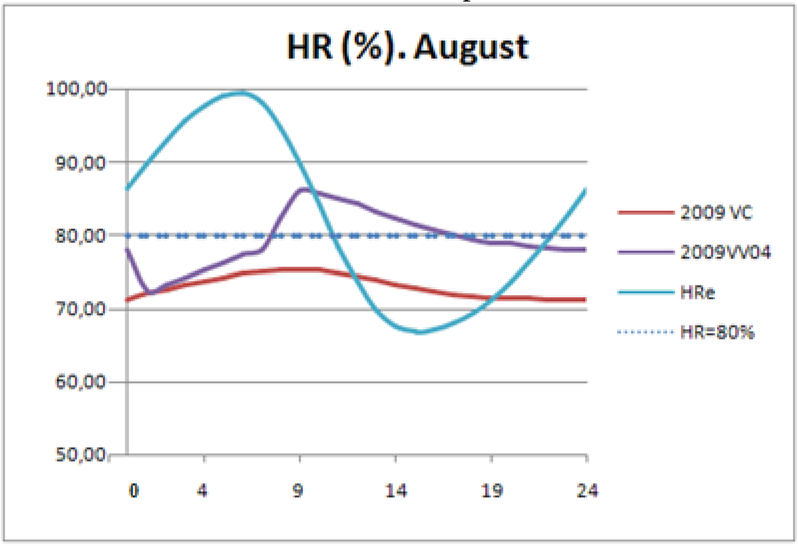


Figure 1.2. Indoor and outdoor relative humidity conditions.

In the graph of relative humidity of Figure 1.2, it is seen how with the profile 1, the indoor relative humidity is over 80% relative humidity, during a certain number of hours.

Where,

2009 VC, red line;	Indoor relative humidity conditions corresponding to profile 2.
2009 VV04, purple line;	Indoor relative humidity conditions corresponding to profile 1.
HRe, blue line;	Outdoor relative humidity conditions.
HR=80%, dark blue line dot;	Limit line to indoor relative humidity conditions.

6 CONCLUSIONS

General conclusions are the following:

- In the provincial capital studied, San Sebastián, belonging to the climate zone D1, for the summer period considered, using a constant ventilation throughout the day (profile 2) limits the risk of formation of surface condensation than profile 1.

- Both profiles considered are out of the comfort range, considered for the indoor relative humidity.

7 REFERENCES

*Código Técnico de la Edificación (*Building Code*).Ministerio de Fomento*. On line.

http://www.codigotecnico.org/web/recursos/documentos/ [Consulted 18/06/2015].

Gavira, M., Linares, P. et al (2005). Comportamiento higrotérmico de la envolvente del edificio según el CTE. Soluciones alternativas: sistemas de ventilación por caudal variable *(Hygrothermal behaviour of the building envelope according to CTE. Alternative solutions: variable flow ventilation systems)*. I Jornadas de investigación, 2, 739-756.

Patologías asociadas a la rehabilitación energética: comportamiento higrotérmico *(Pathologies associated with energy rehabilitation: hygrothermal behavior)*. LARRUMBIDE GÓMEZ-RUBIERA, Enrique. Revista Conarquitectura de Hispalyt, número 53/marzo 2015, Páginas 76 a 81. Depósito legal: M-14395-2001. ISSN: 1578-0201. http://www.hispalyt.es

La problemática de las condensaciones intersticiales en la rehabilitación energética y su aplicación en el DB HE 2013 para las diferentes zonas climáticas peninsulares *(The problem of interstitial condensation in energy rehabilitation and it's application in the DB HE 2013 for different climate zones peninsular)*. Jornadas Internacionales conmemorativas del 80 aniversario del IETcc. Conservar, rehabilitar, innovar. IETcc-CSIC, noviembre de 2014. LARRUMBIDE GÓMEZ-RUBIERA, Enrique; GAVIRA GALOCHA, et al. Libro ponencias: ISBN: ISBN: 978-84-7292-381-2. CD: ISBN: 978-84-7292-382-9. Págs: 10.

Patologías Asociadas a la Rehabilitación Energética *(Energy associated pathologies Rehabilitation)*. Curso de Especialización del CSIC Evaluación, Intervención y Mantenimiento de Edificios y Estructuras. Cursos Avanzados Eduardo Torroja. Seminario 11: La Habitabilidad en el CTE: Novedades y Aplicación a Edificios Existentes. Junio de 2013, IETcc-CSIC. LARRUMBIDE GÓMEZ-RUBIERA, Enrique; Libro ponencias: Págs: 127-126. ISBN: 978-84-7292-414-7.

La problemática de las condensaciones en las islas Canarias, referidas a edificios existentes *(The problem of condensation in the Canary Islands, referred to existing buildings)*. Jornadas Internacionales de Investigación en Construcción: Vivienda, pasado, presente y futuro. IETCC-CSIC, noviembre de 2013. LARRUMBIDE GÓMEZ-RUBIERA, Enrique; et al. Libro resúmenes ISBN: 978-84-7292-421-5. CD ISBN: 978-84-7292-422-2. Págs: 7.

Clima Alfa: una nueva zona climática de invierno para las Islas Canarias *(Climate Alpha: a new Winter climate zone for the Canary Islands)*. Jornadas Internacionales de Investigación en Construcción: Vivienda, pasado, presente y futuro. Junio de 2013, IETcc-CSIC. LARRUMBIDE GÓMEZ-RUBIERA, Enrique; et al. Libro resúmenes ISBN: 978-84-7292-421-5. CD ISBN: 978-84-7292-422-2.

Patologías de Humedad por Condensación Intersticial *(Moisture conditions for interstitial condensation)*. Curso de Sostenibilidad: Eficiencia Energética, Evaluación de Edificios y Estructuras. S5E. Seminario 9: Intervención en Edificios Existentes. Criterios de Seguridad y Habitabilidad. IETcc-CISDEM-CSIC, junio de 2012. SÁNCHEZ RAMOS, Virginia; MARTÍN CONSUEGRA, Fernando; LARRUMBIDE GÓMEZ-RUBIERA, Enrique. Libro de ponencias: Pág: 65-72. ISBN: 978-84-695-3612-4. Págs: 8.

Orden FOM/1635/2013, de 10 de septiembre, por la que se actualiza el Documento Básico DB-HE «Ahorro de Energía», del Código Técnico de la Edificación, aprobado por Real Decreto 314/2006, de 17 de marzo.

Documento de Apoyo al Documento Básico DB-HE Ahorro de Energía (DA DB-HE/2), Comprobación de limitación de condensaciones superficiales e intersticiales en los cerramientos. 2013. Ministerio de Fomento.

VENTILATION TOOL FOR IMPROVING THE USABILITY OF VENTILATION LEVELS RELATED TO SPANISH REGULATIONS

Rafael Villar Burke[1], Daniel Jiménez González[1], Pilar Linares Alemparte[1], and Sonia García Ortega[1]

1 Instituto Eduardo Torroja de Ciencias de la Construcción (IETcc-CSIC)
C/ Serrano Galvache, 4
28033 Madrid, Spain

ABSTRACT

Choosing the right baseline level of ventilation has a big impact in the calculated energy needs of buildings. As designers do usually face usability problems when evaluating this level, they tend to overestimate it, making it needlessly harder to comply with energy efficiency regulations.

In order to help designers to better assess ventilation levels in accordance with the Spanish Technical Building Code (TBC) (CTE) we have developed a tool that can also show the impact on ventilation levels due to usability considerations.

This paper presents this tool as well as some study cases, from which the impact of more flexible ventilation regulations –performance-based- can be appreciated.

The tool is based on existing and in development IAQ regulations.

KEYWORDS

Energy, Building Regulations, Ventilation, Tools

1 INTRODUCTION

With the increase of the energy performance of buildings, both due to energy regulations and general awareness, ventilation (along with infiltration) is becoming a key factor in the building energy needs.

We have observed that designers commonly face usability problems when dealing with ventilation related calculations due to the mismatch between health oriented and energy oriented procedures. This results in an overestimation of the ventilation levels for the evaluation of the energy performance of buildings and to the implementation of ineffective energy saving measures.

This paper shows the two most common failures when evaluating this ventilation levels in the case of residential buildings, their energy impact in two study cases (a single family house and a multifamily block) using the current *Hygiene, health and the environment* Basic Document (DB-HS) of the Spanish building Code (CTE) [CTE06], and what would be the impact of loosing the required ventilation levels according to a draft update of the DB-HS section (DB-HS2015 draft).

The paper shows a tool developed to avoid this kind of sources of error and help designers in the correct assessment of ventilation levels for energy use while staying in conformance to the health requirements in the Spanish Building Code (CTE) [CTE06].

2 CALCULATING VENTILATION LEVELS FOR HEALTH OR ENERGY NEEDS

The *Hygiene, health and the environment* Basic Document (DB-HS) of the Spanish Building Code (CTE) [CTE06] sets minimum ventilation requirements for residential buildings in order to ensure air quality. The ventilation level is calculated from the occupation, space use and area of the several building spaces, but taking in isolation each residential unit of the building and other subsidiary building spaces (like storage rooms, corridors, etc), and considering their supplied and exhausted air needs.

In the case of energy evaluation, the building is instead considered as a whole and the control volume is not coincident with that of the health requirement. More specifically, only the volume inside the thermal envelope is of interest, from which are excluded most spaces out of the residential units. Also, the usual indicator is the net amount of external air which is supplied to the control volume.

These two ventilation levels must be consistent and the health requirements need to be met when evaluating the building energy needs but, due to the mismatches in the two calculation models the common practice incurs in two errors with a significant impact in the energy evaluation of the building, with the effect of overestimating the total air flow:

1. an inconsistent consideration of the building spaces, so the needs of "non inhabited" spaces out of the residential units (such as storage/box rooms) are accounted for in the whole building balance, even when they are not included in the energy model (error 1);
2. a failure to understand the air flows, so the supply and exhaustion air flows are added up instead of balanced (error 2).

3 WEB TOOL

To avoid the aforementioned sources of error, and to assist designers to correctly assess the ventilation needs for energy use, we have developed the web based tool, shown in Figure 1. The user just needs to describe relevant characteristics of the residential unit types and their amount, and the tool displays the ventilation rates for energy modelling, both for the current DB-HS prescriptions and the DB-HS2015 draft.

Figure 31: Web tool user interface

4 STUDY CASES AND SCENARIOS

The building types we have considered are the single family house (Type A) and the multifamily block (Type B) in Figure 2. These buildings are located in Madrid (climate zone D3) and have construction solutions close to the recommendations in Annex E of the *Energy economy and heat retention* Basic Document (DB-HE) of the Spanish Building Code (CTE): energy transmission coefficient U = 0.25W/m^{2}K for walls, U = 0.26W/m^{2}K for roofs, U = 0.44W/m^{2}K for ground-floor and U = 1.6W/m^{2}K for windows.

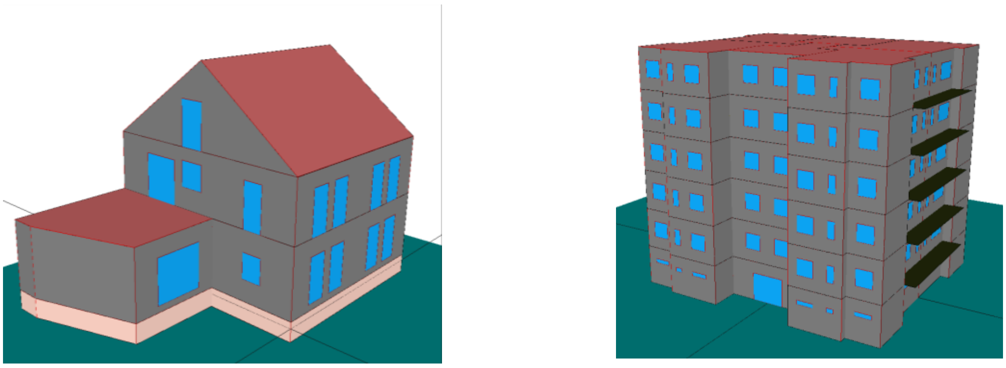


Figure 32: Study cases: building types A and B

We have studied four scenarios for each type, using the current DB-HS evaluation as baseline:

- *scenario 1*, use ventilation levels from DB-HS, but wrongly adding supply and exhaustion flows, and incorrectly considering the ventilation needs of non inhabited spaces;
- *scenario 2*, use ventilation levels from DB-HS, and incorrectly considering the ventilation needs of non inhabited spaces;
- *scenario 3*, use ventilation levels from DB-HS (*baseline*);
- *scenario 4*, use ventilation levels from the DB-HS2015 draft.

Table 1 shows the resulting ventilation levels for the selected scenarios, keeping a reference to the baseline ventilation value.

Table 1: Ventilation levels

Scenario	**Single family house (A)**			**Multifamily block (B)**		
	l/s	ren/h	%	l/s	ren/h	%
1. DB2009 + error 1 + error 2	252	1,00	221	1446	1,32	173
2. DB2009 + error 1	208	0,83	182	1021	0,93	122
3. DB-HS2009	114	0,67	100	835	0,76	100
4. DB-HS2015 (draft)	58	0,34	51	614	0,56	73

We have used the web tool to calculate the ventilation rates for building types A and B in scenarios 3 and 4, as can be seen in Figure 2 and 3.

Ventilación global de edificios de vivienda (DB-HE)

Cálculo del nivel de ventilación global (excluidas infiltraciones) para uso en la modelización energética.

RESULTADOS	DB-HS 2009				DB-HS 2015 (propuesta)			
Ventilación del edificio para cálculos energéticos (DB-HE)	114	l/s	0.67	ren/h	58	l/s	0.34	ren/h

Definición de tipos de vivienda en el edificio

611.55 Volumen habitable del edificio (m³)

Entrada de datos

2 Cantidad de viviendas · apartamento Nombre · 46.5 Sup. útil (m²) · 3 Altura libre (m)

4.5 Zonas comunes asociadas (m²) · 2 Locales húmedos (número)

1 Dormitorios (número) · 1 Baños y aseos (número)

1 Salas de estar, comedores, etc (número) · 6 Superficie de la cocina (m²)

Añadir Borrar Modificar Limpiar

Tipo	Cantidad	Superficie	Altura	Z. comunes	Dormitorios	S. estar	Loc. húmedos	Aseos/baños	Cocinas
prncpial	1	92.85 m²	3 m	9 m²	2	1	3	2	12 m²
apartamento	2	46.5 m²	3 m	4.5 m²	1	1	2	1	6 m²
TOTALES	3	185.85 m²	3 m	18 m²	4	3	7	4	24 m²

Figure 33: Ventilation levels for type A, in with the web tool user interface

Ventilación global de edificios de vivienda (DB-HE)

Cálculo del nivel de ventilación global (excluidas infiltraciones) para uso en la modelización energética.

RESULTADOS	DB-HS 2009				DB-HS 2015 (propuesta)			
Ventilación del edificio para cálculos energéticos (DB-HE)	835	l/s	0.76	ren/h	614	l/s	0.56	ren/h

Definición de tipos de vivienda en el edificio

3952.80(Volumen habitable del edificio (m³)

Entrada de datos

20 Núm. viviendas · 2B Nombre · 54.25 Sup. útil (m²) · 2.7 Altura libre (m)

7.36 Zonas comunes asociadas (m²) · 2 Locales húmedos (número)

2 Dormitorios (número) · 1 Baños y aseos (número)

1 Salas de estar, comedores, etc (número) · 8.4 Superficie de la cocina (m²)

Añadir Borrar Modificar Limpiar

Tipo	Número	Superficie	Altura	Z. comunes	Dormitorios	S. estar	Loc. húmedos	Aseos/baños	Cocinas
2B	20	54.25 m²	2.7 m	7.36 m²	2	1	2	1	8.4 m²
1A	5	39 m²	2.7 m	7.36 m²	1	1	2	1	6 m²
TOTALES	25	1280 m²	2.7 m	184 m²	45	25	50	25	198 m²

Figure 34: Ventilation levels for type B, in the web tool user interface

5 RESULTS

Table 2 shows the resulting heating and cooling energy needs for the selected scenarios, keeping a reference to the baseline. These energy needs were calculated using the Building Code Energy Use tool *Herramienta Unificada LIDER-CALENER [HULC14]* and their values analysed using the *Visol* tool [VILLAR14].

Table 2: Heating and cooling energy needs

Scenario	Single family house (A)		Multifamily block (B)	
	Heating needs [kWh/m²·y]	Cooling needs [kWh/m²·y]	Heating needs [kWh/m²·y]	Cooling needs [kWh/m²·y]
1. DB-HS + error 1 + error 2	85,50 (198%)	15,60 (112%)	52,05 (197%)	15,04 (106%)
2. DB-HS + error 1	71,40 (165%)	15,01 (108%)	33,96 (129%)	14,39 (101%)
3. DB-HS	43,21 (100%)	13,88 (100%)	26,42 (100%)	14,18 (100%)
4. DB-HS2015 (draft)	30,83 (71%)	13,40 (97%)	18,68 (71%)	14,00 (99%)

Table 3 shows the energy needs due to ventilation for the previous scenarios, in absolute terms and relative to the total net energy needs for each scenario.

Table 3: Ventilation weight for heating and cooling energy needs

Scenario	Single family house (A)		Multifamily block (B)	
	Ventilation in heating needs [kWh/m²·y]	Ventilation in cooling needs [kWh/m²·y]	Ventilation in heating needs [kWh/m²·y]	Ventilation in cooling needs [kWh/m²·y]
1. DB-HS + error 1 + error 2	90,60 (106%)	-12,90 (-83%)	75,30 (145%)	-14,80 (-98%)
2. DB-HS + error 1	75,60 (106%)	-13,40 (-89%)	56,00 (165%)	-15,50 (-108%)
3. DB-HS	45,60 (106%)	-14,50 (-104%)	47,60 (180%)	-15,80 (-111%)
4. DB-HS2015 (draft)	31,20 (101%)	-14,90 (-111%)	36,20 (194%)	-16,20 (-116%)

Figure 5 depicts the results from tables 2 and 3.

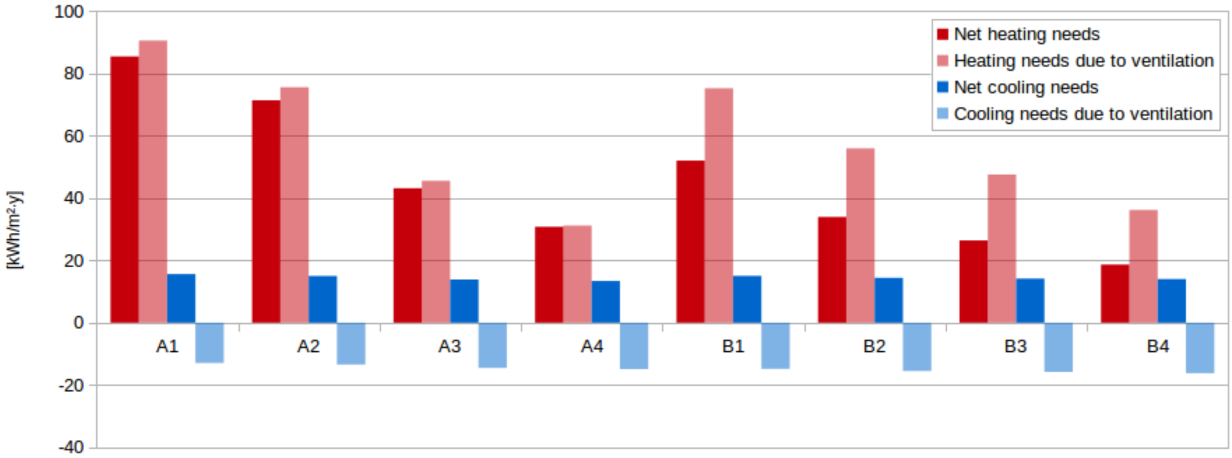


Figure 35: Net energy needs for heating and cooling and energy needs for heating and cooling due to ventilation

6 CONCLUSIONS

For the selected scenarios, building types and the climate zone we have studied, we can see the great impact of the two most commonly detected sources of error when evaluating the ventilation levels for energy use. These errors can almost double the estimated energy needs in some cases, with more impact in the heating needs and less impact in the cooling needs. Also, ventilation and infiltration represent a great share of the energy needs, ranging from roughly the net energy needs for cooling to twice the net energy needs for heating.

It is interesting to note that ventilation tends to have an increasing weight in the net energy needs, as ventilation levels go down, even when, overall, the use of stricter ventilation levels (still complying with air quality requirements) can lead to significant reductions in the building energy needs.

These results show the importance of a correct estimation of the ventilation levels for use in energy modelling, and how calculation errors or regulation requirements have a significant impact in the energy budget and profile of the building, leading to different cost effective energy saving measures. Also, it proves the usefulness of a tool to help users to avoid the most common mistakes.

7 REFERENCES

[CTE06] Real Decreto 314/2006 de 17 de marzo (BOE 28 de marzo de 2006). "Código Técnico de la Edificación". Ministerio de Vivienda, Madrid, 2006.

[HULC14] Herramienta Unificada LIDER-CALENER. Herramienta informática de verificación de la exigencia de Limitación de demanda energética del CTE. Ministerio de Fomento, Ministerio de Industria, Energía y Turismo, IDAE. Madrid, 2014. http://www.codigotecnico.org

[VILLAR14] Villar Burke, R.; Jiménez González, D.; *Visol: Visor de archivos de resultados de LIDER*, Madrid, 2014. http://www.rvburke.com/visorlider.html

IMPACT OF THE NEW RITE 2013 (REGULATION ON THERMAL INSTALLATION) ON INDOOR AIR QUALITY

Pastor Pérez, Paulino

Ambisalud
Velazquez, 100
Madrid, Spain

ABSTRACT

This paper presents a comparison of Indoor Air Quality in several buildings constructed prior to the implementation of the new Spanish regulation on thermal installations (RITE, 2007 modified on 2013) and some new ones that fully accomplish the requirements of this new regulation. The objective is to confirm whether new regulation had a positive impact on the indoor air quality.

The month of April 2014 marked a year since the publication of the amendment of RITE (RD 238/2013 of April 5, 2013 Spanish Regulation on Thermal Installations), among the changes that were included, we highlight two that had a great relevance in terms of Indoor Air Quality. Within the document in the section IT 3.3 describing maintenance programs to be implemented in facilities subject to RITE (> 70kW), two maintenance items where added:

38. Review of the ductwork and AHU hygiene according to the UNE 100012: t.
39. Revision of indoor environmental quality according to the UNE 171330: t.

This is accomplished by two types of reviews including several analysis and visual inspections, first ensure that HVAC systems are in good hygienic conditions (UNE 100012) and secondly that the quality of the air users breathe is satisfactory (UNE 171330).

On the other hand since 2007 RITE makes compulsory to install high quality filters according to outdoor air quality.

In order to estimate the positive impact of such measure we have analyzed Indoor Air Quality according to RITE requirements in 38 buildings half of them with low quality filtration (projects made before 2007) and the other half buildings projected and executed according to new RITE.

The summary of findings indicates an overall positive effect in all indicators related to pollution mainly found outdoor as particulates and fungi, but not so relevant in airborne bacteria since those are produced indoor by the occupants mainly.

Building projected, executed and maintained according to new RITE 2013 requirements offer in average 60% more reduction of pollution than old ones.

It must be taken into account that new buildings are also tighter than the old ones. Carbon dioxide have been used as and indicator of tightness being the Indoor-Outdoor relative CO_2 concentration higher in new buildings

(indoor minus outdoor concentration of CO_2). That means new buildings have higher CO_2 readings but that does not mean worst Indoor Air Quality since filtration has been enhanced.

CONCLUSION:

New RITE buildings accomplish the goal of offering better indoor air quality enhancing at the same time energy efficiency.

KEYWORDS

Indoor Air Quality, Ventilation, Carbon dioxide, Indoor particles, Fungi, Bacteria.

1 INTRODUCTION

This paper presents a comparison of Indoor Air Quality in several buildings constructed prior to the implementation of the new Spanish regulation on thermal installations (RITE, 2007 modified on 2013) and some new ones that fully accomplish the requirements of this new regulation.

In order to estimate the positive impact of such measure we have analyzed Indoor Air Quality according to RITE requirements in 38 buildings half of them with low quality filtration (projects made before 2007) and the other half buildings projected and executed according to new RITE.

The summary of findings indicates an overall positive effect in all indicators related to pollution mainly found outdoor as particulates and fungi, but not so relevant in airborne bacteria since those are produced indoor by the occupants mainly.

The main overall conclusion of our study is that buildings constructed accomplishing RITE 2013 requirements offer 60% higher reduction of pollution and are tighter in terms of infiltration, meaning that enhanced IAQ and energy efficiency can be achieved together.

2 DESCRIPTION OF RITE (SPANISH REGULATION ON THERMAL INSTALLATIONS)

RITE is a regulation mainly intended originally to save energy in thermal installations of buildings. The aspects related to IAQ are applicable to all buildings holding a thermal capacity above 70 kW.

Some key questions of RITE we have used in our study are the following:

1.- Design requirements that affect IAQ:

- 1.1 The filtration must be designed taking into account outdoor level of pollution
- 1.2 There must be a minimum level of ventilation, that can be expressed in terms of ventilation rate (liters/second-person), but also as a maximum CO_2 level concentration.

2.- Periodic review of IAQ

- 2.1 In section IT 3.3 dealing with maintenance requirements RITE obligates to review and register IAQ and hygienic conditions of the building and the HVAC system respectively.
- 2.2 The standards applicable for the reviews are:

Ductwork and AHU hygiene according to the UNE 100012.
Indoor environmental quality according to the UNE 171330.

2.1 RITE: IAQ DESIGN REQUIREMENTS

Spanish HVAC regulation (RITE) establishes the quality of the filtration systems that must be installed in new or refurbished buildings according to the level of outdoor air pollution (ODA -Outdoor Air) existing in the area where the building is to be located range 1 to 3. The quality of the filters depends also on the type of activity which may require different levels of indoor air quality (IDA -Indoor Air range 1 to 4). The rationale has been taken from EN 13779 Standard see Table 1.

Table 1. Filtration requirements according to RITE

Outdoor Air Quality	Indoor Air Quality			
	IDA 1	IDA2	IDA 3	IDA 4
ODA 1	F9	F8	F7	F5
ODA 2	F7 + F9	F6 + F8	F5 + F7	F5 + F6
ODA 3	F7+GF+F9a	F7+GF+F9a	F5 + F7	F5 + F6

GF = Gas filter

The meaning of each aspect is shown on tables 2 and 3:

Table 2. Meaning of ODA

Category	Description
ODA 1	Pure air which may be only temporarily dusty e.g. pollen
ODA 2	Outdoor air with high concentrations of particulate matter and/or gaseous pollutants (within 1 to 1,5 times National Air Quality Standards)
ODA 3	Outdoor air with very high concentrations of particulate matter and/or gaseous pollutants. (above 1,5 times National Air Quality Standards)

Table 3. Meaning of IDA

Category	Description	Example of uses
IDA 1	High indoor air quality	Hospitals , clinics, laboratories and nurseries
IDA 2	Medium indoor air quality	offices, residences (common premises of hotels and similar (retirement and students houses), meeting and reading rooms, museums, courtrooms , classrooms and similar teaching areas and pools
IDA 3	Moderate indoor air quality	Commercial buildings , cinemas , theaters, auditoriums, hotel rooms and similar, restaurants, cafes, bars, clubs, gyms , rooms for sport (except swimming pools) and computer rooms .
IDA 4	Low indoor air quality	Not specified

Filtration systems are rated according to EN 779 see Table 4:

Table 4. Classification of filters

	Filter class	Average arrestance [%]	Average efficiency [%]	Minimum efficiency [%]
Coarse dust filters	G1	50 ≤ Am < 65	-	-
	G2	65 ≤ Am < 80	-	-
	G3	80 ≤ Am < 90	-	-
	G4	90 ≤ Am	-	-
Medium and Fine dust filters	M5	-	40 ≤ Em < 60	-
	M6	-	60 ≤ Em < 80	-
	F7	-	80 ≤ Em < 90	35
	F8	-	90 ≤ Em < 95	55
	F9	-	95 ≤ Em	75

RITE also establishes the levels of mechanical ventilation that must be ensured in order to achieve an acceptable level of air renovation. The level depends on IDA as shown on table 5

Table 5. Level of ventilation

Category	Level of ventilation	
	Liters/second-person	Indoor - outdoor relationship* (ppm)
IDA 1	20	350
IDA 2	12,5	500
IDA 3	8	800
IDA 4	5	1200

* Calculated as absolute CO2 concentration outdoor minus absolute concentration indoor (in ppm)

2.2 RITE: PERIODIC IAQ INSPECTIONS.

The periodic inspection must be done according to standard UNE 171330 which establishes a minimum set of parameters that could be considered as general indicators of the Indoor Air Quality of any building, these are the following:

- Inspection of hygiene of HVAC system (including ductwork test: surface microorganisms, gravimetric analysis of settled dust)
- Carbon dioxide
- Temperature and relative humidity
- Carbon monoxide
- Particle concentration (PM10)
- Particle counting (size 0,5 microns)
- Airborne bacteria
- Airborne fungi

HVAC systems are the basic equipment involved in the control of IAQ, if the general maintenance, not only mechanical but also hygienic, does not meet a minimum standard the system could not only fail in enhancing IAQ, but even worse, could become a source of pollution (particles, fungi, bacteria, bad smell, etc)

Hygienic inspection of HVAC system

This is made according to UNE 100012 requirements and consist on visual inspection (ranging from 1 to 3 meaning 1 clean and 3 dirty), plus surface sampling for microorganisms and total settled dust.

Carbon dioxide

Readings of carbon dioxide indoors are referred to outdoor concentration. Relationship Indoor-Outdoor is the parameter to control. It is an excellent indicator of the quality of the ventilation.

Some critical aspects about carbon dioxide readings are the level of occupancy or timing, a typical building may need at least 1 hour to reach steady state, only at this moment readings are meaningful.

Temperature and relative humidity

Thermal comfort represents around 30% of typical complaints about indoor environments, however this is a parameters not directly related to IAQ therefore it was not included in the study.

Carbon monoxide

Carbon monoxide is rarely a problem, it could be a good indicator of car exhaust either from building parking areas or from general outdoor traffic. It has not been cosidered.

Particle concentration (PM10) and particle counting (size 0,5 microns)

Particle reading is a good indicator of the quality of the filtration system, concentration is a health concern.

Airborne bacteria and fungi

Humans are a source of airborne bacteria, so typically indoor airborne concentration is higher than outdoor, this is a good indicator of the general hygiene of the building and also a complementary indicator of the quality of the ventilation.

Fungi is mainly pulled in the building from outdoor and it is a good indicator of the hygiene of the HVAC and the quality of the filtration system.

All these parameters must be checked on a yearly basis according to Spanish law.

The inspection standard also establishes the number of sampling points to be taken as a result of the formula:

$$P = 0{,}15 \times \sqrt{A}$$

where

P= Nº sampling points

A: Area under study

The idea is that the number of sampling points do not have to increase linearly with the area under study.

3 DESCRIPTION OF THE BUILDINGS

Indoor air quality of 19 buildings designed and operated according to new RITE requirements (filter quality F7-F9 and enhanced hygiene maintenance, from now on Buildings F7/F9) were analyzed and the results have been compared with those in 19 old buildings (filter quality G4, from now on Buildings G4). We have analyzed data from 38 buildings. The inspection was made following UNE 171330 Standard.

The characteristics of the buildings investigated are the following:

BUILDINGS G4

Type of HVAC systems

ALL AIR CONSTANT VOLUME	35%
ALL AIR VARIABLE AIR VOLUME	30%
VRV (VARIABLE REFRIGERANT)	20%
AIR-WATER. PRIMARY AIR UNIT/FCU (FAN COIL UNITS)	15%

Quality of filters:

84%	G4
11%	G3
5%	F5

Cities: Madrid, Barcelona, Bilbao, Jerez and Badajoz.

Size of the buildings:

AVERAGE AREA	8.634 m2
MINIMUN AREA	1.111 m2
MAXIMUN AREA	30.044 m2

Outdoor air quality:

ODA 1	11%
ODA 2	53%
ODA 3	37%

BUILDINGS F7/F9

Type of HVAC systems

ALL AIR CONSTANT VOLUME	10%
ALL AIR VARIABLE AIR VOLUME	50%
VRV (VARIABLE REFRIGERANT)	20%
AIR-WATER. PRIMARY AIR UNIT/FCU (FAN COIL UNITS)	20%

Quality of filters:

F8	28%
F7	56%
F9	17%

Cities: Madrid, Barcelona, Sevilla and Albacete.

Size of the buildings:

AVERAGE AREA	21.294 m2
MINIMUN AREA	2.178 m2
MAXIMUN AREA	64.178 m2

Outdoor air quality:

ODA 1	11%
ODA 2	67%
ODA 3	22%

In both cases the inspections were made within September, 2013 and October, 2014

4 RESULTS OF THE STUDY

The results of the hygienic inspection of the air handling units was:

Table 6: Air Handling Units Hygiene

Air Handling Units	VISUAL INSPECTION	SURFACE BACTERIA	SURFACE FUNGI
	Range 1 to 3	<100 ufc/25cm^2	<100 ufc/25cm^2
BUILDINGS G4	1,7	39	41
BUILDINGS F7/F9	1,8	43	20

The level of dirt visually estimated is similar regardless the type of system, it depends basically on the quality of the maintenance.

About microorganisms bacteria differences are not relevant but fungi is almost 50% less in higher quality buildings.

Fungi are relatively big particles coming mainly from outdoor and therefore can be better controlled by filters. See table 6.

Table 7: Ducts Hygiene

DUCTS	VISUAL INSPECTION	SURFACE BACTERIA	SURFACE FUNGI	SETTLE DUST
	Range 1 to 3	<100 ufc/25cm^2	<100 ufc/25cm^2	mg/100cm^2
BUILDINGS G4	2,1	73	70	17
BUILDINGS F7/F9	2,2	51	47	11

Ductworks systems do not present differences in terms of level of dirt, it is interesting to point that ducts are dirtier than AHU´s, probably because of difficulties of access.

Bacteria, fungi and settle dust levels are significantly better in F7/F9 buildings. Better filtration protects ducts from excessive contamination.

See table 7.

Table 8: Airborne Particles (PM10)

AIRBORNE PARTICLES (PM10)	INDOOR AIRBORNE PARTICLES	OUTDOOR AIRBORNE PARTICLES
	micrograms/m^3	micrograms/m^3
BUILDINGS G4	13,3	19,9
BUILDINGS F7/F9	8,0	18,9
	% REDUCTION	
BUILDINGS G4	-50%	
BUILDINGS F7/F9	-138%	

Table 9: Airborne Particles (0,5 microns)

AIRBORNE PARTICLES (0,5 MICRONS) COUNTING	INDOOR 0,5 MICRONS PARTICLES	OUTDOOR 0,5 MICRONS PARTICLES
	particles/ft^3	particles/ft^3
BUILDINGS G4	2.699.391	3.688.003
BUILDINGS F7/F9	2.024.102	4.564.890
	% REDUCTION	
BUILDINGS G4	-37%	
BUILDINGS F7/F9	-126%	

Airborne particles (PM10) and 0,5 micros are much better retained by F7/F9 filtration systems. See tables 8 and 9.

Table 10: Airborne Fungi

AIRBORNE FUNGI	INDOOR AIRBORNE FUNGI	OUTDOOR AIRBORNE FUNGI
	cfu/m^3	cfu/m^3
BUILDINGS G4	122	371
BUILDINGS F7/F9	24	234
	% REDUCTION	
BUILDINGS G4	-204%	
BUILDINGS F7/F9	-855%	

Airborne fungi reduction in the buildings with filters F7F9 are up to 4 times better than in G4 buildings.

Some fungi species can promote allergic reactions on some people. Reducing the levels can enhance indoor air quality and can help on the durability of decoration materials.

Table 11: Airborne Bacteria

AIRBORNE BACTERIA	INDOOR AIRBORNE BACTERIA	OUTDOOR AIRBORNE BACTERIA
	cfu/m^3	cfu/m^3
BUILDINGS G4	246	268
BUILDINGS F7/F9	107	154
	% REDUCTION	
BUILDINGS G4	-9%	
BUILDINGS F7/F9	-44%	

Indoor airborne bacteria is heavily influenced by the presence of people, main source of those.

Control of bacteria by means of filters is not totally feasible, good ventilation can be more effective in this case. However buildings F7/F9 perform better than G4.

Table 12: Carbon Dioxide

CARBON DIOXIDE	INDOOR-OUTDOOR RELATIONSHIP
	Limit Value < 500 ppm
BUILDINGS G4	178
BUILDINGS F7/F9	237

Finally the results of carbon dioxide readings, show more ventilation in G4 buildings than F7/F9, however this is probably due to the fact that new buildings facades are tighter and then less infiltration is allowed, because mechanical ventilation must be higher according to new requirements.

In any case regardless the origin of the fresh air, excessive ventilation means energy waste, and according to our study this is useless since Buildings F7/F9 even though have less overall fresh air entrance perform better in terms of presence of particles and microorganisms.

Carbon dioxide itself is not a concern in terms of air quality, levels of 600 ppm to 700 ppm are perfectly normal. The air in humans lungs can reach up to 40.000 ppm CO2 concentration.

CONCLUSION

New RITE has been a major advance in terms of enhanced Indoor Air Quality in Spanish buildings built or refurbished according to this new requirements. In the near future the trend is that most buildings will be renovated to fulfill these requirements for the benefit of users.

This will have an impact not only in terms of enhanced quality of life but even from an economic point of view.

5 REFERENCES

Indoor Air Quality Handbook. Mc Graw Hill. John D. Spengler. 2000

Green Building and Productivity. Norm G. Miller. University of San Diego. 2009

DTIE 2.02: Calidad de aire interior Paulino Pastor Pérez. Atecyr 2006

DTIE 2.05 Calidad del Aire Exterior: Mapa de ODAs de las principales capitales de provincia de España Paulino Pastor Pérez. Atecyr 2013

UNE 171330 Indoor Air Quality

UNE 100012 Hygiene of HVAC systems.

INFILTRATION AND VENTILATION IN A VERY TIGHT, HIGH PERFORMANCE HOME

Lisa Ng, Andrew Persily*, and Steven Emmerich

National Institute of Standards and Technology
100 Bureau Drive, MS8600
Gaithersburg, Maryland USA
andyp@nist.gov

ABSTRACT

The Net Zero Energy Residential Test Facility (NZERTF) was constructed at the National Institute of Standards and Technology (NIST) to support the development and adoption of cost-effective net zero energy designs and technologies. Key design objectives included providing occupant health and comfort through adequate ventilation and reduced indoor contaminant sources. The 250 m^2 two-story, unoccupied NZERTF was completed in 2012 with the following design goals: meeting the comfort and functional needs of the occupants; siting to maximize renewable energy potential; establishing an airtight and highly insulated building enclosure designed for water and moisture control; providing controlled mechanical ventilation; and installing highly efficient mechanical equipment, lighting and appliances. The NZERTF achieved its goal of generating more energy than it consumed during its first year of simulated occupancy by a single family, despite a severe winter at the building site. The airtightness goal was achieved through detailed envelope design, careful construction, and during- and post-construction commissioning techniques. The NZERTF is one of the tightest residential buildings in North America with a whole building pressurization test result of roughly 0.6 h^{-1} at 50 Pa. The ventilation goals were met with a heat recovery ventilator sized to comply with ASHRAE Standard 62.2-2010, which corresponds to roughly 40 L/s or 0.1 h^{-1} for this building.

This paper describes the design and construction methods used to achieve such a tight building as well as the performance measurements made to verify that the building achieved its airtightness and ventilation goals. Tracer gas measurements of air change rates are reported, as well as multizone airflow model predictions of these same rates for comparison. This study highlights some of the measurement and modelling challenges in very tight buildings.

KEYWORDS

Airtightness; netzero energy; residential; ventilation.

1 INTRODUCTION

Residential buildings in the U.S. and other countries have historically been ventilated by infiltration, supplemented by window openings and local exhaust ventilation. As energy efficiency has become a priority, buildings were built to be more airtight and mechanical ventilation started to be used to meet building ventilation requirements. The U.S. was slower in making these changes compared to some countries, particularly the Nordic countries in

Europe, but U.S. homes are getting tighter (Chan et al., 2013) and mechanical ventilation is becoming more common (Persily, 2015). It goes without saying that infiltration is not a very good way to ventilate a building as the rate and air distribution is not controlled, the entering air is not be filtered for outdoor contaminants or dehumidified, and the rates tend to be highest during more severe weather when the energy penalty is greatest. Mechanical ventilation allows the rates to be controlled and the incoming air to be treated, as well as providing the opportunity for heat recovery. In general, mechanical ventilation will provide better performance when combined with a tight envelope, or in the words of Arne Elmroth "Build tight, ventilate right" (Elmorth, 1980).

The Net Zero Energy Residential Test Facility (NZERTF) was built on the campus of the National Institute of Standards and Technology to demonstrate low energy residential technologies with the goal of netzero energy use on an annual basis. This paper describes the design and construction methods used in the NZERTF to achieve a very tight building with reliable mechanical ventilation, as well as the results of selected performance measurements in the building. This study highlights some of the measurement and modelling challenges in very tight buildings.

2 DESIGN AND CONSTRUCTION OF THE NZERTF

The NZERTF is a 250 m^2 two-story, unoccupied house located in Gaithersburg, Maryland with an unfinished basement and an attic, both within the conditioned space. The building envelope was constructed using advanced framing techniques (i.e., studs of greater depth than typical of U.S. construction, allowing for more insulation to be installed) with a continuous fully-adhered membrane air and moisture barrier sealed down to the foundation wall (Figure 1). The nominal R-value of the wall assembly was R-7.9 m^2·K/W. The roof insulation was part of the roof structure, a nominal R-value of R-12.7 m^2·K/W. A 10.2 kW photovoltaic system was located on the main roof. Four solar thermal collectors were located on the roof of the front porch to provide domestic hot water. More details on the building design and construction can be found in Pettit et al. (2014). Internal loads, energy and water usage of a virtual family of two adults and two children were simulated according to daily schedules (Omar and Bushby, 2013). Sensible heat from the occupants was simulated throughout the house, while the latent loads were located only in the kitchen. Energy performance results for the first year of operation of the house, which demonstrated that the facility achieved better than net zero energy, are found in Fanney et al. (2015).

While the NZERTF was designed with several heating, ventilating, and air conditioning (HVAC) options for future research purposes, only a two-speed air-to-air heat pump with a dedicated dehumidification function has been used to date. Ventilation was provided continuously by a heat recovery ventilator (HRV) with dedicated ductwork. It supplied air to the living room on the first floor and the three bedrooms on the second floor. The air returned to the HRV was drawn from a bathroom on the first floor and two bathrooms on the second floor. It was sized to comply with ASHRAE 62.2-2010 (ASHRAE, 2010) which corresponds to roughly 40 L/s. Based on the available HRV fan settings, the actual ventilation supplied to the house was 56 L/s, as measured by duct traverse. It is interesting to compare the ventilation rate based on Standard 62.2 with the requirement based on other standards. For example, the historical ventilation requirement for residences in Standard 62, which last appeared in 62.1-2001, was 0.35 h^{-1} or 123 L/s. A recent review of residential ventilation requirements in Europe showed that several countries require 0.5 h^{-1}, which equals 176 L/s for the NZERTF (Dimitroulopoulou, 2012). It is worth noting that a literature review of ventilation rates and health found that air change rates about 0.5 h^{-1} have been associated with reduced risk of allergic symptoms in children in Nordic climates (Sundell et al., 2011).

Figure 1. Construction of NZERTF showing the air barrier (top) and completed structure (bottom).

Indoor air quality (IAQ)-based guidelines were developed for this project to support the design goal of providing good IAQ in this low-energy residence, in particular to guide the selection of interior finishes and insulation. The guidelines were mostly prescriptive, requiring use of certain builder installed products and avoidance of others, with the objective of reducing common sources of volatile organic compounds (VOC) that affect health and comfort. Emphasis was placed on reducing sources of formaldehyde emissions based on its known health impacts (IARC, 2012). Reduced emissions of VOCs in solvents were addressed by incorporating maximum VOC content requirements for wet-applied products. Guidelines were also included for adhesives and sealants, paints and coatings, built-in cabinetry, woodwork, doors, countertops, floor coverings, and insulation. As a result of careful material selection, the formaldehyde levels measured in the NZERTF over the course of eight months were on average 80 % less than the average measured in other new homes and 60 % less than the average measured in existing homes (Poppendieck et al., 2015). Levels of acetic acid, toluene, and other VOCs were also lower on average than in new and existing homes. These measurements also showed that when the HRV was off, steady-state concentrations of selected VOCs could rise nine times higher than outdoors. Note that the house contained no furniture, which is another source of VOC emissions. The NZERTF IAQ guidelines have been updated and formalized into a detailed architectural specification intended for use in new residential construction and major renovations. This specification is written in a manner so that it can be applied to any project and is available in Bernheim et al. (2014).

3 AIRTIGHTNESS, INFILTRATION AND VENTILATION MEASUREMENTS

The airtightness and ventilation rates of the NZERTF were measured to verify its performance relative to its design goals.

3.1 Envelope Airtightness

Five blower-door tests were performed at the NZERTF to confirm that the envelope airtightness met the design targets (Figure 2). The first three tests (without windows, pre-drywall, and substantial completion) were conducted by third-party testing companies (Pettit et al., 2014). The final tests (#4 and #5) were performed by NIST after the house was complete, according to the methods in ASTM E779-10 (ASTM, 2010). These results have an uncertainty of about 10 %. Test #4 was performed with the kitchen and dryer vents sealed and yielded an airflow rate of 195 L/s at 50 Pa, which corresponds to 0.55 h^{-1}. Test #5 was performed with those vents unsealed, yielding 223 L/s at 50 Pa or 0.63 h^{-1}. This airtightness value is compared with several guidelines in Table 1 and is tighter than the requirements in LEED and ENERGY STAR and slightly leakier than the Passiv Haus requirement. The Normalized Leakage value for the house equals 0.06. Based on statistical analysis of Lawrence Berkeley National Laboratory Residential Diagnostics Database (ResDB) by Chan et al. (2013), the NZERTF is tighter than well over 99 % of U.S. homes.

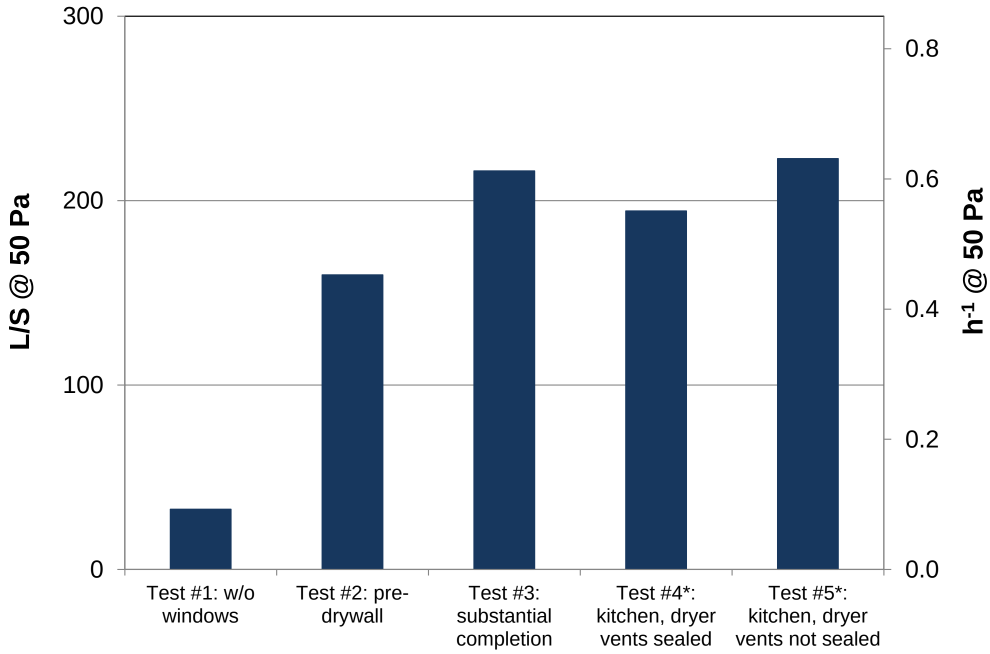


Figure 2. Blower door test results at various stages of construction. * indicates the tests performed by NIST

Table 1. Summary of NZERTF airtightness and relevant guidelines for airtightness

Guideline/Standard	**Target Airtightness (L/s at 50 Pa)**
NZERTF Design Target	381
DOE Challenge Home (DOE, 2013)	953
ICC 700 National Green Building Standard (testing option) (NAHB/ICC, 2012)	2648
LEED BD+C: Homes v4 (for 2 points) (USGBC, 2014)	706
ENERGY STAR v3.1 (rev. 06) (EPA, 2015)	1059
Passiv House (PHI, 2015)	212

3.2 Infiltration

The total outdoor air change of the NZERTF was measured on several occasions using tracer gas decay (ASTM, 2011) with the HRV on continuously and with it off. During these measurements, the heat pump and its air distribution fan were controlled by the thermostat. Measurements were made in July 2014, August 2014 and January 2015. For the summer measurements, with an average indoor temperature of 27 °C, an average outdoor temperature of 23.0 °C and an average wind speed of 1.6 m/s, the average outdoor air change rate with the HRV on was 0.17 h^{-1} and 0.02 h^{-1} with the HRV off. In the winter, with an average indoor temperature of 21 °C, an average outdoor temperature of -2.9 °C and an average wind speed of 2.9 m/s, the average outdoor air change rate with the HRV on was 0.19 h^{-1} and 0.06 h^{-1} with the HRV off.

The fan off air change rates were obviously very low, which raises questions regarding their measurement accuracy. The ASTM tracer gas dilution standard E741 (ASTM, 2011) states that the uncertainty associated with these measurements is 10 %. That standard also notes that the lower the air change rate, the longer the decay measurement required to achieve this uncertainty. For air change rates on the order of 0.03 h^{-1}, the standard suggests that decay tests need to last on the order of 24 h. The measurements reported on here, and in most field studies, rarely last more than a few hours, which presents a challenge when measuring such low air change rates. It is also worth noting that even with essentially zero indoor-outdoor temperature difference and a very low wind speed, the tracer gas decay measurements yielded a non-zero air change rate. It is unclear if these results were a result of the measurement uncertainty or if they reflected physical mechanisms inducing air change beyond the traditionally considered stack and steady-state wind effects. Previous measurements, dating back decades and performed in much leakier buildings, have shown nonzero air change rates even when there are very low driving forces. Regardless of any questions regarding the accuracy of these measurements and the values measured under very mild weather conditions, the infiltration rates are quite low.

3.3 Ventilation

As noted earlier, the HRV in the NZERTF was sized to comply with ASHRAE Standard 62.2-2010 (ASHRAE, 2010), which corresponds to roughly 40 L/s or 0.1 h^{-1} for this building. The airflow through the HRV was measured periodically using a hot wire anemometer (accuracy ±3 % or 0.015 m/s), yielding an average flow of 56 L/s. The airflow of the HRV supplies and returns were measured using a balometer, with a stated uncertainty from the manufacturer of ±3 % plus 2.5 L/s. Table 2 summarizes these measurements, along with the exhaust airflows associated with the kitchen exhaust and the clothes dryer, also measured with the balometer. The sums of the HRV supply and return vents match within their measurement accuracy but are below the values measured at the unit itself, which is likely a reflection of the measurement uncertainties of the flows, particularly for the balometer measurements at the individual vents, as well as the existence of duct leakage. Measuring such low airflow rates in the field is known to be challenging, and a previous study has recommended development of a measurement standard that takes “real world” conditions into account (Stratton et al., 2012). It is worth noting that the measured envelope infiltration rates, even in this extremely tight house, are on the order of 15 % to 40 % of the HRV ventilation rate based on Standard 62.2.

Table 2: Measured System Airflow Rates

Heat Recovery Ventilator	Supply	Return
1st Floor	15	19
2nd Floor	30	27
SUM	45 (0.13 h^{-1})	46 (0.13 h^{-1})
Ducts at HRV unit	56 (0.16 h^{-1})	54 (0.15 h^{-1})
Local exhaust		
Kitchen hood	49	
Clothes dryer	47	

* All flows in L/s except where otherwise indicated.

4 MULTIZONE MODELING

A CONTAM (Walton and Dols, 2013) model was created of the NZERTF using as-built documents and on-site system airflow measurements. On average, the difference between the measured air change rates measured by tracer gas decay and the model predictions were roughly 15 %. Based on the physical theory on which CONTAM and other multizone airflow simulation tools are based, an air change rate of zero is predicted when the wind speed and temperature difference are both zero. As noted above, nonzero air change rates were measured under these conditions and it is not clear whether these were measurement artifacts or reflect actual airflow dynamics. The prediction of air change rates with low driving forces, particularly in such tight houses, is another challenge in characterizing airflow in such houses that merits additional study. However, when considering very tight houses with very low infiltration rates, the accuracy of the predicted rates are generally less important than in leakier buildings.

5 CONCLUSIONS

Infiltration and ventilation of residences has been studied for decades, with trends towards the mantra of “build tight, ventilate right.” The design and construction of the NZERTF was consistent with that philosophy, resulting in a very tight envelope and controlled mechanical ventilation. The airtightness and ventilation measurements in this facility identify challenges for very tight homes, including appropriate airtightness limits and the accuracy of measurements at very low airflow rates. A range of airtightness target values exist, as described in this paper, and it is not clear how tight net zero or near-zero energy, high-performance homes really need to be. Also, even for this extremely tight house, the measured air change rates with the HRV off are still on the order of 15 % to 40 % of the air change rate due to the HRV. Even in this very tight house, the remaining infiltration is nontrivial compared with the intentional ventilation rate supplied in accordance with industry standards. It should also be noted that ventilating right in very tight homes is critical to maintaining acceptable IAQ since such low infiltration rates will not be able to adequately maintain indoor levels of contaminants.

6 ACKNOWLEDGEMENTS

The authors gratefully acknowledge Bill Healy and Mike Lubliner for their insightful reviews of this paper.

7 REFERENCES

ASHRAE (2010). Standard 62.2-2010: Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASTM (2010). ASTM E779-10 Standard Test Method for Determining Air Leakage Rate by Fan Pressurization. Philadelphia: American Society of Testing and Materials.

ASTM (2011). ASTM E741-11 Standard Test Method for Determining Air Change in a Single Zone by Means of a Tracer Gas Dilution. West Conshohocken, PA: American Society for Testing and Materials.

Bernheim, A., P. White and A. Hodgson (2014). High Performance Indoor Air Quality specification for Net Zero Energy Homes. Gaithersburg, MD: National Institute of Standards and Technology. http://dx.doi.org/10.6028/NIST.GCR.14-980. NIST GCR 14-980.

Chan, W. R., J. Joh and M. H. Sherman (2013). Analysis of air leakage measurements of US houses. Energy and Buildings 66(0): 616-625. http://dx.doi.org/10.1016/j.enbuild.2013.07.047.

Dimitroulopoulou, C. (2012). Ventilation in European dwellings: A review. Building and Environment 47(0): 109-125. http://dx.doi.org/10.1016/j.buildenv.2011.07.016.

DOE (2013). DOE Challenge Home (Rev. 03). Washington, D. C.: U. S. Department of Energy.

Elmorth, A. (1980). Building Tight, Ventilating Right. Air Infiltration Review 1(40): 5.
EPA (2015). ENERGY STAR Certified Homes, Version 3.1 (Rev. 06).
Fanney, A. H., V. Payne, T. Ullah, L. Ng, M. Boyd, F. Omar, M. Davis, H. Skye, B. Dougherty, B. Polidoro, W. Healy, J. Kneifel and B. Pettit (2015). Net-zero and beyond! Design and performance of NIST's net-zero energy residential test facility. Energy and Buildings 101(0): 95-109. http://dx.doi.org/10.1016/j.enbuild.2015.05.002.

IARC (2012). Chemical Agents and Related Occupations: Volume 100 F A Review of Human Carcinogens. IARC Monographs on the Evaluation of Carcinogenic Risks to Humans. Lyon, France, International Agency for Research on Cancer. 100 F.

NAHB/ICC (2012). ICC 700-2012 National Green Building Standard. Washington, D. C.: National Association of Home Builders and International Code Council

Omar, F. and S. T. Bushby (2013). Simulating Occupancy in the NIST Net-Zero Energy Residential Test Facility. Gaithersburg, MD: National Institute of Standards and Technology. TN-1817.

Persily, A. K. (2015). Field measurement of ventilation rates. Indoor Air. http://dx.doi.org/10.1111/ina.12193.

Pettit, B., C. Gates, A. H. Fanney and W. Healy (2014). Design Challenges of the NIST Net Zero Energy Residential Test Facility. Gaithersburg, MD: National Institute of Standards and Technology. TN-1847.

PHI (2015). Passive House requirements from http://www.passiv.de/en/index.php.
Poppendieck, D. G., L. C. Ng, A. K. Persily and A. T. Hodgson (2015). Long Term Air Quality Monitoring in a Net-Zero Energy Residence Designed with Low Emitting Interior Products. Building and Environment. http://dx.doi.org/10.1016/j.buildenv.2015.07.001.

Stratton, J. C., I. S. Walker and C. P. Wray (2012). Measuring Residential Ventilation System Airflows: Part 2-Field Evaluation of Airflow Meter Devices and System Flow Verification. Berkeley, CA: Lawrence Berkeley National Laboratory. LBNL-5982E.

Sundell, J., H. Levin, W. W. Nazaroff, W. S. Cain, W. J. Fisk, D. T. Grimsrud, F. Gyntelberg, Y. Li, A. K. Persily, A. C. Pickering, J. M. Samet, J. D. Spengler, S. T. Taylor and C. J. Weschler (2011). Ventilation rates and health: multidisciplinary review of the scientific literature. Indoor Air 21(3): 191-204. 10.1111/j.1600-0668.2010.00703.x.

USGBC (2014). LEED BD+C: Homes | v4 - LEED v4 from http://www.usgbc.org/credits/homes/v4.

Walton, G. N. and W. S. Dols (2013). CONTAM User Guide and Program Documentation. Gaithersburg, MD: National Institute of Standards and Technology. NISTIR 7251. NISTIR 7251.

IMPACT OF AIR INFILTRATION RATES ON MOISTURE BUFFERING EFFECT OF WOODEN SURFACES

Dimitrios Kraniotis*[1], Tormod Aurlien[2], Christoph Brückner[1] and Kristine Nore[1]

1 Norwegian Institute of Wood Technology
Forskningsveien 3B
0373 Oslo, Norway

2 Norwegian University of Life Sciences
Drøbakveien 31, P.O. box: 5003 IMT,
1432 Ås, Norway

**Corresponding author: dimitrios.kraniotis@treteknisk.no*

ABSTRACT

Interior wooden surfaces have the capacity to buffer the maxima and minima of relative humidity (RH) indoors. Especially in high performance buildings, where high airtightness levels as well as high indoor air quality (IAQ) are required, there is great potential for energy savings by reducing the mechanical ventilation demand. The last decade, the moisture buffer phenomena has been widely researched. Relevant findings showed that the moisture buffering effect is reduced when the ventilation rates increase. The air infiltration is usually taken into consideration simply as an additional fraction of air exchange. However, infiltration has a rather complex nature and can result into more localized effects compared to ventilation (mechanical) that mostly applies globally to a room. In this paper, the moisture buffering phenomena linked to the variation of air infiltration rates are studied in a cross laminated timber (CLT) test house. Both CO_2 and moisture are released in the room. The infiltration rates are calculated using tracer gas techniques (CO_2 decay method), while the air exchanges are calculated based on the decay of RH as well and the results are compared. In addition, the indoor relative humidity (RH) and moisture content in the construction elements are measured in different monitoring positions. The impact of airtightness and infiltration rates on the overall ('global') moisture buffer capacity is studied. Furthermore, the influence of the leakage location on the 'local' moisture buffer capacity of each monitoring position / element is also tested. Finally, the potential of using moisture is as 'tracer gas' for estimating the air exchanges in a room is researched.

KEYWORDS

moisture buffering, air infiltration, tracer gas techniques, air exchanges, moisture content

1 INTRODUCTION

The last decade the awareness of influence of hygroscopic materials use in buildings has increased. The moisture buffer capacity has been analysed in several studies (e.g. Simonson *et al.*, 2001; Rode *et al.*, 2005; Osanyintola *et al.*, 2006). The results show that hygroscopic materials hold a great potential to damp the maxima and minima of relative humidity (RH), narrowing the range of the latter. This fact results in reduction of ventilation loads and consequently on the energy savings. In addition, it secures a better indoor microclimate, which is not too humid nor too dry. In particular, Simonson *et al.* (Simonson *et al.*, 2001) showed that when the internal surfaces of a wooden apartment building were permeable, the maximum indoor RH was lower compared to the impermeable case assumed (impermeable paint). In addition, the RH dropped below 20% for less period of time compared to the impermeable case.

The potential indirect savings from adjusting the ventilation rate and indoor temperature while maintaining adequate indoor air quality and comfort are in the order of 5 % for heating while they range from 5% to 20% for cooling. Woloszyn *et al.* (Woloszyn *et al.*, 2009) confirmed that the use of gypsum-based moisture-buffering materials is an efficient way combined with a relative humidity sensitive (RHS) ventilation system could reduce the mean ventilation rate of 30% – 40% and generate 12% – 17% of energy savings in the cold period. It was even possible, by the combined effect of ventilation and wood as buffering material to keep the indoor RH at a stable level, between 43% and 59%.

One of the 'non-material' parameters that affect the moisture buffer capacity of surfaces is ventilation by means of both mechanical and natural (Simonson *et al*., 2004). Higher ventilation rates result in reduced buffering effect (Yoshino *et al.*, 2009). Usually, the infiltration is taken into account only as a constant fraction of the total air exchanges (Rode *et al.*, 2008) or it is even neglected for simplification reasons (Yang *et al.*, 2014). Yang *et al.* (Yang *et al.*, 2012) and Li *et al.* (Li. *et al.*, 2012) have included infiltration in their models when evaluating moisture buffer capacity of wooden interior surfaces in an environmental chamber. Moreover, the dynamic nature of air infiltration have been recognized (Haghighat *et al.*, 2000). Taking into account the natural unsteady characteristics of air infiltration and their impact on estimating better the air exchanges has been presented (Kraniotis *et al.*, 2014).

This study explores the impact of in-situ-measured infiltration rates on the moisture buffer capacity (drying process) of hygroscopic surfaces. The experiment has been conducted in a cross-laminated-timber (CLT) test house with controlled leakages. A previous study conducted in the same test house showed how the mechanical ventilation rates affect the levels of indoors RH and the moisture buffering in the wooden structure (Katavic *et al.*, 2014). In the current paper, the focus is on the influence of air infiltration on the moisture buffer capacity, and thus the mechanical ventilation is shut down. The study compares the 'global' (overall) infiltration rate with the corresponding decay of the indoor RH. Carbon dioxide (CO_2) is used as tracer gas and the concentration decay method for calculating the infiltration rates (exfiltration). Furthermore, localizing the leakages allows the deeper investigation of the influence of air infiltration on the moisture buffer capacity of the wooden surfaces. The air exchanges of the test house are estimated using both traditional tracer gas techniques (CO_2) as well as the decay of the RH levels indoors. Thus, an evaluation of the potential of using moisture as 'tracer gas' is presented.

2 METHODOLOGY

2.1 Experimental site

The experimental facilities consists of a test house located in a meteorological field, affiliated to the Norwegian University of Life Sciences (NMBU), Ås, Norway. The single-compartment test house is constructed of CLT made of spruce and is insulated with mineral wool. The thickness of the walls is 100mm, while the ceiling is 140mm thick. The test house is externally insulated; 100mm mineral wool in the South and East wall, 150mm in the North and West and 250mm in the roof. Two layers of open-diffusive weather resistant barriers (wind barriers) have been placed; one on the interior side of the insulation layer and one on the exterior side. The roof has only the internal layer of wind barrier. Fig. 1 shows a cross section of the South wall of the test module.

The floor has 14 mm oak parquet over 22 mm chipboard and 100 mm mineral wool insulation. The modules are placed on top of 200 mm Rockwool with vapour barrier inhibiting any moisture to penetrate from the ground. The internal dimensions of the modules are 7.0m x 3.6m, while the net height of the room is 2.24m (Fig. 2a, 2b).

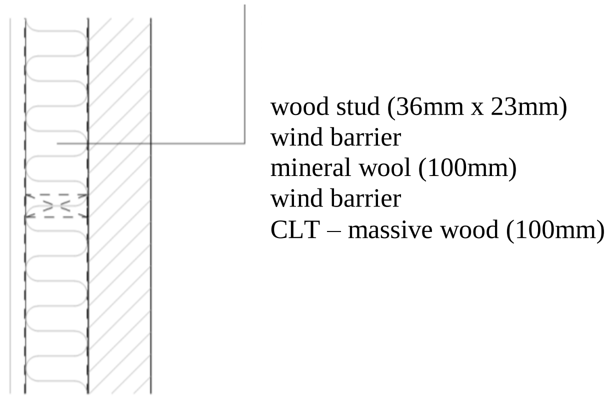


Figure 1: Cross section of the external wall (South) of the test house.

The leakage area of the room consists of 8 holes, of circular section of diameter δ = 16mm (A $\approx$ 2cm^2), that cross the whole assembly of the South and the North walls. They are located as 'pairs' in four positions of the walls. The leakages are noted as e_1 – e_8 and are shown in Fig. 2a and 2b. The lower ones, i.e. e_1, e_3, e_5 and e_7, are 120mm above the floor, while the upper ones, i.e. e_2, e_4, e_6 and e_8, are 270mm above the floor (Fig. 3a).

2.2 Structure of experiment

The experiment consisted of 3 phases:

- Phase 1: the moisture release (moisture generation) and mixing – wetting phase,
- Phase 2: the CO_2 release and mixing and
- Phase 3: the 'decay' phase – drying.

The moisture release lasted for 8 hours and a humidifier, in the centre of the room (noted by b in Fig. 2a, 2b), was used for the moisture generation in a constant rate (375g/h). In order to ensure sufficient moisture mixing and thus uniformity within the room, two fans were used located diametrically at the NE and the SW corner of the test house, noted as c_1 and c_2 in Fig. 2a and 2b. During this phase the leakages were hermetically sealed, thus it is assumed that the released moisture migrates and is getting absorbed by the hygroscopic CLT panels.

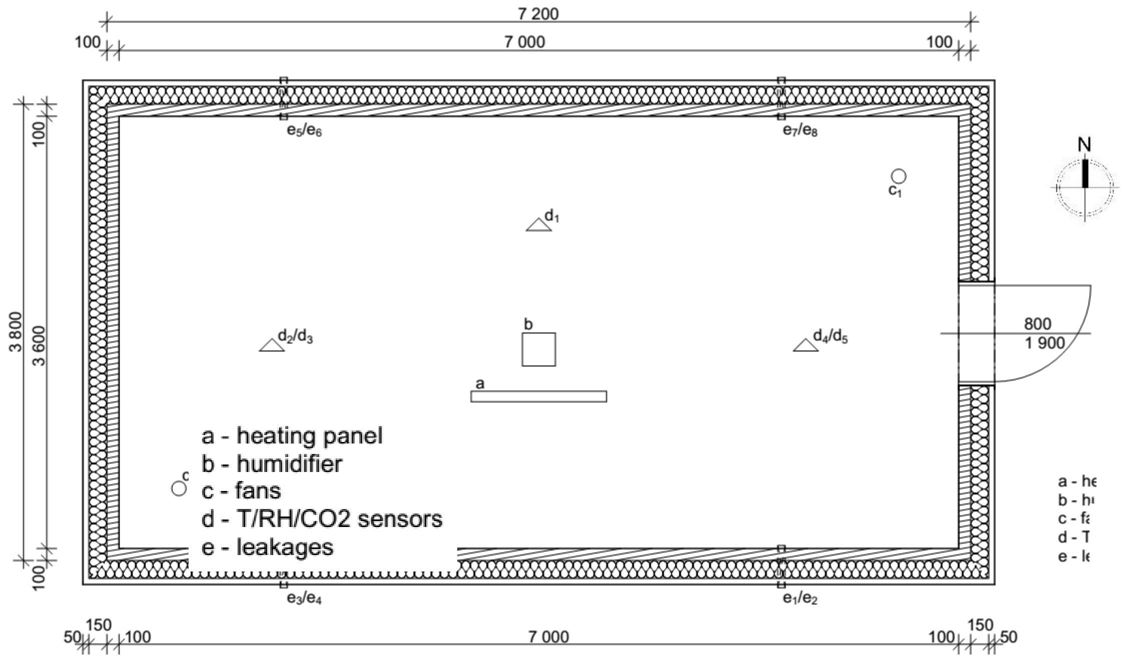
After 8 hours, the humidifier was turned off and the CO_2 was released. The fans were working for 10min more in order to ensure the sufficient mixing of the gas and guarantee that the required uniformity (ASTM – E741-06, 2006).

The previous short phase follows the 'decay phase' and the drying out of the wooden structure. The CO_2 concentration decay method was used to calculate the infiltration rates. In order to ensure reliability of the rates, the duration of this last phase was determined as 24h, respecting the requirements of the relevant standards.

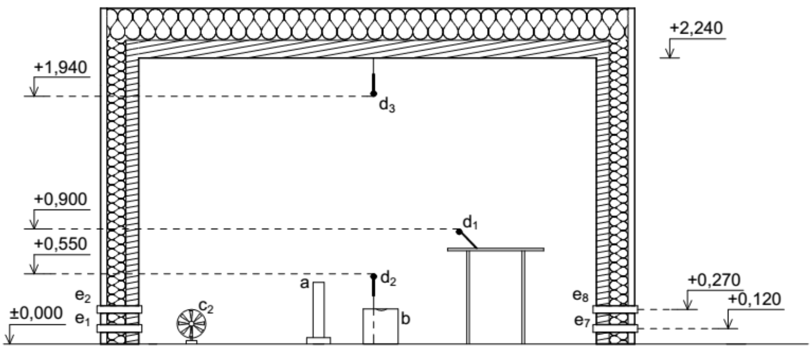
In total, the experiment consists of six measurements. Three experiments were conducted will all the eight holes open during the phase 3 (decay – drying) and they are noted as A_1 – A_3. In addition, three measurements were executed with only the four holes open, i.e. e_1, e_2, e_7 and e_8, means the holes located at the 'east' part of the room. The rest of the holes remained sealed, i.e. e_3, e_4, e_5 and e_6. These three measurements are noted as $A_4 – A_6$.

2.3 Instrumentation

The CO_2 concentration was measured and recorded using CO_2 loggers that had been placed in five positions, i.e. $d_1 – d_5$ in Fig. 2a and 2b. The loggers have also the possibility to measure and log temperature T and relative humidity RH. The current set-up aims: i) to confirm the sufficient spatial distribution of CO_2 and moisture by monitoring the [CO_2] and the indoors RH in five different positions (phase 1 and 2) and ii) to detect possible differences during the last phase 3, as the loggers had been strategically placed, i.e. d_1 at the centre of the room, d_2 - d_3 close to the leakages at the 'west' part of the room and d_4 - d_5 close to the ones at the 'east' part. In addition, the loggers d_2 and d_4 have been placed on a lower height, i.e. 550mm, while the d_3 and d_5 in a higher height, i.e. 1940mm. The logger d_1 is on level 900mm above the floor at the centre of the room (Fig. 2b).



(a)



(b)

Figure 2: (a) Plan view and (b) cross section of the test house, where schematic representation and positions of the instruments used are shown.

In addition, 15 sensors that measure the equilibrium moisture content (EMC) in the CLT panels have been mounted in characteristic positions, i.e. 'aligned' with cross sections of the leakages and at a lower and medium height (750mm and 1500mm respectively). The sensors also record also T and RH locally. They are noted by 'H_1-H_{15}' and the exact positions are shown in Fig. 3.
Finally, the test house is equipped with an electric heating system to maintain the indoor temperature at 20°C. Technically an extract ventilation system exists, but the since the objective of the study was to investigate the phenomena linked to infiltration, the ventilation was shut down and the exhaust was sealed throughout the experiment.

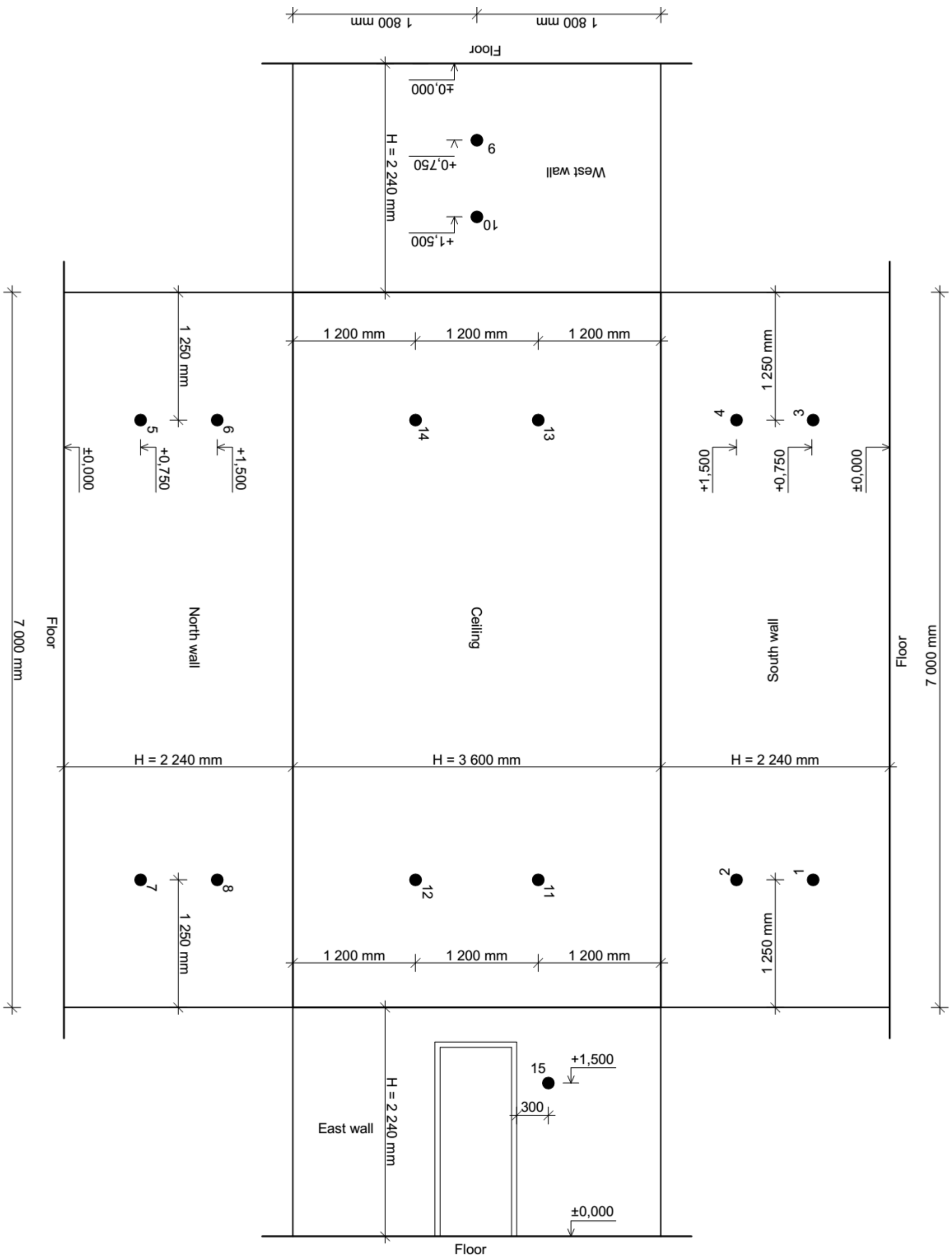


Figure 3: Plan view of the ceiling and view of the four walls that show the positions of the T/RH/EMC sensors.

2.4 Calculation of infiltration rates (decay of [CO_2])

The infiltration rates were calculated based on the relevant methodology described in the standard ASTM – E741-06 regarding the 'concentration test decay method' (ASTM – E741-06, 2006). For each day, the initial and final times are determined and a normalized concentration C_N is calculated in each measurement (Roulet *et al.*, 2002):

$$C_N = \frac{C(t) - C_o}{C(0) - C_o} \tag{1}$$

where C(t) the average CO_2 concentration in the test room, C(0) the initial CO_2 concentration in the test room, C_o the outdoors CO_2 concentration (≈ 400ppm) and t the time.
Since the CO_2 concentration fluctuating as decaying the 'average method' cannot be applied. Thus, in order to compute the ACH, the 'optional regression method' is used and a regression of ln*C(t)* needs to be performed according to (ASTM – E741, 2006):

$$ln\, C_N = -At + ln\, C_N\,(0) \tag{2}$$

The infiltration rates are represented by the fit line calculated after the regression analysis and in particular by the slope of the lnC_N against the time.

2.5 Decay of indoor RH

The decrease of the RH was also recorded simultaneously with the decay of the CO_2 concentration. During the last phase 3, the decline of the moisture content in the room is caused by the air leakages. To calculate the air exchanges using the reading of humidity, the vapor concentrations indoors and outdoors are needed. The volumetric concentration is given as:

$$Cv = \frac{p_w}{P} \tag{3}$$

where C_v is the concentration of water in air (as a ratio of water vapor volume to total wet air volume), p_w the partial water vapor pressure and P the total system pressure.
The RH is defined as the ratio of water vapor pressure p_w to the saturation water vapor pressure (p_{ws}) at the gas temperature:

$$RH = \frac{p_w}{p_{ws(T)}} 100\% \tag{4}$$

$$p_w = RH * p_{ws(T)} * 100\% \tag{5}$$

The $p_{ws(T)}$ is calculated as follows:

$$p_{ws(T)} = e^{(77.3450 + 0.0057T - \frac{7235}{T})} / T^{8.2} \tag{6}$$

Assuming the atmospheric pressure as the total system pressure P, the C_v calculation is feasible for both indoors and outdoors, i.e. $C_v(t)$ and C_{vo}, using the T and the RH inside the room and the ambient air respectively. A normalized concentration C_{vN} is calculated for each measurement:

$$C_{vN} = \frac{C_v(t) - C_{vo}}{C_v(0) - C_{vo}} \tag{7}$$

In analogous way to the previous infiltration rate (CO_2 decay), the current 'moisture-based' infiltration rate is a natural logarithm function over time. The slope of the function in a semi-log graph gives the infiltration rate.

3 RESULTS

The whole measurement period for the day A_5 is presented as example in Fig. 4. The wetting phase (phase 1) is clear, which lasted for 8h, as well as the phase of decay (phase 3). The RH increased during moistening of the room, while the concentration of CO_2 stayed stable, just somewhat higher than the outdoors levels. After the phase 1, the humidifier was turned off and the release/mixing of CO_2 started (phase 2). Right after only 10min, the phase 3 started.

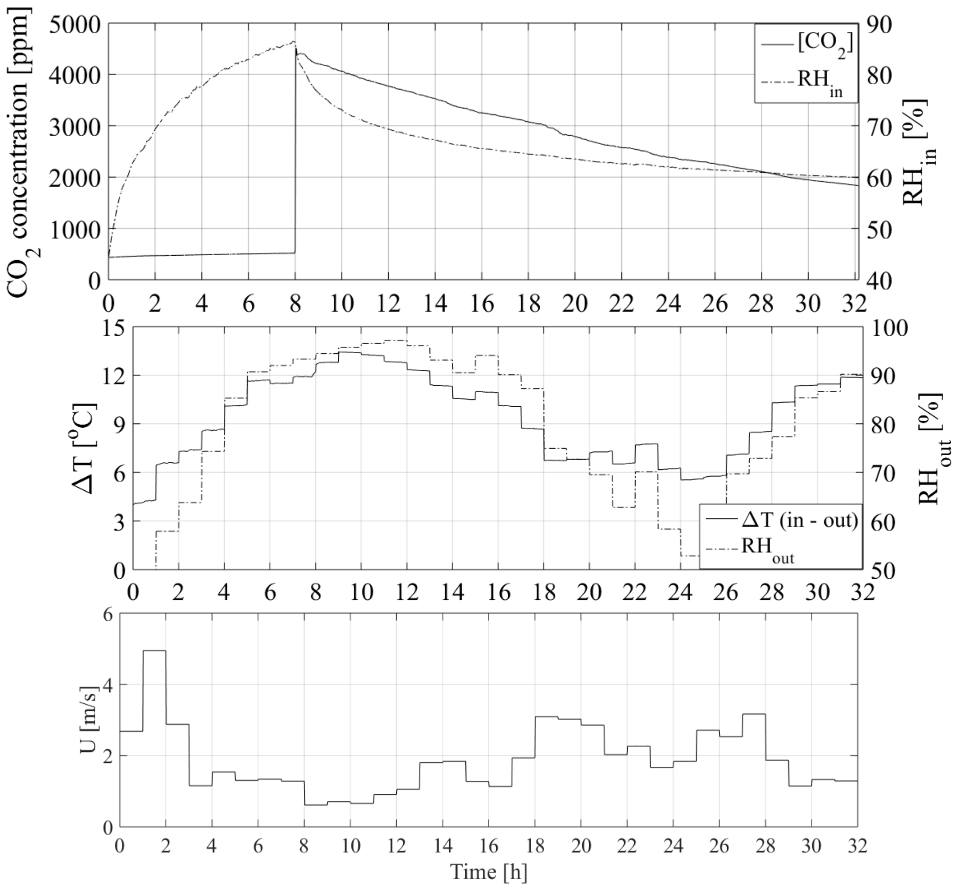


Figure 4: Overview of whole measurement period for test A_5. The variation of $[CO_2]$ and RH_{in} is shown as well as the time series of RH_{out}, temperature difference ΔT and wind speed U.

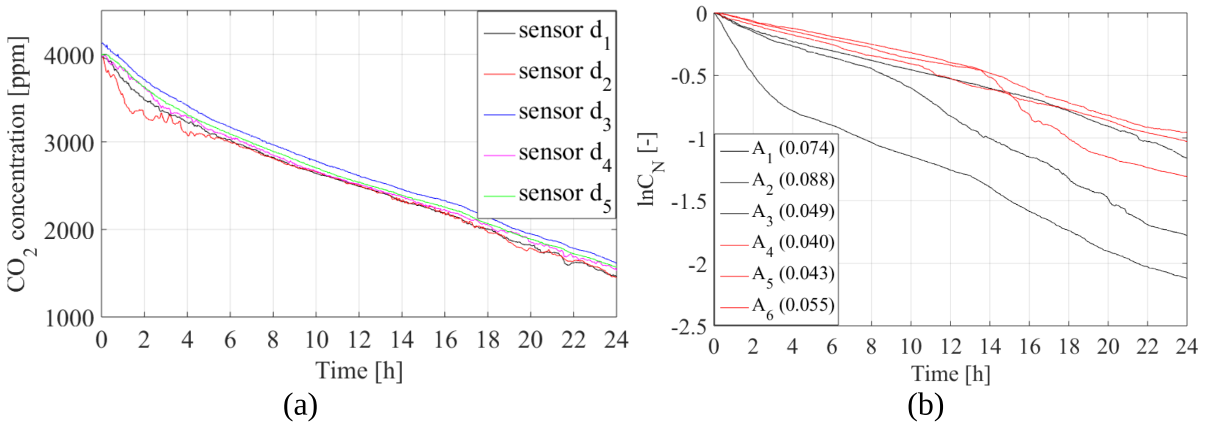


Figure 5: (a) The decay of $[CO_2]$ as recorded at the five positions ($d_1 - d_5$) during the test A_3. (b) The infiltration rates (averaged positions) calculated for the six measurements ($A_1 - A_6$).

Table 1: Infiltration rates based on the CO2 decay and the decay of the indoors RH.

Measurement	**Status of holes**	**ACH@50Pa**	**ACH n_{CO2} (CO_2)**	**ACH n_{moist} (moisture)**
A_1	all holes open (8x2 = 16cm^2)	1.1	0.074	0.040
A_2	all holes open (8x2 = 16cm^2)	1.1	0.088	0.024
A_3	all holes open (8x2 = 16cm^2)	1.1	0.049	0.068
A_4	only e_1-e_2-e_7-e_8 open (4x2 = 8cm^2)	0.5	0.040	0.023
A_5	only e_1-e_2-e_7-e_8 open (4x2 = 8cm^2)	0.5	0.043	0.034
A_6	only e_1-e_2-e_7-e_8 open (4x2 = 8cm^2)	0.5	0.055	0.030

In Fig. 5a the recordings from the sensors d_1-d_5 are shown. The results from the sensors d_3 and d_5 reveal that the CO_2 decay at the area of the ceiling is smoother than on the level of 55cm above the floor, i.e. d_2 and d_4, which are located closer to the leakages. The findings are consistent with previous studies, showing that the phenomena are more fluctuating at the neighbourhood of leakages (Kraniotis, 2014).

The logarithm of the normalized CO_2 concentration is shown against the time in Fig. 5b. The calculated infiltration rates based on the CO_2 decay, n_{CO2} are also depicted in Table 1. When all the 8 holes were open (A_1-A_3), n_{CO2} tends to be higher compared to the days that four of the holes were sealed (A_4-A_6). In addition, the tighter the room was, the smaller the variations among the infiltration rates.

In Table 1 the ‘moisture-based’ infiltration rates n_{moist} are also presented. The n_{moist} show smaller values compared to n_{CO2}. It would be reasonable to claim that n_{moist} can represent that in the cases A_4-A_6 the room was tighter, however the agreement between n_{CO2} and n_{moist} is rather not consistent. In contrast to the outdoors CO_2 levels, the vapour pressure and concentration outdoors are not constant (Fig.6). Thus, a ‘vapours injection’ from outdoors to indoors takes place even during the decay of indoors moisture making the calculation of ‘moisture-based’ infiltration rates more complex.

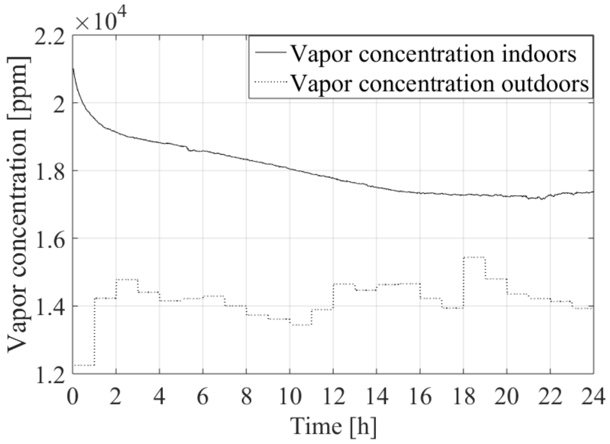


Figure 6: The vapour concentration indoors and outdoors during the phase 3 of test A_6.

In order to study the influence of local in- and exfiltration on moisture contents in the test house elements, the MC in some characteristic positions is shown (Fig. 7). The first two graphs show the results from the test A_3, in which all the 8 holes were open during the phase 3. All the four sensors presented are next to leakages and a similar behaviour is shown. However, the levels of MC is very low and it is rather ambitious to draw concrete conclusions based on the current findings.

The other two graphs of Fig. 7 present the some sensors from the test A_5. It is important to note that the tests A_3 and A_5 have similar infiltration rate calculated, i.e. 0.049 and 0.043 respectively, despite the fact that in A_3 8 holes were open while in A_5 only 4 of them. The sensors next to open holes (leakages), i.e. S_6 and S_{11}, have similar behaviour as before. The comparison of S_6 and S_{11} though with the sensors S_8 and S_{13} that are located next to sealed holes during the test A_5, show small differences among the results. In particular, the sensors S_8 and S_{13} have a rather smoother decay compared to S_6 and S_{11}, revealing that the local infiltration phenomena is likely to have impact on the moisture drying process in hygroscopic surfaces. However, the fact that the room is quite tight results in narrowing down the influence. In addition, the low levels of MC makes more difficult to highlight the differences.

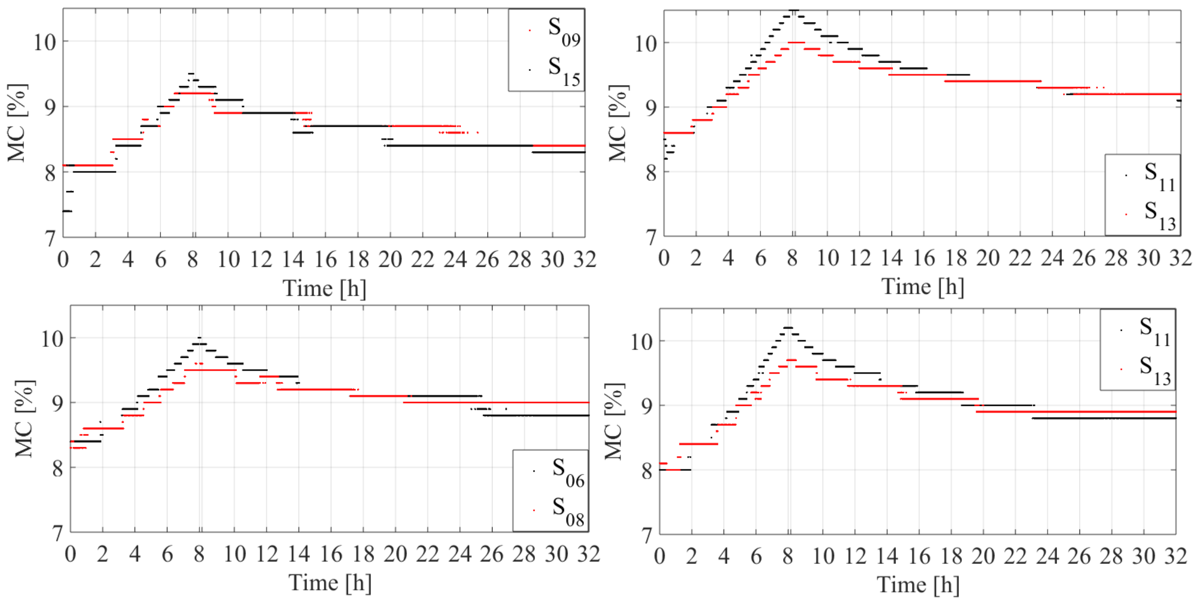


Figure 7: Moisture content MC in various positions (a) west wall and door in test A_3. (b) ceiling in test A_3. (c) north wall in test A_5. (d) ceiling in test A_5. (the detailed position of the sensors is shown in Fig. 3)

4 CONCLUSIONS

The current paper elaborates with simultaneous decay of CO_2 and vapours in a relatively tight and insulated cross laminated timber (CLT) test house, in order to investigate the influence of infiltration on moisture buffer (drying process). The results show that it is likely that the moisture content dries out faster in positions closer to leakages compared to the positions that are not close to them. Even when the global infiltration rates between two case-studies are very similar, differences on the moisture content can be observed based on where the leakages are located. However, the differences in the current set of experiments presented were rather small. The fact that the room is quite tight results in narrowing down the influence of in- and exfiltration on moisture buffer capacity. In addition, the low levels of MC makes more difficult to highlight these differences. The authors believe that the impact would be more clear in a looser building, where infiltration phenomena play significant role.
Moreover, the infiltration rates were calculated using both CO_2 and moisture (water vapours) as tracer gases. The result showed that using water vapours as tracer gas might be more complex because, in contrast to the outdoors CO_2 levels, the vapour pressure and concentration outdoors are not constant. Thus, a 'vapours injection' from outdoors to indoors takes place even during the decay of indoors moisture. Further elaboration with the latter could be of interest.

5 ACKNOWLEDGEMENTS

The authors want to thank the Norwegian Research Council (NRC) for funding this study in the context of the research project HOME – Holisitc Monitoring of Indoor Environment. The authors gratefully acknowledge Signe Kroken from NMBU for providing the meteorological data of Ås. Kolbjørn Mohn Jenssen, director of Mycoteam, is also acknowlegded for his valuable contribution to the execution of experiment.

6 REFERENCES

ASTM International E741-06. (2006). Standard test method for determining air change in a single zone by means of tracer gas dillution.

Haghighat, F., Brohus, H., Rao, J. (2000). Modelling air infiltration due to wind fluctuations – A review. *Building and Environment*, 35, 377-385.

Kraniotis, D., Aurlien, T. Thiis, T.K. (2014). Investigating instantaneous wind-driven infiltration rates using the CO_2 concentration decay method. *Int. Journal of Ventilation*, 13(2), 111-123.

Li, Y., Fazio, P., Rao, J. (2012). An investigation of moisture buffering performance of wood panelling at room level and its buffering effect on a test room. *Building and Environment*, 47, 205-216.

Osanyintola, O.F., Simonson, C.J. (2006). Moisture buffering capacity of hygroscopic building materials: Experimental facilities and energy impact. *Energy and Buildings*, 38, 1270-1282.

Rode, C., Peuhkuri, R., Mortensen, L.H., Hansen, K.K., Time, B., Gustavsen, A., Ojanen, T., Ahonen, J., Svennberg, K., Harderup, L.E,, Arfvidson, J. (2005), *Moisture buffering of building materials.* Project No.: 04023. Nordic Innovation Centre.

Rode, C., Grau, K. (2008). Moisture buffering and its consequence in whole building hygrothermal modeling. *Journal of Building Physics* 31, 333-360.

Roulet, C.A., Foradini, F. (2002). Simple and cheap air change rate measurement using CO_2 concentration decays. *Int. Journal of Ventilation*, 1 (1), 39-44.
Simonson, C.J., Salonvaara, M., Ojanen ,T. (2001). *Improving Indoor Climate and Comfort with Wooden Structures*. Technical Research Centre of Finland, VTT Publications 431. Espoo, Finland.
Simonson, C.J., Olutimayin, S., Salonvaara, M., Ojanen, T., O'Connor, J. (2004). *Potential for Hygroscopic Building Materials to Improve Indoor Comfort and Air Quality in the Canadian Climate.* ASHRAE Conference Proceedings 2004.
Yang, X., Fazio, P., Ge, H., Rao, J. (2012). Evaluation of moisture buffering capacity of interior surface materials and furniture in a full-scale experimental investigation, *Building and Environment*, 47, 188-196.
Yang, X., Ge, H., Fazio, P., Rao, J. (2014). Evaluation of parameters influencing the moisture buffering potential of hygroscopic materials with BSim Simulations. *Buildings*, *4(3)*, 375-393.
Yoshino, H., Mitamura, T., Hasegawa, K. (2009). Moisture buffering and effect of ventilation rate and volume rate of hygrothermal materials in a single room under steady state exterior conditions. *Building and Environment*, 44, 1418-1425.
Woloszyn, M., Kalamees, T., Abadie, M.O., Steeman, M., Kalagasidis, A.S. (2009). The effect of combining a relative-humidity-sensitive ventilation system with the moisture-buffering capacity of materials on indoor climate and energy efficiency of buildings. *Building and Environment*, 44, 515-524.

NUMERICAL EVALUATION OF THE AIRTIGHTNESS IMPACT ON AIRFLOW PATTERN IN MECHANICALLY VENTILATED DWELLINGS IN FRANCE

F. Richieri[1], B. Moujalled[2], T. Salem[3], F.R. Carrié[4]

[1] *SNIA - Rue 12 avenue Pythagore - BP 70285 - 33697 MERIGNAC CEDEX – France – Email : fabrice.richieri@aviation-civile.gouv.fr*

[2] *CEREMA – DTer Sud-Ouest - Rue Pierre Ramond - BP 10 33166 Saint-Médard-en-Jalles cedex – France*

[3] *Notre Dame University – Louaize - P.O. Box 72, Zouk Mikael Zouk Mosbeh – Lebanon*

[4] *ICEE, 93 rue Molière, 69003 Lyon – France*

ABSTRACT

The objective of this paper is to assess the impact of the envelope airtightness on airflow patterns for single detached dwellings depending on the ventilation system.
A numerical approach based on the energy simulation tool TRNSYS coupled to a multi-zone air-flow and contaminant transport model COMIS was used to analyze the local and global performance of four mechanical ventilation strategies in terms of energy and ventilation efficiencies. The following ventilation strategies were modelled: constant flow exhaust-only, humidity-sensitive exhaust-only with and without humidity-controlled inlets, and constant flow supply and extract. The multi-zonal approach allows studying the local disturbances in the functioning of these systems, which differs from most studies on the subject. The impacts of the location of leakages in the building envelope (facades and/or ceiling) were also investigated. Numerical results showed a significant impact of air permeability on the ventilation efficiency, with a different impact depending on the mechanical ventilation system. The wind effect exposes the building more frequently to the influence of crossing airflow, and as a result the normal distribution of fresh air from main rooms to wet rooms could be disturbed, with a risk of damaging indoor air quality. We also analysed the fresh air origin, by quantifying the amounts of airflows from leakages and inlet/supply units for each ventilation systems. Results show that in the case of humidity-sensitive ventilation systems, a specific attention should be given to the air permeability in order to maintain the desired airflow pattern from main rooms to service rooms.

KEYWORDS

Simulation, airflow displacement, airtightness, ventilation system.

1. INTRODUCTION

As building energy efficiency is becoming a major issue to reduce energy use and greenhouse gas emission, energy regulations are evolving towards more stringent rules at the national and

European level. The European energy efficiency directive requires all new buildings to be nearly zero-energy by 2020. With the increased levels of fabric insulation, airtightness of the building envelope is expected to play an increasingly important role in buildings energy efficiency (Pan, 2010). However, energy saving targets in buildings must be achieved while maintaining good indoor air quality and comfort level. Therefore, the ventilation is a key concern in energy efficient buildings. It allows outdoor air to be intentionally provided to a space while stale air is removed. Ventilation is essential for securing acceptable air quality in buildings by diluting and removing indoor pollutants. At the same time, it accounts for 30% or more of space conditioning energy demand (Liddament, 1996). Ventilation standards and regulations set the requirements for the minimum ventilation rates, with a possibility to reduce airflow during unoccupied period (Richieri, 2007). The ventilation needs can be achieved by natural or mechanical means. In countries with colder climates, mechanical systems are mostly used, which are either Exhaust-only or balanced, with or without heat recovery units (Dimitroulopoulou, 2012). In France, the following mechanical systems are typically used: Exhaust–only ventilation, balanced ventilation units with heat recovery, humidity sensitive exhaust ventilation, either integrated to air outlets (called “Ventilation Hygro A”), or to both air inlets and outlets (called “Ventilation Hygro B”).

These systems ensure a permanent displacement flow of fresh air into the building. Air inlets are located in main rooms, which can be adjustable or self-adjustable, but cannot be blocked, and air exhausts in wet room, such as kitchen, bathroom(s), toilet(s). Ventilation system must be able to ensure a minimum exhaust airflow rates greater than a limit depending on number of main rooms. The siting of the inlets and outlets is intended to have the air move from main rooms to service parts (wet rooms), so that moisture and other pollutants are removed at source and the whole dwelling is ventilated (Durier, 2008)

Humidity-sensitive ventilation systems are generally based upon the modulation of the air-cross section – and so of the airflow – at the air inlets and outlets units level, using a mechanical sensor which directly drives the shutter set in the air stream. These systems can reduce the amount of the exhaust airflow to a low level of background ventilation during unoccupied periods or low activities. Several studies have confirmed that relative humidity sensitive ventilation is a good way of reducing building energy demand in residential buildings as the mean ventilation rate is reduced when the building is not used (Woloszyn, 2009). They are nearly always used in new French residences, as they appear to be cost-effective solutions to improve the energy performance and make it easier to meet current French energy regulation requirements.

However, as for balanced ventilation, humidity sensitive ventilation systems induce globally a weaker under-pressure that can be lower than the weather induced pressure (Liddament, 1996). Therefore the total (and local) air change rate could be more subjected to variations due to the air infiltration through the building envelope. Unlike ventilation process, air infiltration is the uncontrolled airflow into the building through unintentional gaps and cracks in the building envelope (Liddament, 1996) (Dimitroulopoulou, 2012).

The air infiltration rates depend on the airtightness of the envelope, the wind speed and direction, and the air temperature (stack effect). When the pressures acting across air inlets of the ventilation systems located in the building envelope are dominated by weather conditions, these inlets can also become routes for undesired air infiltration. This additional outdoor air entering the building will lead to an increase of energy consumption (Pan, 2010) (Chan, 2013). Also, it may distort the intended airflow pattern resulting in parts of the house being inadequately ventilated and leading to poor air quality and a risk of moisture problems (Liddament, 1986).

Thus a poorly implemented ventilation strategy, combined with an inappropriately designed building envelope, may adversely affect both energy efficiency and indoor climate (Pan,

2010) (Liddament, 1996). Airtightness improvement in mechanically ventilated building allows a better indoor air quality (QUAD BBC; 2009).

Until now, models developed to calculate the impact of air permeability on heating demand used calculation methods based on annual average air-infiltration EN ISO 13789 (EN ISO 13789, 2007) or on an hourly calculation of standard EN NF 15242 13789 (EN NF 15242, 2007) also used by French regulation calculation method with a monozonal configuration. These both methods do not take into account the dynamic behaviour of the local humidity-controlled air inlets and outlets in main and service rooms as function of wind exposure and envelope airtightness. As a result, both methods lead to substantially different results.

Our work aims to go one step further by evaluating the airtightness impact on energy needs (Richieri, 2013), as well as airflow patterns whose topic is especially discussed in this paper. A dynamic simulation tool of a multi-zone building was developed, including controlled-loop of air inlets and outlets for different ventilation systems. The relationship between airtightness, ventilation system is investigated. Also the impact on air flow pattern is discussed.

2. METHODOLOGY

2.1. Studied configuration

Building description

The building considered is a typical French single-detached dwelling with 119 m² of living area, and inhabited by a 4-person household (a couple and two children). Thermal and airflow zoning covers 11 connected areas according to the following detail:

- **main rooms**: Living-room; Office; 3 bedrooms;
- **service rooms**: Kitchen; Bathroom; WC;
- **auxiliary rooms**: storeroom (without water point), corridor;
- **highly ventilated attic**.

Figure 1 shows the plan view of the dwelling. Main thermal properties of walls and windows are described in Table 1.

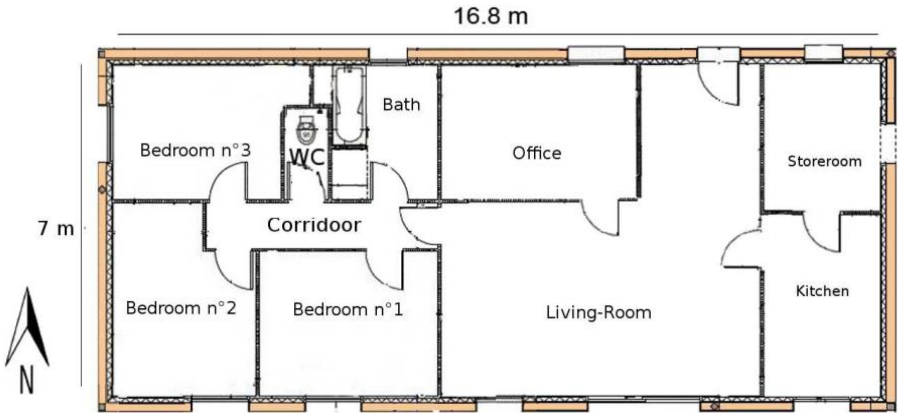


Figure 1: Plan of the dwelling

Table 1: Envelope composition

Wall	Composition	U value W/(m^2.K)
External Wall	20 cm hollow brick + 10 cm glass wool insulation	0.21
Upper Ceiling	1.3 cm gypsum + 27 cm glass wool insulation	0.14
Floor	4 cm concrete + 9 cm Polyurethane + 12 cm concrete	0.24
Living Room Windows	Low emissivity double-glazing with aluminium frame	1.7
Other Windows	Low emissivity double-glazing with PVC frame	1.4
Pitched Roof	Clay roof tile (High ventilated attic)	5.6

The dwelling is supposed to be heated by radiator and thermostatic valves (set à 20.4°C – corresponding to the French regulation reference) from 1st October to 20th May during occupied period (18h-9h on weekdays and on weekends). The internal temperature is 3°C reduced during unoccupied period (Monday – Fridays from 09h to 18h)

Ventilation systems description

Four ventilation systems are modelled in accordance with national requirements on ventilation as indicated in Table 2. The first one is “self-adjusting mechanical exhaust ventilation”. Service rooms are equipped with self-adjusting air-outlets, and main rooms by self-adjusting air-inlets integrated in facades. The second is mechanical balanced ventilation with an 80% efficiency heat exchanger. The exhaust airflow rates are the same as the self-adjusting mechanical exhaust ventilation. The pre-heated fresh air is distributed in main rooms by air supply units. Finally, two strategies of mechanical humidity-controlled ventilation are modelled following technical agreement specifications (CSTB, 2014). “Humidity-controlled Hygro A” is composed of self-regulated air-inlets in main rooms integrated on facades and humidity-controlled air-outlets in service rooms. For “Humidity-controlled Hygro B”, both air-inlets and air-outlets are relative humidity controlled. All mechanical systems enable boosting airflow rate in the kitchen (135 m^3/h) during cooking.

Table 2: Ventilation systems characteristics

System	Inlets				Outlets				
	Type	Living room	Bedrooms & office	Total	Type	Kitchen	Bathroom	Toilet	Total
Self-adjusting exhaust ventilation	Facade air inlets	2 x (22 m^3/h)	30 m^3/h	164 m^3/h	Air exhaust units	45 m^3/h	30 m^3/h	30 m^3/h	105 m^3/h
Balanced ventilation	Air supply units	35 m^3/h	18 m^3/h	107 m^3/h	Air exhaust units	45 m^3/h	30 m^3/h	30 m^3/h	105 m^3/h
Humidity-controlled Hygro A	Facade air inlets	2 x (22 m^3/h)	30 m^3/h	164 m^3/h	Air exhaust units	12-45 m^3/h at RH 50-83%	10-45 m^3/h at RH 25-60%	30 m^3/h	52-120 m^3/h
Humidity-controlled Hygro B	Facade air inlets	2 x (6-45 m^3/h) at RH 45-60%	6-45 m^3/h at RH 45-60%	36-270 m^3/h at RH 45-60%	Air exhaust units	10-45 m^3/h at RH 24-59%	10-40 m^3/h at RH 36-66%	5 m^3/h (30 m^3/h; 30’)	25-90 m^3/h

2.2. Simulation tools

Our approach is based upon the coupled thermal and airflow network model, using the commercially available simulation tools TRNSYS (building energy simulation) and COMIS (airflow network) (Feustel, 1999). It allows us to calculate heat transfer, airflow and pollutant transport (e.g., moisture) in a multi-zone building under transient boundary conditions. Besides, it enables dynamic modelling of closed-loop control for different systems (especially humidity sensitive ventilation system). TRNSYS has been validated for the calculation of transient heat exchange through the surfaces composing a zone (Voit et al., 1994). Also, COMIS has been validated for air and contaminant flows resulting from infiltration through cracks and ventilation systems (Furbringer, 1996).

Moisture generation

Moisture vapour is produced through activities such as cooking, showering or washing/drying clothes, or through metabolic processes such as respiration and sweating. Moisture is produced at various times according to occupation in different locations and in variables quantities. More generally, elevated levels of humidity are related with other pollutants such as CO_2 (produced by the metabolism) and cooking odours.

Consequently, moisture vapour and sensible heat schedules were created for each zone according to the activities of a 4-person family (2 adults and 2 children) - activities related to human metabolism and domestic. For each schedule, the amount of water vapour and sensible heat were calculated hour by hour. Table 4 presents the values of the heat and moisture ratios of the activities that have been considered in the calculation.

Table 4: Humidity ratio of different sources

Sources	**Humidity ratio**
Occupant metabolism : sleeping	15.2 kg/week
Occupant metabolism : seated at rest	
Occupant metabolism : seated light activity	
Cooking	11.0 kg/week
Shower	8.4 kg/week
Drying cloths	7.0 kg/week
Floor cleaning	0.5 kg/week

The humidity ratios represent the total amount of moisture per week that has been calculated from ratios mentioned in (Furbringer, 1996) and (Brogat, 2003). For example, a ratio of 300 g/person for a shower is used. In our case, we have considered in the bathroom 4 showers per day – 2 in the morning and 2 in the evening. Thus, an emission of 0.195 g/s has been defined in the bathroom from 06:00 till 07:00 and from 18:00 till 19:00 during weekdays. Figure 2 illustrates the sum of total moisture emission schedules for all zones for weekdays and weekend (Fig2).

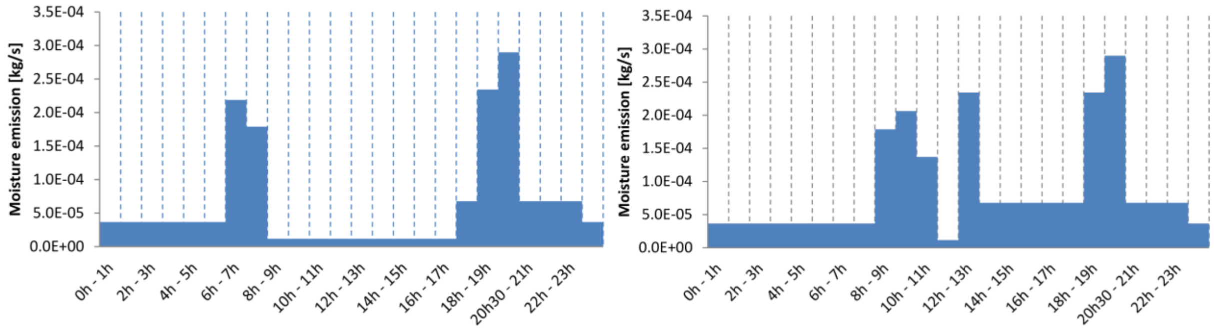


Fig. 2: Sum of total moisture emission schedules for all zones during Weekdays (left) and Weekends (right)

Flow paths through building components

The building is considered as a zone network linked by airflow components such as cracks on envelope, inlets and outlets of ventilation systems, and internal circulations between 11 zones. For each main room, inlets are located in accordance with plans, positioned at 2.2 m height. Specific behaviour of self-adjusting or humidity sensitive inlets is modelled as defined in Table 2. As the attic is generally highly ventilated, we added large cracks for the pitched roof with the following characteristics: 0.5 for the flow exponent, and 0.3 kg/(s.$Pa^{0.5}$) for the flow coefficient (CSTB, 2006).

Leakages are distributed over the 227 m² building envelope (A_{Tbat}). In each zone, two cracks have been defined on external walls at 0.63 m and 1.88 m height, and one crack on the ceiling at 2.5 m height as recommended in (EN ISO 13789, 2007). The flow exponent of cracks is equal to 0.65. The zones are interconnected by 90 cm wide doorways considered as closed during simulations. To ensure air circulation from dry to wet rooms, each doorway is considered with a 1 cm undercut for all inner doors and 2 cm undercut for the kitchen door, according to (XP P50-410, 1995). The door undercut is modelled as a large crack with 0.5 flow exponent.

Wind pressure coefficients

As wind pressure coefficients (Cp) depend on the surrounding ground and the wind angle relative to each facade, we have considered two exposure conditions: “exposed” and “surrounded by obstructions”. The local wind velocity depends on the surface roughness and the height. The wind profile used by Comis follow a power law detailed in (Anon, 1991), with a roughness coefficient of 0.17 for exposed condition. The wind pressure coefficients Cp depend on four parameters: the spatial location on the building envelope, the building shape, the surroundings and the wind direction. The Cp values are considered in accordance with (Liddament, 1996) for the shielding conditions “exposed”.

3. RESULTS AND DISCUSSION

3.1. Impact of air permeability on heating needs

Simulation is done for the four ventilation systems using the multi-zone model with the oceanic climate of Poitiers (HDD 18°C 2321°C.Day). Figure 4 illustrates the results of energy needs and figure 5 the energy needs sensitivity towards air permeability.

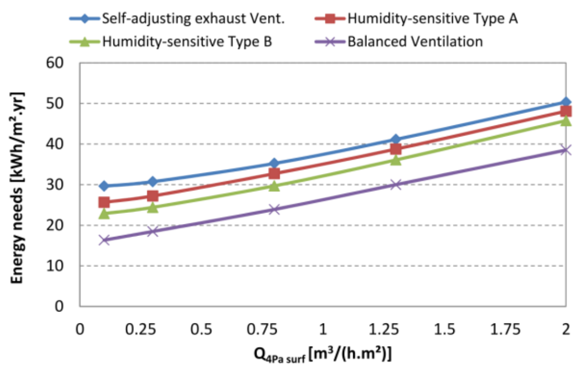


Figure 3: Influence of ventilation systems on energy needs

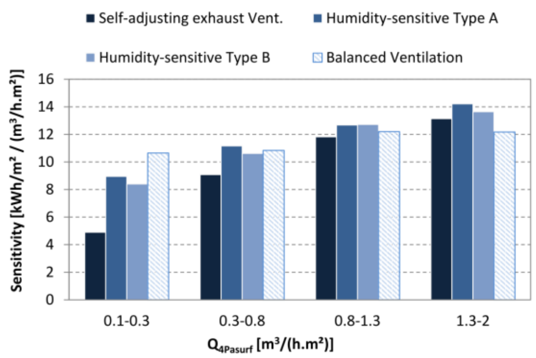


Figure 4: Sensitivity of energy needs for different ventilation systems

As expected, energy needs increase with air-permeability. The balanced ventilation enables the higher energy savings followed by humidity sensitive and self-adjusting ventilation (specific electricity for fans is not countered). A clear distinction can be based upon the level of air permeability:

- For good airtightness (air permeability below 0.3 $m^3/(h.m^2)$), the sensitivity of balanced ventilation (11 kWh/(m².Year) per unit of air permeability) is the highest followed by humidity sensitive ventilation (8 kWh/(m².Year) per unit of air permeability), and self-adjusting extract ventilation (5 kWh/(m².Year) per unit of air permeability). This can be explained by a weaker under-pressure in the case of balanced and humidity sensitive ventilation systems - that can be lower than the weather induced pressure. This exposes the building more frequently to the influence of crossing airflow, and as a result the total air change rate becomes higher.
- For higher air permeability (above 0.8 m3/(h.m²)), the energy needs increase in the same amount (12 to 14 kWh/(m².Year) per unit of air permeability) whatever the ventilation systems. This can be explained by a leakage air renewal which significantly exceeds the air renewal related to the different ventilation systems.

3.2. Impact on air flow pattern

As observed with the previous simulations, the wind effect could disturb the normal distribution of fresh air from main rooms to wet rooms, with a risk of damaging indoor air quality. In fact, inversion of airflow pattern contributes to the dispersion of air pollutants or odours from service rooms to main rooms. In order to analyse this phenomenon, the yearly occurrences of airflow disruption (flow inversion: service rooms -> main rooms) have been countered during the occupancy period. For example, an occurrence of 10% means that the flow direction is correct 90% throughout the year, regardless of the wind conditions. Figure 5 illustrates the occurrences of airflow inversion for the four ventilation systems.

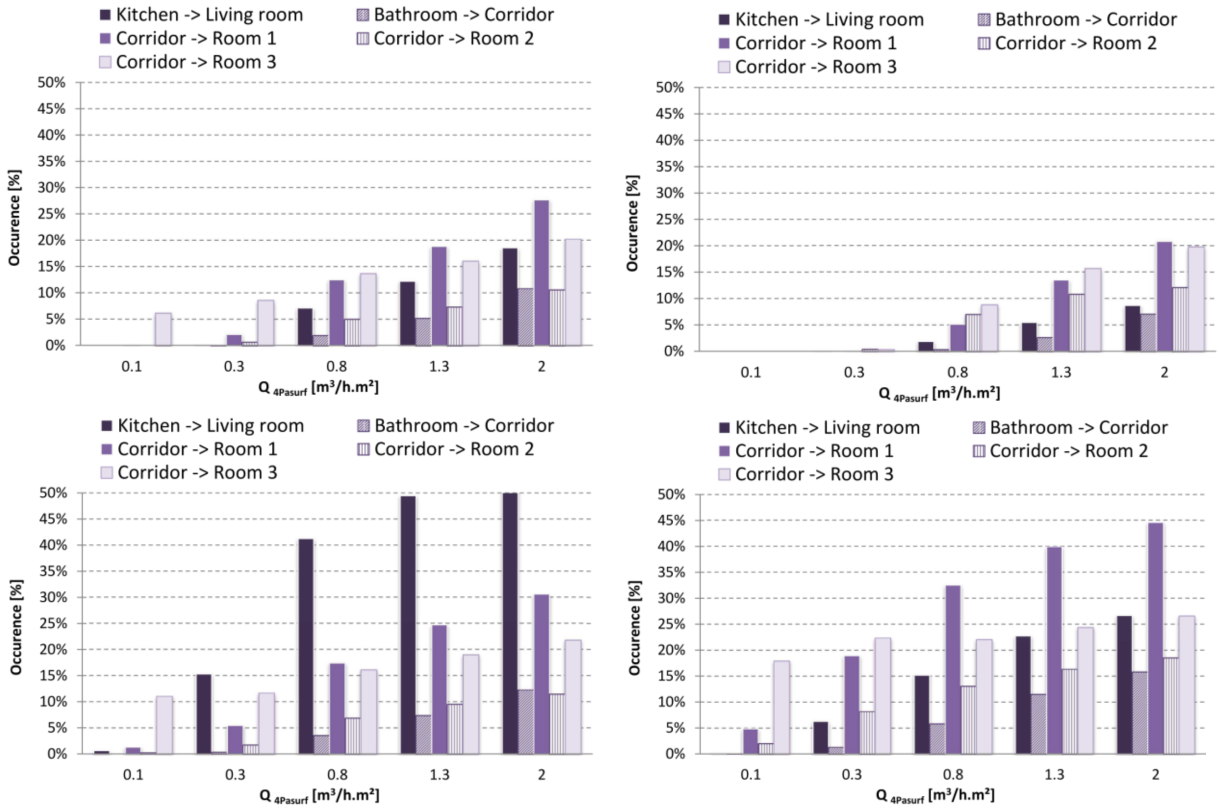


Figure 5: Occurence of Airflow inversion for self-exhaust ventilation (up-left) – balanced ventilation (up-right) - Humidity sensitive Hygro A (down-left) and Hygro B (down-right)

From figures 5, we can conclude the following:

- an increased risk of airflow inversion and pollutants dispersion from service rooms in the case of humidity sensitive ventilation systems, particularly if the airtightness is not clearly improved, with an occurrence from 10 to 25% (Hygro B), up to 45% (Hygro A);
- a significantly lower risk with balanced ventilation system, and self-adjusting extract ventilation, even if airtightness is close to 0.8 $m^3/h.m^2$;
- the factors disturbing correct airflow circulation are principally weaker under-pressure due to modulation of the fan speed extractor, and the facade self-adjusting air inlet located downwind. Thus, the most affected ventilation system is the humidity sensitive ventilation Hygro A with fixed air-cross section, and the less affected is balanced ventilation system;
- a poor airtightness (> 1.3 $m^3/h.m^2$) causes a heterogeneous air renewal of the rooms, yet all equipped with the same natural or mechanical air-inlet. The prevailing winds in Poitiers are from the southwest sector. Therefore, room No. 3 located downwind has the highest occurrences;
- only excellent airtightness would ensure an accurate and uniform air flow.

The air vented from the home by exhaust fans must be replaced by outside air. This new air comes into the home either through controlled inlets, supply units or through air leakage. Many of the air leaks come from undesirable locations such as crawl spaces, attics, or cross insulation lining and may threaten air quality by pulling gas pollutants or dust into the home.
In order to analyse the impact of leakages distribution on airflow pattern, an heterogeneous leakages distribution has been tested with 30% of total leakage areas being grouped on bathroom envelope (instead of 6% with an uniform distribution as in the previous

simulations). Figure 6 shows a comparison of the occurrences of airflow inversion between uniform and heterogeneous leakages distributions for the case of humidity sensitive hygro B ventilation system (most widely distributed) and an airtightness of 0.8 $m^3/(h.m^2)$ (reference used before the last French regulation RT2012).

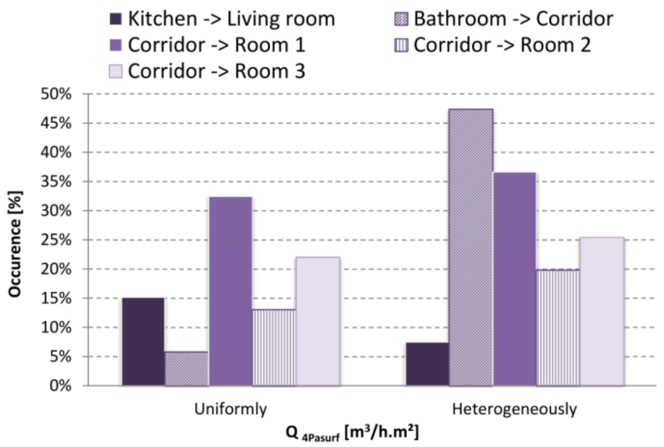


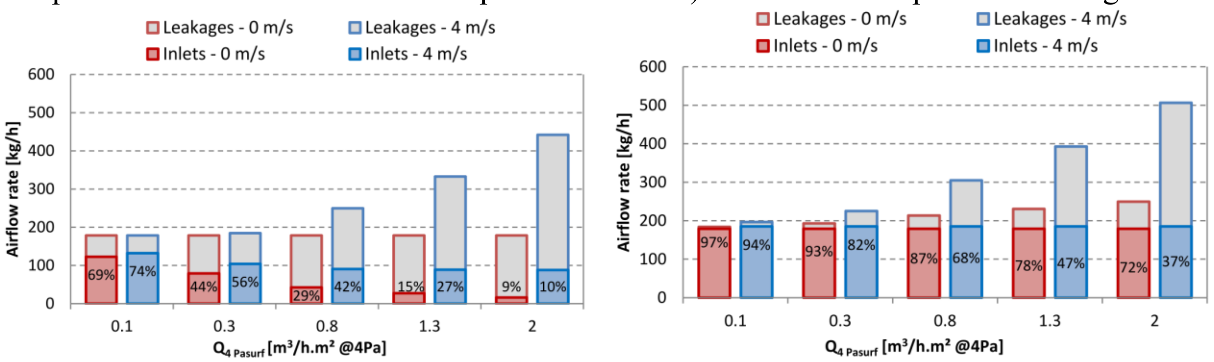
Fig. 6 Comparison of airflow inversion between uniform and heterogeneous leakages distribution

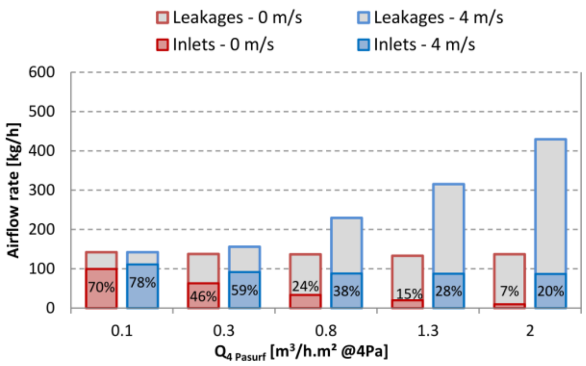
The leakages distribution method does not cause noticeable difference on heating needs between a uniform and heterogeneous distribution (respectively 31.6 kWh/m² against 31.8 kWh/m²). However, the ventilation efficiency is significantly affected. The occurrence of airflow inversion in bathroom (from bathroom to corridor) rose from 6% to 47%, pulling humidity into the main rooms. In the kitchen, the occurrence of airflow inversion is slightly decreased. Therefore, leakages located in wet rooms could significantly disturb the overall air-renewal into the dwelling.

3.3. Impact on air renewal through inlet and supply units

A last set of simulations has been performed to analyse the fresh air origin, by quantifying the amounts of airflows from leakages and inlet/supply units for each ventilation systems. Two calculations have been done with Comis under the following stationary conditions:

- Case n°1 without wind and thermal drift effects (indoor and outdoor temperatures at 20°C)
- Case n°2 with a 4 m/s wind speed oriented SSW, and a 10°C thermal drift (indoor temperature at 20°C and outdoor temperature at 10°C). The results are presented on figures 7.





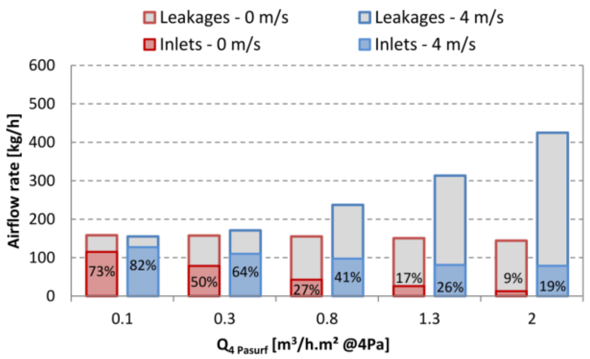


Figure. 7: Airflow rate through leakages and Inlets for self-exhaust ventilation (up-left) – balanced ventilation (up-right) - Humidity sensitive Hygro A (down-left) and Hygro B (down- right)

Under the wind effects, outside air flow and leakage rate increases with the airtightness degradation. With air-inlets, the outside air cross mainly air leakages than air-inlets. Among the exhaust ventilation systems, the proportion of outside air from leakages seems slightly higher with humidity controlled ventilation Hygro A, compared with the other two systems. For example, with a poor airtightness of 1.3 $m^3/(h.m^2)$ @ 4 Pa, the proportion of outside air from the air inlets is between 15 and 30% with exhaust ventilation systems, against 45 to 80% with balanced ventilation systems, depending on weather conditions. For an excellent airtightness less than 0.3 $m^3/h.m^2$ @ 4 Pa, the proportion of outside air from supply units is always greater than 80% with balanced ventilation, against 45 to 60% with exhaust ventilation.

As a result, with a poor airtightness, dispersion of pollutants (chemical pollutants and dust) could be probably more important with exhaust ventilation systems compared with balanced ventilation systems. Therefore, an excellent airtightness treatment seems to be recommended for all mechanical ventilation systems to ensure proper functioning of hygienic ventilation (with ensure the good functioning of the ventilation systems). A good airtightness improves the performance of Exhaust-only ventilation system by giving more authority to the air inlets (lower flows through leakage is compensated by a better control of inflows).

4. CONCLUSIONS

A multi-zonal model TRNSYS-COMIS has been developed to study the envelope air-tightness influence on the functioning of four ventilation systems. The numerical investigations illustrate the influence of air-tightness on heating demand and ventilation efficiency. For a French oceanic climate, heating demand increase in the same amount (12 to 14 kWh/(m².Year) per unit of air permeability, whatever the ventilation system choice. Demand control ventilation systems - such as humidity controlled exhaust systems - seem to be more sensitive to air permeability, especially in terms of airflow control. With a low airtightness performance (Q4pasurf > 0,3 $m^3/h.m^2$), the simulations indicate that normal airflow pattern is clearly disturb and could contribute to the dispersion of air pollutants or odours from service rooms to main rooms. At the same time, airtightness default on envelop (local crack) seems induce heterogeneous air renewal of the main rooms.

5. ACKNOWLEDGEMENTS

This work has been financed by the French Ministry of Housing. The sole responsibility for the content of this publication lies with the authors. It does not necessarily reflect the opinion of the Ministry.

6. REFERENCES

Pan, W. (2010) *Relationships between air-tightness and its influencing factors of post-2006 new-build dwellings in the UK*. Building and Environment 45 (2010) 2387-2399.

Liddament M. (1996) *A Guide to Energy-Efficient Ventilation*. AIVC. March 1996.

Richieri, F. et al. (2007) *Control of Outdoor Ventilation Airflow Rate - Evaluation of Setting Methods*. International Journal of Ventilation: 1 December 2007, Vol. 6, No. 3, pp. 207-218.

Dimitroulopoulou C. (2012) *Ventilation in European dwellings: A review*. Building and Environment 47 (2012) 109-125.

Durier F. (2008) *Trends in the French building ventilation market and drivers for changes*. Ventilation Information Paper n°19 – AIVC

Woloszyn, M. et al. (2009) *The effect of combining a relative-humidity-sensitive ventilation system with the moisture-buffering capacity of materials on indoor climate and energy efficiency of buildings*. Building and Environment 44 (2009) 515– 524.

Chan, W.R. et al. (2013) *Analysis of air leakage measurements of US houses.* Energy and Buidlings 66 (2013) 616–625.

Liddament M. (1986) *Air infiltration calculation techniques – an application guide*. AIVC, June 1986.

QUAD-BBC (2012), *final report*, convention ANR 08-Habisol007, January 2012

Limoges et al. (2008) PREBAT PABHI – *Etude de la perméabilité à l'air des bâtiments hyper isolés*, ISRN EQ-CT69-DVT/RE—08-91—FR

EN ISO 13789:2007 (2007) *Thermal performance of buildings — Transmission and ventilation heat transfer coefficients — Calculation method.*

EN NF 15242:2007 (2007) *Ventilation des bâtiments - Méthodes de calcul pour la détermination des débits d'air dans les bâtiments y compris les infiltrations*.

Richieri, F. et al. (2013) *Numerical evaluation of the airtightness impact on energy needs in mechanically ventilated dwellings*, Buildair symposium

Savin, J.-L. et al. (2011). *Demand-controlled ventilation: an outline of assessment methods and simulation tools*. AIVC conferences, 32nd, Proceedings. Presented at the 32nd AIVC conference ; 1st TightVent conference : Towards optimal airtightness performance, Brussels, Belgium: AIVC.

CSTB (2014) Avis Technique 14/13-1911 - *Systèmes de ventilation mécanique hygroréglable*, January 28th

H.E. Feustel, (1999) COMIS - *an international multizone air-flow and contaminant transport model.* Energy and Buidlings, April 1999, Volume 30, Issue 1, pp. 3-18.

Voit P, et al. (1994) *Common EC validation procedure for dynamic simulation programs – application with TRNSYS*, TRANSSOLAR GmbH. In: Conference of international simulation societies 94. Zürich, 1994.

J.M. Furbringer, et al. (1996) Annex 23 *Multizone air flow modelling – Evaluation of Comis*. Swiss Federal Institute of Technology, Lausanne, 1996

Brogat, B. et al. (2003) *Bâtiment et santé – Ventilation des bâtiments – Réhabilitation dans l'habitat collectif.* Edition CSTB - ISBN 2-86891-306-7

IEA ECBCS (2002) Annex 27 - *Simplified tool Handbook - Evaluation and Demonstration of Domestic Ventilation Systems* – ISBN 0-9542670-1-x

CSTB (2006). *Guide réglementaire* - Règle Th-Bât – Th-U 1/5 coefficient Ubât.

XP P50-410 (1995) DTU 68.1 – *Installation de ventilation mécanique contrôlée : règles de conception et de dimensionnement* (Indice de classement : P50-410).

Anon. (1991) *Code of practice for ventilation principles and designing for natural ventilation*, BS 5925:1991; British Standards Institute, London

CONCRETE IN VENTILATED FACADES FOR NATURAL COOLING OF BUILDINGS. SINHOR PROJECT

Servando Álvarez[1], César Bartolomé[2],

1 Grupo de Termotecnia, Universidad de Sevilla
salvarez@us.es

2 IECA
cbartolome@ieca.es

SUMMARY

The framework in which the SINHOR project takes place is the "Service Contract R + D + i Relating to Competence Scope of the Ministry of Public Works and Housing" with the research project entitled "Analysis of the energy performance of closures concrete based on maximizing the benefits derived from the thermal inertia".

Meeting the "20-20-20" targets for reduction of CO_2 emissions necessarily involves a drastic reduction of energy consumption in buildings.

SINHOR project is oriented to promote the use of concrete solutions in buildings based on maximizing the benefits of its thermal inertia for both heating and cooling periods.

The active solution developed has two configurations, one for summer (cooling mode) and another for winter (heating mode). In the cooling mode, the constructive solution is similar to a ventilated facade that is formed by a thermally insulated outer layer of concrete, an intermediate air layer and an inner layer of concrete. The inner layer is cooled at night by forced ventilation using an outdoor - outdoor scheme. The concrete building facades are used as heat sinks. The aim is to cool the inner layer of concrete moving outdoor air through the air layer during night taking advantage of the low nigh-time air temperatures. The cool stored is released to the interior spaces when the maximum peak load of the space takes place (see figure).

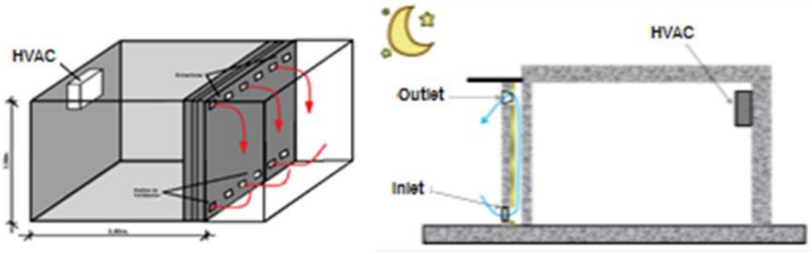


Figure: module test for the analysis of the influence of night cooling of the concrete wall through the chamber ventilated by forced ventilation.

The aim of the presentation is to show the potential of special concrete walls as solutions to reduce energy demand in residential buildings by heat storage and thermal offset in Spanish Mediterranean climates.

MATERIALS TO FIGHT URBAN CLIMATE CHANGE.

Matthaios Santamouris

National and Kapodistrian University of Athens
Physics Department
Group Building Environmental Research
Panepistimioupolis, 15784 Athens, Greece

msantam@phys.uoa.gr

EXTENDED SUMMARY

Local climate change and in particular, the urban heat island, is the more documented phenomenon of climate change. It deals with the development of higher ambient temperatures in the dense areas of cities suffering from high anthropogenic heat release, compared to the temperature of the surrounding suburban or rural areas. Actually, there are available measurements for about 400 cities around the world. Heat island exists at any latitude and may be present during the day or night period, depending on the local thermal balance. The average magnitude of the phenomenon varies between 3 to 6 K, while much higher values are reported in many cities.

The urban heat island has a very serious impact on the energy consumption of building during the summer period while it increases the peak electricity demand spend by buildings. In parallel, it has a serious impact on the environmental quality of the indoor and outdoor spaces, increases the concentration of harmful pollutants like the tropospheric ozone and has a serious impact on human health.

To counterbalance the impact of local climate change and urban heat island, specific mitigation techniques have been developed and proposed. Mitigation techniques deal with the use of advanced materials that reflect the solar radiation and also emit highly infrared radiation, the use of additional green in open spaces like parks and street green and buildings like green roofs, the reduction of anthropogenic heat, the use of cool sinks that present a lower temperature than the atmosphere, etc.

Several types of mitigation materials have been developed and are available in the market. Products may be clustered in different generations, according to their maturity and suitability to mitigate the urban heat phenomenon. The first generation involves all natural materials, the second highly reflective white materials, the third infrared reflective colour materials, the fourth nanotechnological PCM doped reflective materials and finally the fifth generation involves smart chromic materials that are able to change colour as a function of the environmental conditions.

HOW COOL ROOFS INTERACT WITH PCMS: INVESTIGATING THERMAL-ENERGY BEHAVIOR OF A COOL ROOF MEMBRANE WITH PARAFFIN BASED PCM INCLUSION

Anna Laura Pisello[*1], Luisa F. Cabeza[2], Franco Cotana[1]

1 Department of Engineering, University of Perugia
Via G. Duranti, Perugia – Italy
anna.pisello@unipg.it

2 GREA Innovació Concurrent, Universitat de Lleida, Lleida, Spain. Edifici CREA, Pere de Cabrera s/n, 25001, Lleida, Spain

1 EXTENDED SUMMARY

The effect of Phase Change Materials (PCMs) to optimize indoor thermal comfort conditions and reduce cooling energy requirement when included in envelope components and materials is demonstrated by an extensive scientific literature (Zhou et al., 2012. Orò et al., 2012). In this view, this research consists of the development and prototyping of an innovative passive cooling polyurethane based membrane with PCMs inclusion for roofing applications (Pisello et al., 2015). The nanoscale morphological characteristics of the composite membrane are assessed and compared to the classic one without PCM inclusions. Then, thermal-optical properties are determined through laboratory analyses. Additionally, calorimetry tests are performed to study the role of different concentration and phase change temperature of PCMs in terms of thermal storage/release potential. In particular, PCMs with transition temperatures at 25°C and 55°C are considered. Finally, the thermal-energy analysis of such composite membrane is performed by means of calibrated dynamic simulation of a prototype building located in central Italy (Figure 1). In particular, the effect of PCMs with varying phase change temperature is assessed in different climate conditions both in winter and summer conditions. To this aim, the roof configurations are simulated and compared with and without the integration of PCMs with different transition point i.e. 25°C, 35°C, 45°C, and 55°C, in Mediterranean climate, i.e. Rome, and in Hot desert climate, i.e. Abu Dhabi. Additionally, the inclusion of PCMs into non-cool bitumen based membrane is evaluated with the purpose to optimize the choice of the proper PCMs by taking into account the passive cooling capability of the inclusion medium.

2 DISCUSSION

The prototyping procedure showed that non-capsulated paraffin-based PCMs are able to be stored in the existing cool roofing membrane and to be distributed into the medium with no technical and operative problems. The results of heat transfer calculation showed that configurations with cool polyurethane-based membrane as roof coating always present lower external surface temperatures and heat gains through the roof, in both summer and winter and for both climate conditions, i.e. Mediterranean (Rome) and Hot desert (Abu Dhabi), due to the higher solar reflectance capability. In summer, the integration of the proper PCM in the cool membrane for Rome and Abu Dhabi, provides further cooling effect. PCMs changing phase at higher temperature (from 35°C to 45°C) better optimize the performance of the non-cool black membrane in terms of external surface temperature and heat gains reduction both in Rome and Abu Dhabi.

Figure 1: Continuously monitored buildings representing the field case study of the research.

Therefore, in order to obtain effective thermal benefits, PCMs should be selected not only with varying indoor and outdoor boundary conditions, but also with varying materials properties, in particular its passive cooling capability, e.g. solar reflectance and thermal emittance.

The dynamic thermal-energy simulation showed key energy saving produced by PCMs inclusion in summer conditions in Mediterranean climate and in winter conditions in hot desert climate. In particular, the integration of PCMs in the water-proof membrane was able to decrease the energy demand for cooling up to 22% and 12% with respect to the simple cool membrane and of 23% and 11% with respect to the non-cool membrane in the hottest months in Rome and the coolest months in Abu Dhabi, respectively. However, during winter in Rome PCMs showed to be not activated.

In this panorama, this work combined numerical and experimental campaigns to show that PCMs can be effectively included in waterproof polyurethane based roofing membranes and that the integration of passive cooling system such as cool roofs and thermal storage systems such as phase change materials is operatively possible and effective, showing promising opportunities for energy saving in buildings in hot weather conditions. The thermal-energy analysis also pointed out that the proper choice of PCM melting temperature should be carried out not only by taking into account a complex integration of boundaries such as indoor and outdoor environmental conditions, but also considering the optic-energy characteristics of the media, such as solar reflectance capability, largely affecting their thermal behavior under the sun.

3 REFERENCES

Zhou D., Zhao C.Y., Tian Y. (2012). Review on thermal energy storage with phase change materials (PCMs) in building applications. *Applied Energy*, 92, 593-605.

Oró E., de Gracia A., Castell A., Farid M.M., Cabeza L.F. (2012). Review on phase change materials (PCMs) for cold thermal energy storage applications. *Applied Energy*, 99, 513-533.

Zhang Y., Zhou G., Lin K., Zhang Q., Di H. (2007). Application of latent heat thermal energy storage in buildings: State-of-the-art and outlook. *Building and Environment*, 42, 2197-2209.

Pisello A.L., Castaldo V.L., Pignatta G., Cotana F., Santamouris M. (2015). Experimental in-lab and in-field analysis of waterproof membranes for cool roof application and urban heat island mitigation. *Energy and Buildings*, http://dx.doi.org/10.1016/j.enbuild.2015.05.026.

ARE WOMEN FEELING COLDER THAN MEN IN AIR-CONDITIONING BUILDINGS?

Bjarne W. Olesen

Intl. Centre for Indoor Environment and Energy, Technical University of Denmark
Nils Kopple's Alle building 402
DK-2800 Lyngby, Denmark
**Corresponding author: bwo@byg.dtu.dk*

1 INTRODUCTION

Recently the international media like in USA, Canada, UK, Denmark, Germany etc. has been discussing the issue of differences between men and women regarding thermal comfort and the preferred room temperature. This presentation will discuss the issue of thermal comfort and the existing knowledge on the influence of gender, age, race, etc.

2 RESULTS

2.1 The Dutch study

In their paper, researchers Boris Kingma and Wouter van Marken Lichtenbelt state, "Indoor climate regulations are based on an empirical thermal comfort model that was developed in the 1960s. Standard values for one of its primary variables—metabolic rate—are based on an average male, and may overestimate female metabolic rate by up to 35 percent". According to their experiment with 16 females performing sedentary work, they measured a significant lower metabolic rate for women than the metabolic rate for a "standard man", which they found in literature. This will according to the authors result in a higher preferred room temperature for women.

2.2 The existing literature

Fanger (1982), Fanger and Langkilde (1975), and Nevins et al. (1966) used equal numbers of male and female subjects, so comfort conditions for the two sexes can be compared. The experiments show that men and women prefer almost the same thermal environments. Women's skin temperature and evaporative loss are slightly lower than those for men, and this balances the somewhat lower metabolism of women. The reason that women often prefer higher ambient temperatures than men may be partly explained by the lighter clothing normally worn by women.

First, the primary reason is that we are overcooling buildings in summer, using enormous amounts of energy, and creating uncomfortably cold conditions for everyone. A study at Lawrence Berkeley National Laboratory found that average temperatures in office buildings in the U.S. are colder in the summer than in the winter (exactly the opposite of what they should be), and are actually lower than the minimums recommended by the standards

2.3 The existing standards

Existing international standards like ISO EN7730, EN15251 and ASHRAE 55 are based on the same basic studies described above. These standards do not specify different room temperatures for women and men when doing the same work and dressed in similar clothing.

As explained by Gail Brager, UC-Berkeley:

Contrary to what has been suggested, these standards are not devised exclusively for men. They are based on extensive laboratory studies of both men and women wearing the same clothing, engaged in the same activity, and exposed to a wide variety of thermal conditions (air temperature, surface temperature, humidity and air movement). Metabolic heat production was simply a proxy for the kind of activity. And while it is one of many variables used in an empirical formula, it is not an input to a heat balance equation, as one might find in a thermo-physiological model (which exists, but was not the basis for the standards)

The primary reason is that we are overcooling buildings in summer, using enormous amounts of energy, and creating uncomfortably cold conditions for everyone. A study at Lawrence Berkeley National Laboratory found that average temperatures in office buildings in the U.S. are colder in the summer than in the winter (exactly the opposite of what they should be), and are actually lower than the minimums recommended by the standards.

In the main studies, where they did the same sedentary work and wore the same type of clothing, there were no differences between the preferred temperature for men and women. So the researchers' finding of a lower metabolic rate for females will not influence the recommended temperatures in the existing standards. Also their study is not conclusive. They only studied 16 females at a sedentary activity. They should also have studied 16 men at the same activity to be able to compare. The reason why we, in some field studies, find that women prefer higher room temperature than men is attributed to the level of clothing. Women adapt better their clothing to summer conditions while men are still wearing suit and tie. So if the thermostat is set to satisfy the men, the women will complain about being too cold. In the standard, this adaption of clothing to summer is taken into account so if the standard is followed the women would be satisfied; but maybe not the men

3 CONCLUSIONS

The extensive studies, which form the basis for existing international standards for the thermal environment (ANSI/ASHRAE Standard 55, ISO EN 7730) included equal amount of male and female subjects and no difference in preference was observed. Despite this fact, we may often find women are colder during summer time in air conditioned offices. This can however, in most cases be attributed to the difference in clothing level between men and women. It seems the thermostat settings in summer in air-conditioned buildings are often too low and below the recommended range in existing standards.

4 REFERENCES

ASHRAE 2013. *Thermal Environmental Conditions for Human Occupancy* ASHRAE standard 55-2013

Fanger, P.O., and G. Langkilde. 1975. *Interindividual differences in ambient temperature preferred by seated persons*. ASHRAE Transactions 81(2): 140-147.

Fanger, P.O. 1982. *Thermal Comfort*. Robert E. Krieger, Malabar, FL

Kingma, B. and van Marken Lichtenbelt, Wouter.2015. *Energy consumption in buildings and female thermal demand*. Nature climate change, Letters, August 3, 2015

Nevins, R.G., F.H. Rohles, Jr., W.E. Springer, and A.M. Feyerherm. 1966. *Temperature-humidity chart for thermal comfort of seated persons. ASHRAE Transactions* 72(1):28

FIELD TRIALLING OF A NEW AIRTIGHTNESS TESTER IN A RANGE OF UK HOMES

Edward Cooper*[1], Xiaofeng Zheng[1], Christopher Wood[1], Mark Gillot[1], David Tetlow[1], Saffa Riffat[1] and Lia De Simon[1]

[1] *Architecture, Climate and Environment Research Group, Faculty of Engineering, University of Nottingham, University Park, Nottingham NG7 2RD, UK*
**Corresponding author: edward.cooper@nottingham.ac.uk*

ABSTRACT

A new low pressure 'quasi-steady' pulse technique for determining the airtightness of buildings has been developed further and compared with the standard blower-door technique for field-testing a range of typical UK homes. The reported low pressure air pulse unit (APU) has gone through several development stages related to optimizing the algorithm, pressure reference and system construction. The technique, which is compact, portable and easy to use, has been tested alongside the standard blower-door technique to measure the airtightness of a range of typical UK home types. Representative of the UK housing stock, the homes mostly have low levels of airtightness, resulting in poor energy performance, poor indoor air quality and poor thermal comfort. Some of these homes have been targeted for retrofitting and a quick, low cost and simple method for accurately determining their airtightness has clear advantages for correctly predicting the benefits of any improvements. A comparison between the results given by the two techniques is presented and the field trials indicate that the latest version of the pulse technique is reliable for determining building leakage at low pressure. Repeatability of multiple APU tests in the same house is found to be within +/-5% of the mean. A test where the leakage is increased by a known amount shows the APU is able to measure the change more accurately than the blower-door test. The APU also gives convenience in practical applications, due to being more compact and portable, plus it doesn't need to penetrate the building envelope. The field trials demonstrate the pulse test has the potential to be a feasible alternative to the standard blower-door test.

KEYWORDS
Airtightness, Building leakage, Blower door, Pulse test, Steady pressurisation;

1. INTRODUCTION

The impact of infiltration as a consequence of poor airtightness can be considerable; research by Jones (Jones, 2015) predicts that unintended infiltration across the UK housing stock may be responsible for as much as 5% of total UK energy demand. However, the standard blower-door method used to measure airtightness is something of a compromise and concerns about this technique have led to numerous attempts to find alternative, more preferable, ways of determining building airtightness. A partial selection of these attempts include AC techniques by Sharples (Sharples, 1996), Siren (Siren, 1997), Sherman (Sherman, 1986), Watanabe (Watanabe, 1999), Nishioka (Nishioka, 2003) and Modera (Modera, 1989), gradual decay techniques by Granne (Granne, 2001) and Mattsson (Mattsson, 2007), acoustic techniques by Varshney (Varshney, 2013) and Card (Card, 1978), and pulse techniques by Carey (Carey, 2001), and Cooper (Cooper, 2004, 2007, 2007). However, to date, none of these attempts have led successfully to widespread use by the airtightness testing industry.

The focus of this paper is the further development and field trialling of a low pressure pulse technique, described in a previous paper by Cooper (Cooper, 2014), referred to herein as the Air Pulse Unit (APU). Its historical development comprises of three versions, namely a gravity driven piston, a compressed air driven piston and most recently a nozzle unit. The latest nozzle test unit has recently been through several developmental stages related to optimizing the algorithm, pressure reference and system construction. It has been simplified from a bulky and heavy unit into a more compact, portable and quick-to-use version.

For validation purposes, the APU has been used to measure the airtightness of a range of UK houses, as shown in Figure 1. They are listed in the format of House Number-House type. The key parameters of the test houses are listed in Table 1.

Figure 1 Test houses (D: detached; SD: semi-detached; ET: end-terraced; MT: Mid-terrace)

Table 1: Key parameters of the test houses

House Number	Volume (m^3)	Age (years)	Position	Construction type
1	157	>100	End-terrace	Solid wall
2	196	>100	Mid-terrace	Solid wall
3	196	10-100	Semi-detached	Cavity wall
4	213	10-100	Semi-detached	Cavity wall
5	203	10-100	Semi-detached	Cavity wall
6	230	10-100	Detached	Cavity wall
7	447	10-100	Detached	Cavity wall
8	343	<10	Detached	Modern SIP
9	157	10-100	Semi-detached	Solid wall
10	371	>100	Semi-detached	Solid wall

The homes are representative of those commonly found in the UK housing stock; most have low levels of airtightness, resulting in poor energy performance, poor indoor air quality and poor thermal comfort. Five of the test homes (No 1-5) have been identified for retrofitting as part of the EU FP7 Holistic Energy Retrofit of Buildings (HERB) project and will be tested both pre and post-retrofit, with only the pre-retrofit tests reported here. A quick, low cost and simple method for accurately determining their airtightness has clear advantages for correctly predicting the benefits of any improvements.

2. METHODOLOGY

For purposes of comparison, all the houses were tested with both the APU device and the standard blower-door technique under the same conditions, with additional thermography carried in houses 1 to 5 for identifying the nature of leakage pathways.

Prior to the testing of both techniques, all the houses were prepared according to the UK's Air Tightness Testing and Measurement Association's Technical Standard L1 (ATTMA TSL1) for measuring air permeability of building envelopes in dwellings (ATTMA, 2010).

The blower-door tests followed the guidelines set out in the above mentioned ATTMA TSL1 and the BS EN:13829 (BSI, 2001). As such, the results should be comparable with those carried out for demonstrating compliance with the UK Building Regulations. The tests were conducted with the fan mounted in a suitable doorway, as shown in Figure 2, and under both pressurisation and depressurisation.

The latest APU device used for the tests is shown in Figure 3. It works by releasing a known volume of air from a pressurised tank for a short period of time. This creates a pulse in the internal air pressure and generates an airflow rate through adventitious openings. A period of quasi-steady flow is established that gives a leakage characteristic at low pressure, which with minor adjustments can be used to determine the airtightness in the same way as the high pressure technique. Further details of the equipment, test procedure and proof of the pulse concept used for the APU can be found in previous papers (Cooper, 2007, 2014).

Figure 2: The blower-door

Figure 3: The latest Air Pulse Unit (APU)

3. RESULTS

3.1. Repeatability

For the APU device to be a valid alternative to the standard blower-door test it needs to demonstrate a good degree of repeatability within a given test space. Table 2 shows the results of 18 such tests conducted in house No. 8. The relationship between pressure difference and leakage rate is represented in the table by a standardised leakage rate at 4 Pa, or V_4. The value is obtained by a curve fit to data obtained directly at the low pressures typically experienced under natural conditions. The repeatability is good, with a most of the tests falling comfortably within +/-5% of the mean V_4. If required, the repeatability could be improved further, simply by using a larger tank.

Table 2: V_4 (m^3/s) of 18 repeated test runs in house No. 8

Test ID	1	2	3	4	5	6	7	8	9	AVE
V_4	0.1166	0.1189	0.1219	0.1199	0.1182	0.1182	0.1241	0.1241	0.1148	
RPD	-2.94%	-1.01%	1.47%	-0.16%	-1.55%	-1.60%	3.37%	3.34%	-4.39%	
Test ID	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	0.1201
V_4	0.1232	0.1207	0.1231	0.1194	0.1160	0.1157	0.1252	0.1194	0.1227	
RPD	2.59%	0.48%	2.47%	-0.62%	-3.44%	-3.68%	4.21%	-0.62%	2.18%	

NOTE: AVE and RPD stand for 'mean average' and 'relative percentage difference from mean' respectively.

The graph in Figure 4 shows the internal pressure pulses generated for each of the 18 repeated test runs in house No. 8, after adjustment to still-air conditions. The method used for the adjustment, by accounting for changes in background pressure, is described in a previous paper (Cooper, 2007). The labels identify the quasi-steady period used for analysis and the closing point of the pressurised tank valve. Interestingly, it can be seen there is considerable variation in the valve closing time for these tests, however, this part of the pulse is not used for analysis and, importantly, has no impact on the quasi-steady period, which shows good repeatability. On investigation, the variation in these tests was identified as a faulty power supply, which was replaced and subsequent tests show good repeatability for the closing point.

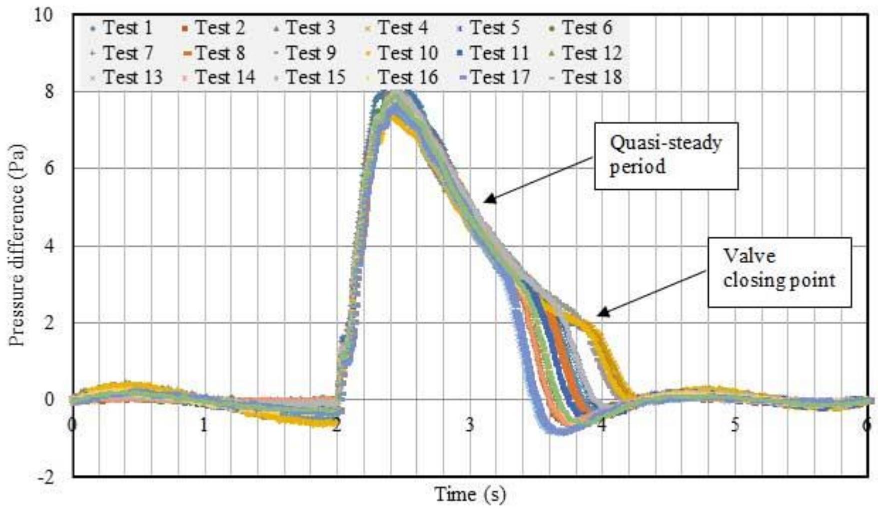


Figure 4: Adjusted internal pressure pulses from 18 repeated test runs in house No.8

3.2. Comparison between the blower-door and APU for measuring permeability

The blower door measures the leakage characteristic in a high pressure range (e.g. 10-60 Pa) and the leakage is typically presented at 50 Pa, whilst the APU measures it in a low pressure range (e.g. 1-20 Pa) and presents the leakage rate at 4 Pa. In order to make a direct comparison at the same pressure level, either at high or low level, either one or the other has to be extrapolated and uncertainties are introduced to the results as a consequence. In this report, considering that 4 Pa is typical of the pressure difference experienced by buildings under natural conditions, the test results are compared as an air permeability at 4 Pa, known as Q_4.

The power law equation is widely used for the blower door test to mathematically represent the relationship between pressure difference and leakage rate.

$$V = C\Delta P^n \tag{1}$$

Where, V is the leakage rate (m^3/s), C is an air leakage coefficient, ΔP is the pressure difference (Pa) and n is a pressure exponent. In order to predict the air leakage rate at 4 Pa from the blower-door test, Eq.(1) can be used to extrapolate down from higher pressures. In practice, the two techniques should not be expected to agree perfectly, due to the uncertainties in extrapolation, but they should be expected follow the same trends from house to house.

In Figure 5, the permeability at 4 Pa predicted by the blower-door, $Q_4(BD)$, is compared with the permeability measured directly at 4 Pa by the APU, $Q_4(Pulse)$. The comparison shows they follow a similar pattern and interestingly the APU gives lower Q_4 values than the blower-door in 9 out of 10 houses. The exception is house No.2, where the APU gives a higher Q_4 than the blower-door. However, during the blower-door tests in this house, it was noticed that the upper part of a loosely installed plasterboard panel, shown in Figure 6, opened when the blower-door depressurised the building, but not when it was pressurised (the mode also used by the APU). The thermographic image on the right side of Figure 6 shows the gap during depressurisation; the cool air being drawn into the building through the gap can be seen clearly by the plume surrounding the opening. The higher the pressure difference, the bigger the opening becomes and consequently the higher the leakage rate. Perhaps counterintuitively, this actually leads to a lower $Q_4(BD)$ for the depressurisation than pressurisation, due to the lower gradient of the relationship between leakage and pressure, as illustrated in Figure 7. In this graph, the annotated line represents the power law curve fit between the building leakage rate and pressure difference across the envelope if the position of the plasterboard were not affected by the induced depressurisation. It can be seen to make a significant difference at low pressure. In practice, the effect is reduced by using an average of the pressurisation and depressurisation results, but the impact would still be enough to explain the difference in the trend between house No.2 and the other houses.

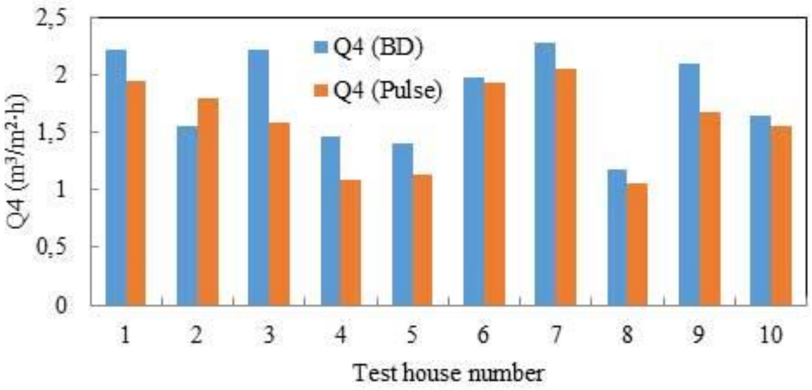


Figure 5: The permeability @4Pa, Q_4, predicted by the blower-door (BD) and measured by the APU (Pulse).

Figure 6: Photograph and thermographic image of a loosely installed plasterboard panel in house No. 2.

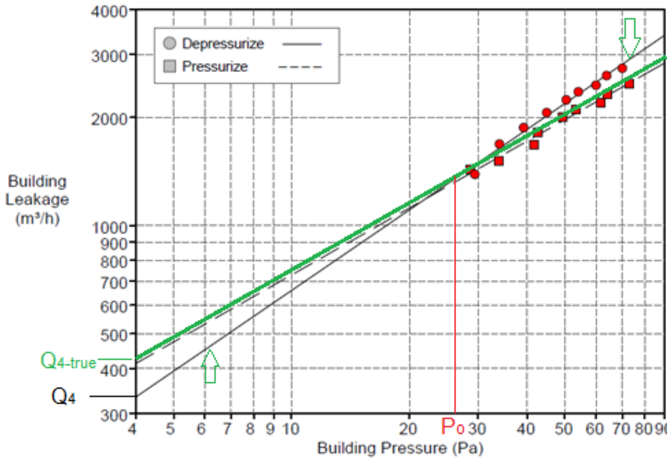


Figure 7: Logarithmic plot of blower-door test result for house No.2

3.3. Known opening test for accuracy

A full explanation for why the APU consistently gives a lower permeability than the blower-door is beyond the scope of this paper, however a simple check to see which technique is more accurate at measuring an added known opening can be made.

A short sharp-edged circular orifice with a diameter of 100mm was added into a window in house No.8, as shown in Figure 8. Assuming an appropriate discharge coefficient of 0.61 therefore gives an effective leakage area of 4.7909×10^{-3} m^2. Tests were conducted for both techniques with and without the added opening. The increase in leakage rate measured for both techniques was then converted to an effective leakage area and compared to the known opening, as shown in Table 3. It can be seen that the measurement made by the APU is much closer to the known effective area than the blower door measurement in this case. Other similar tests have been conducted with the same conclusion being drawn each time.

Figure 8: Setup of the known opening in house No. 8

Table 3: Results of known opening tests using the blower door and APU

Method	Measured area of the opening, m^2	Percentage difference from the actual known opening of 4.791×10^{-3} m^2
Blower door (@ 50 Pa)	5.9264×10^{-3}	23.7%
Blower door (@ 4 Pa)	6.0913×10^{-3}	27.1%
APU (@ 4 Pa)	5.0349×10^{-3}	5.1%

4. CONCLUSIONS

The low pressure air pulse unit (APU) has been through several development stages related to optimizing the algorithm, pressure reference and system construction. The technique was tested alongside the standard blower-door technique to measure the airtightness of a range of typical UK home types. A comparison between the results given by the two techniques was conducted and the field trials indicated that the latest version of the pulse technique is reliable for determining building leakage at low pressure. Repeatability of multiple APU tests in the same house was found to be within +/-5% of the mean. A test where the leakage was increased by a known amount showed the APU was able to measure the change more accurately than the blower-door test. The APU also gives convenience in practical applications, due to being more compact and portable, plus it doesn’t need to penetrate the building envelope. The field trials demonstrated the pulse test has the potential to be a feasible alternative to the standard blower-door test.

6.1 ACKNOWLEDGEMENTS

The work reported in the paper relates to the InnovateUK ‘Scaling up retrofit - Development of a novel low pressure technique for measuring the air tightness of buildings’, EU FP7 ‘HERB’ and EU Horizon2020 ‘Built2Spec’ funded projects; the authors wish to thank all involved parties.

6.2 REFERENCES

ATTMA (2010). *Technical Standard L1. Measuring Air Permeability of Building Envelopes (Dwellings)*. Air Tightness Testing and Measurement Association. October 2010 Issue.

BSI, (2001). *Thermal performance of buildings - determination of air permeability of buildings - fan pressurization method*, British Standards Institution, London, UK, 2001.

Card W.H., Sallman A., Graham R.W., and Drucker E.E. (1978). Air leakage measurement of buildings by an infrasonic method, *Department of Electrical Engineering, Syracuse University, Tech. Rep. TR-78-1*.

Carey PS, Etheridge DW (2001). Leakage measurements using unsteady techniques with particular reference to large buildings. *Building Serv. Eng. Res. Technol. 2001; 22:* 69-82.

Cooper EW, Etheridge DW (2004). Measurement of building leakage by unsteady pressurisation. *25th AIVC Conference*, Prague, 2004.

Cooper EW, Etheridge DW (2007). Determining the adventitious leakage of buildings at low pressure. Part 1: uncertainties. *Building Serv. Eng. Res. Technol. 2007; 28*: 71-80.

Cooper EW, Etheridge DW (2007). Determining the adventitious leakage of buildings at low pressure. Part 2: pulse technique. *Building Serv. Eng. Res. Technol. 2007; 28*: 81-96.

Cooper E, Zheng X.F., Gillot M., Riffat S., Zu Y.Q.(2014). A nozzle pulse pressurisation technique for measurement of building leakage at low pressure. *35th AIVC conference, Poznan, September 2014*.

Granne, F. (2001). Air and Water Tightness in Building Envelopes – Evaluation of Methods for Quality Assurance, *Report R-01/187-SE, Stockholm, KTH*.

Jones B. et al (2015). Assessing uncertainty in housing stock infiltration rates and associated heat loss: English and UK case studies. *Building and Environment 92 (2015)* pp. 644-656.

Mattsson, B. (2007). A transient pressurisation method for measurements of airtightness. *Journal of Building Physics, Vol. 31, No.1*.

Modera M.P. and Sherman M.H. (1983). A detailed examination of the LBL infiltration model using the Mobile Infiltration Test Unit, *ASHRAE Trans. 89 (2B)*, 157-179 (1983).

Nishioka T., Chen Q., Arai N., Fujiwara K., Umemiya N. and Okura R. (2003). Unsteady pressurization method to measure the airtightness of the building envelope. *Research in building physics international conference; 2nd, Building physics; Research in building physics; September 14-18, 2003.* pp 771-776.

Sharples S. and Thompson D. (1996). Experimental Study of Crack Flow with Varying Pressure Differentials. In Proceedings: *Optimum Ventilation and Air Flow Control in Buildings, 17th Air Infiltration and Ventilation Centre Conference, Gothenburg, Sweden, September 17-20, 1996*, pp. 243-253.

Sherman M.(1987). Estimation of infiltration from leakage and climate indicators. *Energy and Buildings*, *10 (1987)* pp.81-86.

Sherman M.H. and Modera M.P. (1986). Low Frequency Measurement of the Leakage in Enclosures, *Review of Scientific Instruments*, *Vol. 57 (7), 1986*, pp. 1427-1430.

Siren K. (1997). A Modification of the Power Law Equation to Account for Large-Scale Wind Turbulence. In Proceedings: *Ventilation and Cooling*, *18th Air Infiltration and Ventilation Centre Conference*, *Athens*, *Greece*, *September 23-26*, *1997*, pp. 557-561.

Varshney K., Rosa J.E., Shapiro I. (2013). Scott D . Air-infiltration Measurements in Buildings using Sound Transmission Loss through Small Apertures. *International Journal of Green Energy, 10:5*, pp.482-493.

Watanabe Y., Kobayashi H., Utsumi Y (1999). Development of validation of AC-pressurization measuring of leakage area of houses. In proceedings: *Building Simulation '99, Sixth International IBPSA Conference, Kyoto, Japan. September 13-15, 1999*. pp. 807-814.

CALIBRATING MEASUREMENT GAUGES – EXPENSE AND FINDINGS

Dipl.-Ing. Paul Simons[1], Dipl.-Ing. Stefanie Rolfsmeier[2], Dipl.-Phys. Francisco Pujiula[3]

1, 2, 3 BlowerDoor GmbH
Zum Energie- und Umweltzentrum 1
31832 Springe, Germany
info@blowerdoor.de

KEYWORDS

Minneapolis BlowerDoor; calibration; gauge; accuracy; traceability to national standards

ABSTRACT

This presentation will explain what exactly calibration should mean. It will also look into the hierarchy of and the differences between the calibration labs of national standards, accredited labs (like DAkkS, a national government-appointed accreditation body in Germany, or Cofrac – Comité français d'accréditation – in France), and manufacturer's calibration labs. Even the labs themselves must have their own measuring devices checked in order to guarantee what is known as traceability to national standards, and to determine the measurement uncertainty of their testing devices. This is necessary if you want to make a clear statement on adherence to the accuracy, which is known as a statement of compliance or declaration of conformity. Another topic covers the measuring range and the accuracy of pressure gauges and measuring fans, as well as their calibration process, by looking at the example of the Minneapolis BlowerDoor.

INTRODUCTION

In order to achieve the most precise, reproducible and therefore comparable measurement results possible when measuring the airtightness of buildings, gauges are required that can measure flow rates and building pressure differentials with a defined level of precision. The user can check the gauges' accuracy in calibration labs. But what exactly does calibration mean, how precise are the gauges, what level of quality can be expected from a calibration certificate, and are there differences between the calibration labs? These questions will be addressed below and illustrated using a Minneapolis BlowerDoor.

ACCURACY

Every gauge is assigned an accuracy level by the manufacturer. With this information, the user can estimate the extent to which the measured value shown (calculated) may deviate from the real value to be measured. If, for example, a thermometer with an accuracy level of ±2°C shows a reading of 21°C, the real temperature can fall within 19°C to 23°C. The more accurate it is, the closer the reading will reflect the actual value.

CALIBRATION

Calibration simply refers to comparing the readings of a device to be calibrated (DG-700) with those of a reference standard. The calibration laboratory records the readings of both the device and the reference standard (see Figure 2) and calculates the deviations (absolute and/or in percent). The customer can then compare these with the specifications of their gauge in order to judge its effectiveness. The calibration shows the accuracy of the calibrated device at the time of calibration; it is a snapshot in time and does not provide information on the device's long-term stability.

In the following example (Figure 1), the reference standard shows a measurement of 12 Pa while the device being tested shows 13 Pa. The deviation between device and reference is +1 Pa. The device's accuracy (for example, as provided by the manufacturer) is given as ±4 Pa. The deviation is therefore within the specification of the range between -4 Pa and +4 Pa. The device fulfills its requirements in this aspect.

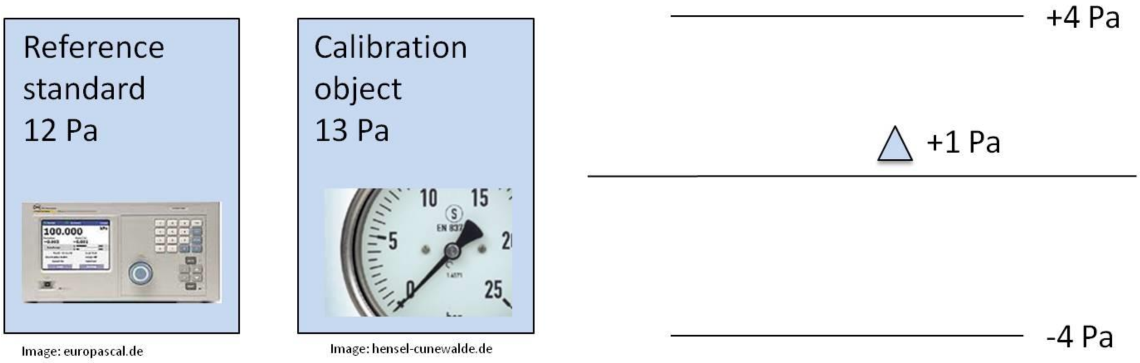


Figure 1: Measurements displayed by reference standard and by calibrated device, showing a deviation that lies within specifications.

Standard		Channel A		% Difference
9,9	Pa	9,9	Pa	0,0%
25,7		25,7		0,0%
40,5		40,5		0,0%
50,0		50,0		0,0%
64,2		64,2		0,0%
91,6		91,6		0,0%

Figure 2: Excerpt from calibration log with measurements from reference standard (here referred to as "Standard") and from calibrated device (here referred to as "DG-700 Channel A") and the deviation in percent.

In Figure 3, the calibrated device shows a measurement of 6 Pa, significantly lower than the reference standard of 12 Pa. The deviation amounts to -6 Pa and is therefore outside the specification of ±4. The specs are not met.

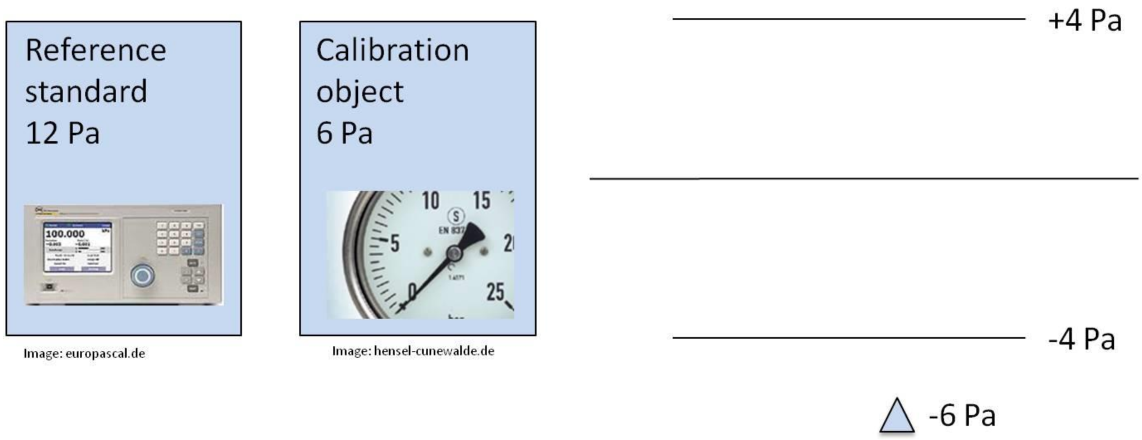


Figure 3: Measurements displayed by reference standard and by calibrated device, showing a deviation that lies outside of specifications.

The only way to achieve information on how the device performs over time is to perform repeated calibrations at certain intervals: the calibration intervals.

CALIBRATION INTERVALS

The user is responsible for regular calibration. Recommendations for calibration intervals are provided, for example, by manufacturers or can be found in standards or regulations. In the field of airtightness measurements, calibration intervals for gauges are provided in the standard DIN EN 13829 and the corresponding supplementary sheet from FLiB e. V. (Professional Association of Airtightness in Architecture) for Germany or the governmental ordinance GA P50-784_V2014 for France.

EVALUATING REFERENCES AND MEASUREMENT RESULTS

The calibration laboratories themselves also have their own reference standards regularly calibrated in order to identify any measurement uncertainty. This is necessary to be able to determine the accuracy of the calibration. This is also necessary for laboratories to be able to provide a declaration of conformity on compliance with accuracy as stipulated by, for example, regulations such as DIN EN 13829.

Inaccuracies in the reference standard influence measurement uncertainty, as do additional factors that can affect measurement results, such as the accuracy of temperature measurements and the long-term stability of the reference standard. When several years of observation demonstrate that the reference is reliably accurate, measurement uncertainty decreases.

The following figure/example shows a +1 deviation between the calibrated device and the measurement standard. The measurement uncertainty of this measurement is displayed as an error bar in the diagram. In DAkkS calibration logs, this area is provided – as agreed upon – in such a manner that the measurement value lies within the bar with a probability of 95%.

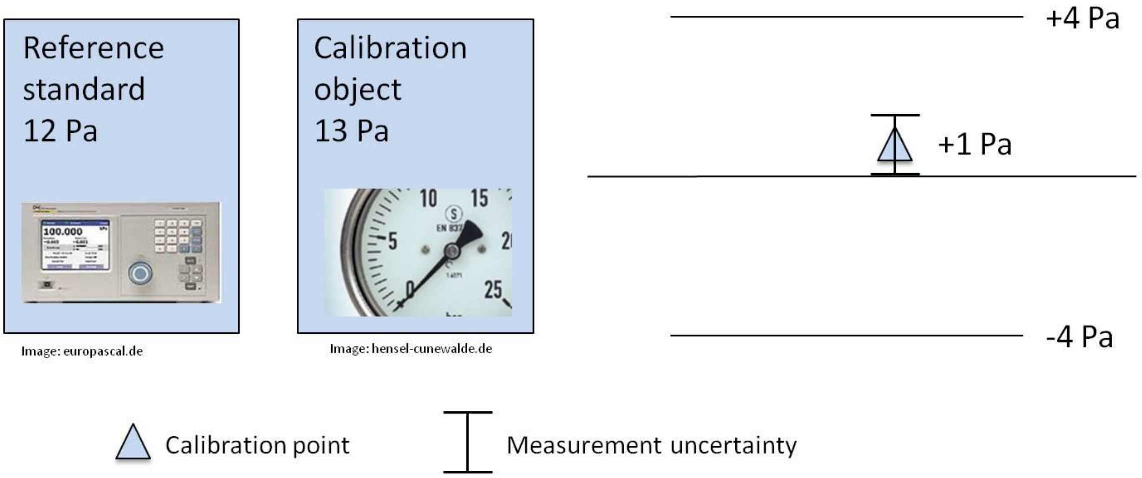


Figure 4

Measurement uncertainty can vary widely according to the reference standard used by the calibration laboratory. If it is clear that the reading provided by the reference standard lies within the determined tolerance zone, then both the difference and the sum of the reading and the measurement uncertainty must also lie within the specification limits applied. Specifications can be determined on the customer end, either taken directly from the manufacturer or from a standard, from national regulations, etc.

When all calibration points, including their measurement uncertainty levels, lie within specifications, conformity can be declared with, for example, corresponding norms or national regulations. Should these measurements, including their measurement uncertainty levels, lie outside error margins, then conformity cannot be declared.

In the following example (Figure 5), the calibrated device’s same deviation of +1 Pa can be seen. For the reference standard (left) with the low measurement uncertainty, conformity can be declared within the error margins. In the second case (right), the inaccuracy of the reference standard is so great that the error bar ends outside the error margins. Here, conformity cannot be confirmed.

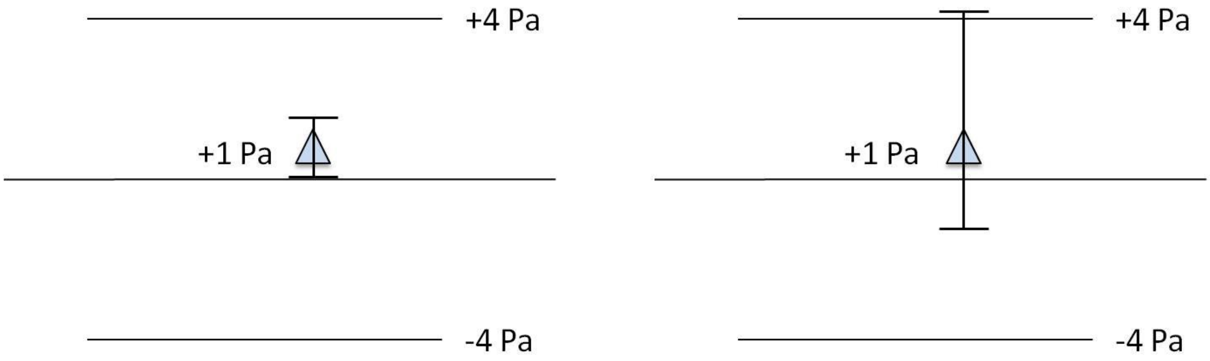
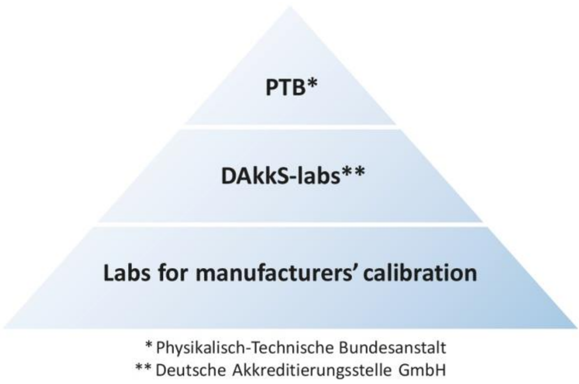


Figure 5: Relationship between measurements (deviation), measurement uncertainty and specification.

TRACEABILITY TO NATIONAL STANDARDS

A basic requirement for ensuring comparability in BlowerDoor measurements is that the gauges are calibrated on a uniform basis. This uniform basis is provided by the respective national standards. In Germany, this is the National Metrology Institute of Germany, or Physikalisch-Technische Bundesanstalt (PTB); in France, it is the National Laboratory of Metrology and Testing, or Laboratoire national de métrologie et d'essais (LNE); and in the United States, it is the National Institute of Standards and Technology (NIST). The PTB is at the tip of the calibration chain in Germany, followed by accredited laboratories and finally labs for in-house calibrations.



National standard

Accredited according to DIN 17025, specification published

Laboratories

Figure 6: The hierarchy of calibration laboratories

Error analysis for the reference standard contains the calibration chain up to the national standard. The shorter the chain between the reference standard and the national standard, the more accurate the reference is.

In terms of maintaining their reference standards, calibration processes and internal organization, DAkkS labs are accredited according to DIN EN 17025. These laboratories are inspected regularly through an internal audit and by Germany's national accreditation body, Deutsche Akkreditierungsstelle GmbH (DAkkS). DAkkS determines the smallest measurement uncertainty that the laboratory is to claim. This enables customers to compare laboratories with the same measurement areas in terms of their performance. DAkkS calibrations can be most quickly and easily identified by the DAkkS logo on the calibration log.

Permanent laboratory

Measurement/ calibration object	Measurement range	Measurement conditions/process	Smallest measurement uncertainty possible [1)]	Notes
Pressure Positive and negative overpressure p_e	-2500 Pa to 2500 Pa	DKD-R 6-1: 2014 EURAMET cg-17 Version 2.0	$1{,}0 \cdot 10^{-4} \cdot p_e + 0{,}37$ Pa	Pressurizing medium: gaz

Figure 7: Specifications for the DAkkS-accredited laboratory of BlowerDoor GmbH

In-house calibration laboratories are not accredited and can work according to their own rules. The quality of the work performed by in-house calibration labs can certainly be very high.

CALIBRATION OF AIRTIGHTNESS GAUGES

For the field of airtightness measurement, the European Standard EN 13829 – which determines the air tightness of buildings – also requires regular calibration of gauges in addition to setting down requirements for their accuracy.

ACCURACY REQUIREMENTS AND CALIBRATION INTERVALS

DIN EN 13829-2000 requires the following accuracy levels for gauges:

- Pressure gauges for measuring building pressure differential of 0 to 60 Pa: ±2 Pa
- Flow rate meter (device for measuring air flow volume): ±7 % of measurement value
- Thermometer: ±1 K

Calibration should be performed regularly in accordance with manufacturers' recommendations or a standardized quality control system.

REQUIREMENTS IN GERMANY

In Germany, the Fachverband Luftdichtheit im Bauwesen, or FLiB e.V., is a professional body for airtightness in buildings that has provided additional information on calibration in a supplementary sheet to DIN EN 13829 from 2008 (a new edition is planned for 2015). According to this, inspection/calibration should take place every two years for pressure gauges and every four years for flow rate meters. Details on the form a meaningful inspection should take are not provided. A function check (to determine if, for example, a pressure gauge is defective) can be performed through comparison with the readings of a similar device. A rough estimation of flow rate can, for example, be conducted using an aperture plate. These measures do not replace a calibration, however.

REQUIREMENTS IN FRANCE

In addition to EN 13829, France has the regulation GAP50-784. This prescribes the following calibration rules:

- Pressure gauges for measuring building pressure differential of 0 to ±100 Pa: ±1 Pa or 1% of reading
- Gauge for determining flow rate from a pressure differential: ±1 Pa or 1% of reading
- Bellows gauge with three measuring points per configuration (plate): ±2 m³ or 5% of reading

The calibration interval for the pressure gauges is one year, and two years for the bellows gauge.

CALIBRATION BASICS FOR MINNEAPOLIS BLOWERDOOR INSTRUMENTS: PROCESS AND PHILOSOPHY

With the Minneapolis BlowerDoor measuring system, the bellows gauge and the pressure differential gauge are calibrated separately. The error level can be calculated using the accuracy of the pressure differential gauge and the bellows gauge along with the law of error propagation. The measuring system's philosophy is that the bellows gauge, the pressure differential gauge and the software can all be exchanged and combined as desired. This makes the system very flexible to use and reduces transfer errors. Components can be individually switched out as needed.

Calibration of the bellows gauge is conducted using the standard calibration parameters for every configuration (orifice plate). A change in the calibration parameters is not intended by the manufacturer, even if it would be possible for relatively extensive calibration work. If calibration shows that the bellows gauge's standard values are outside the accuracy levels stated by the manufacturer, then it is usually sufficient to undertake repairs followed by another calibration. The calibration of the pressure differential gauge takes place over the device's entire positive and negative measuring range.

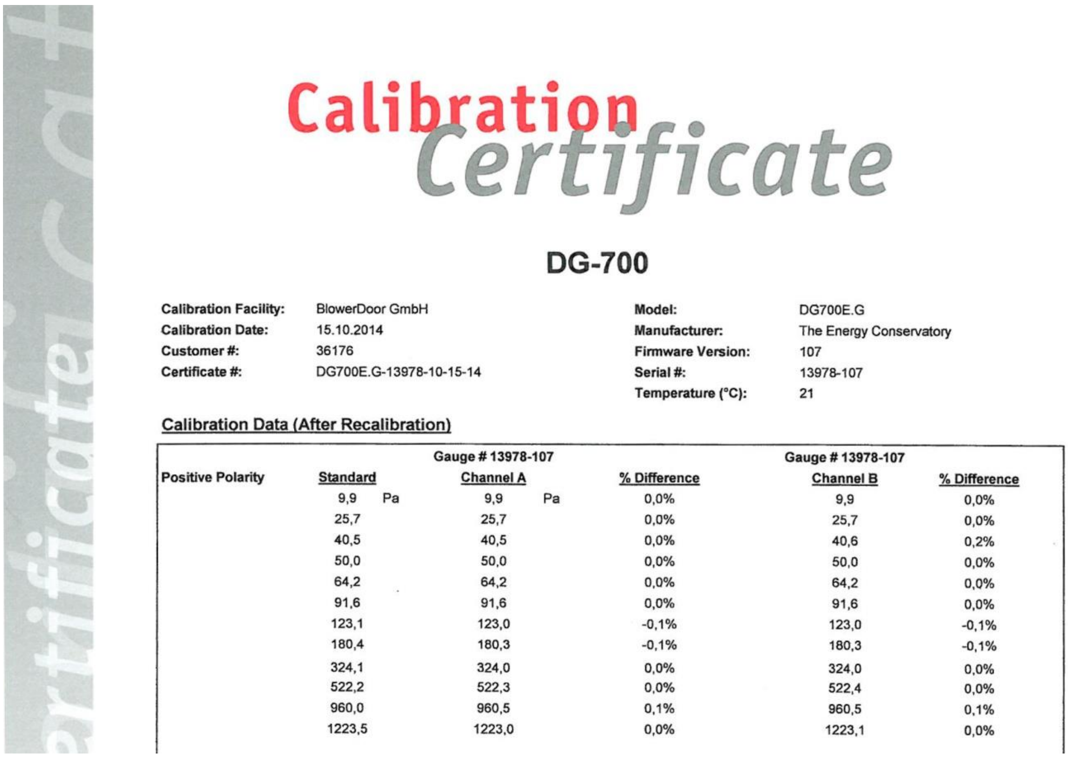
ACCURACY OF THE MINNEAPOLIS BLOWERDOOR

Afterwards, the measuring range and accuracy of pressure gauges and bellows gauges are presented, along with their calibration processes.

IN-HOUSE CALIBRATION OF PRESSURE GAUGES AT BLOWERDOOR GMBH

The pressure differential gauge DG-700 with two differential pressure channels are calibrated using a reference standard that reflects the national standard. In addition to its own calibration, the device undergoes the following standard process:

1. Update of firmware, if necessary.
2. Function check of device to be calibrated before calibration (inspection of buttons, activation of bellows gauge, etc.).
3. Initial calibration of device with documentation of reference standard and device measurements as well as current deviations. 12 measuring points are recorded each for both the positive and the negative ranges.
4. Adjustment of device being calibrated in order to achieve maximum accuracy.
5. End calibration after adjustment with documentation of reference standard and device measurements as well as new deviations. 12 measuring points are recorded each for both the positive and the negative ranges.


Calibration Certificate

DG-700

Calibration Facility:	BlowerDoor GmbH	**Model:**	DG700E.G
Calibration Date:	15.10.2014	**Manufacturer:**	The Energy Conservatory
Customer #:	36176	**Firmware Version:**	107
Certificate #:	DG700E.G-13978-10-15-14	**Serial #:**	13978-107
		Temperature (°C):	21

Calibration Data (After Recalibration)

		Gauge # 13978-107		Gauge # 13978-107	
Positive Polarity	**Standard**	**Channel A**	**% Difference**	**Channel B**	**% Difference**
	9,9 Pa	9,9 Pa	0,0%	9,9	0,0%
	25,7	25,7	0,0%	25,7	0,0%
	40,5	40,5	0,0%	40,6	0,2%
	50,0	50,0	0,0%	50,0	0,0%
	64,2	64,2	0,0%	64,2	0,0%
	91,6	91,6	0,0%	91,6	0,0%
	123,1	123,0	-0,1%	123,0	-0,1%
	180,4	180,3	-0,1%	180,3	-0,1%
	324,1	324,0	0,0%	324,0	0,0%
	522,2	522,3	0,0%	522,4	0,0%
	960,0	960,5	0,1%	960,5	0,1%
	1223,5	1223,0	0,0%	1223,1	0,0%

Figure 8: Excerpt from in-house calibration log for DG-700

DAKKS CALIBRATION OF PRESSURE GAUGES FROM BLOWERDOOR GMBH LABORATORY

For calibrations in a DAkkS-accredited laboratory, at least 9 points are calibrated in the range from -1250 to +1250 Pa, and the calibration cycle is repeated four times. If it is necessary to adjust the gauge, this should be done before the calibration. The example below (Figure 9) shows a DAkkS calibration log for the pressure differential gauge DG-700. M1 to M4 show the results of the calibration cycles.

Pressure at reference level UUT p_{Normal}	Reading UUT $p_{\text{Display (Output)}}$				Average M_{iw} = (M1+M2+M3+M4)/4	Deviation Δp $M_{iw} - p_e$	Repeatability b'_{mittel} = max(b'_{auf}, b'_{ab})	Hysteresis (|(M2-M10)-(M1-M10)|+|(M4-M30)-(M3 M30)|)/2	Extended uncertainty U
	M1	M2	M3	M4					
Pa	Pa	Pa	Pa	Pa	Pa	Pa	Pa	Pa	Pa
-1200,245	-1200,27	-1200,35	-1200,32	-1200,44	-1200,35	-0,10	0,14	0,06	0,68
-960,152	-960,94	-961,01	-961,02	-961,10	-961,02	-0,87	0,11	0,08	0,66
-720,078	-720,87	-720,96	-720,98	-721,05	-720,96	-0,89	0,08	0,08	0,64
-480,018	-480,42	-480,55	-480,57	-480,62	-480,54	-0,52	0,05	0,07	0,61
-239,979	-240,03	-240,18	-240,20	-240,28	-240,17	-0,19	0,02	0,04	0,59
0,000	-0,22	-0,36	-0,41	-0,48	-0,37	-0,37	0,00	0,05	0,57
240,099	239,78	239,64	239,57	239,51	239,63	-0,47	-0,01	0,06	0,59
480,119	479,89	479,77	479,67	479,62	479,74	-0,38	-0,03	0,07	0,61
720,130	719,94	719,84	719,70	719,65	719,78	-0,35	-0,05	0,08	0,63
960,134	959,82	959,75	959,55	959,52	959,66	-0,47	-0,08	0,11	0,66
1200,132	1199,50	1199,45	1199,20	1199,15	1199,33	-0,81	-0,11	0,11	0,68

Figure 9: Excerpt from a DAkkS calibration log for a pressure differential gauge DG-700.

The diagram below (Figure 10) illustrates the calibration results in graph form.

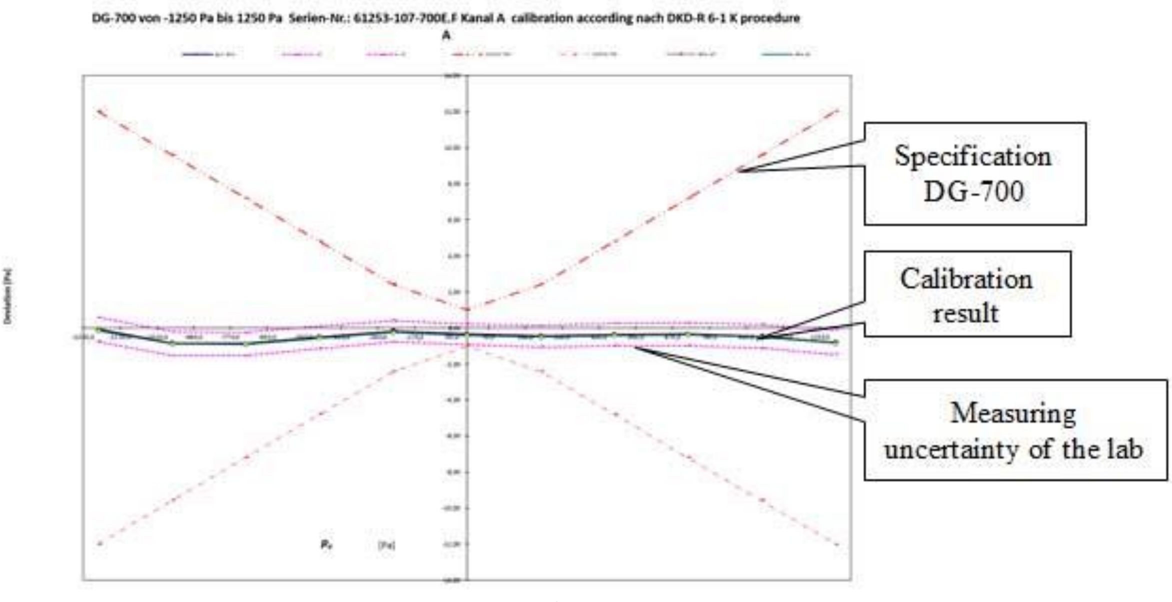


Figure 10: Results curve for the calibration in Figure 9.

The example log shows that the calibrated DG-700 is significantly more accurate than stated in the manufacturer's specification.

IN-HOUSE CALIBRATION OF BELLOWS GAUGES AT BLOWERDOOR GMBH

The calibration of a bellows gauge is presented using a Minneapolis BlowerDoor bellows model 4.1. The bellows gauge is calibrated with three measurement points for each of the six configurations (open bellows, plus five plates with various aperture sizes). The calibration points are found in the smallest, middle, and highest measurement ranges of each plate. That makes 18 total points per bellows. 50 Pascal is selected for the counterpressure (comparable with the building pressure during an airtightness measurement). For calibrations in France, 30 Pascal is the counterpressure setting.

Figure 11: Calibration station for measuring flow rates with a BlowerDoor bellows gauge.

Figure 12 shows the manufacturer's statement on measurement plate accuracy as a red line. The deviations from the calibration are visible as blue dots and the measurement uncertainty of the calibration station appears as an error bar.

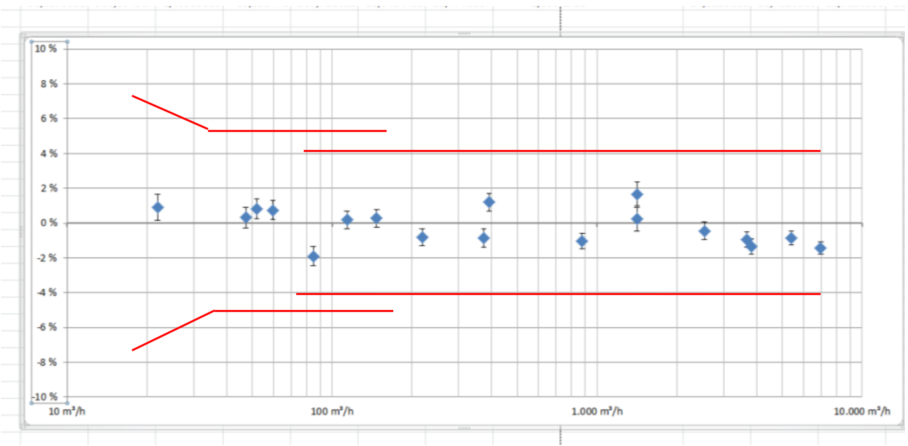


Figure 12: Calibration points of a bellows gauge [ ] with the measurement uncertainty of the chamber [ ] and the manufacturer's specifications [ ].

All measuring points, including measurement uncertainty of the testing chamber, are within specifications.

Calibration Certificate

page 2/2

Calibration parameters of fan

CE 3432	Offen	A	B	C	D	E
C	745,4	273,2	81,69	19,3	12,31	4,761
n	0,4848	0,4952	0,4968	0,5157	0,5032	0,5166

Measuring results

Ring	Chamber back pressure	Reference flow at ρ (UUT)	Fan flow	Deviation	Deviation	Uncertainty of meas. k=2
	Pa	m³/h	m³/h	m³/h	%	m³/h
Offen	49,5	7075,4	7013,2	-62,2	-0,89	± 25,6
Offen	49,9	5462,0	5313,9	-148,1	-2,79	± 21,1
Offen	50,1	3866,6	3798,3	-68,3	-1,80	± 16,6
A	49,6	3703,5	3650,2	-53,3	-1,46	± 16,2
A	49,8	2538,0	2507,2	-30,8	-1,23	± 12,9

Measurement uncertainty calculated according to GUM with a coverage factor k=2 and incudes the uncertainty of the process as well as of the device being tested (calibration object).

Figure 13: Excerpt from calibration log; here, 4 of 18 calibration points are pictured

Results - Declaration of conformity

	Offen	A	B	C	D	E
According to specification of manufacturer	X	X	X	X	X	X
Not according to specification of manufacturer						

Accuracy specification of manufacturer

Rings: Offen, A, B, C ± 4 % of reading

Rings: D, E ± 5 % of reading, but at least ± 1,7 m³/h

Figure 14: Example of a declaration of conformity for the bellows gauge

CONCLUSION

In order to achieve comparable results with airtightness measurements, gauges must be used that work within specific accuracy parameters. The user is responsible for ensuring that the devices fulfill the required accuracy specifications.

MULTIVARIANT MEASUREMENTS OF AIRTIGHTNESS OF MULTI-FAMILY BUILDING

Andrzej Górka[1], Radosław Górzeński[1], Michał Szymański[1*], Karol Bandurski[1]

[1] Poznan University of Technology
Institute of Environmental Engineering
pl. Marii Sklodowskiej-Curie 5
60-965 Poznan, Poland
**corresponding author: michal.szymanski@put.poznan.pl*

ABSTRACT

The paper presents airtightness measurements results of the multi-family building. The tests were carried out in several ways, the results obtained by different methods were compared and the likely reasons for the discrepancy of results were indicated. The object of measurements was a six storey building with 47 dwellings equipped with natural ventilation. Air inlet to the rooms through the window trickle vents controlled by relative humidity of air. Air is extracted through vertical extract ducts made of ceramic blocks.
The measurement for the whole building was carried out twice: with open and manually closed trickle vents. Additionally, the airtightness of each dwelling was measured. For several exemplary apartments, detailed measurements of airtightness were performed with and without pressure compensation in the adjacent zones.

KEYWORDS

airtightness measurement, multi-family building, multi-zone measurement

1 INTRODUCTION

The airtightness test is one of a few building envelope measurements used in practice which is quantitative, not just qualitative as e.g. infrared thermography. The airtightness test result may be a measure of the building design and construction quality and could also be used for the energy demand for heating and cooling analyses.
Although large number of single family houses were measured during last 40 years, only few results for multi-family residential buildings are available worldwide. Lack of measurement database makes it difficult to assess the energy impact of the infiltration and to formulate guidelines and requirements. In Poland almost half population live in multi-family building and even small corrections in buildings' envelopes and ventilation systems may result in great scale IAQ improvements and energy consumption reduction.
It is extremely difficult to measure occupied multi-family buildings as a whole. As a result single dwellings measurement are conducted. With adjacent zones not pressurised equally the result is influenced with interzonal leakages. In addition the leakages of common spaces are often quite different than dwellings', which makes the estimated result unreliable.
First sections of this paper provides a review of multi-family buildings market and the presentation of the airtightness measuring methods. Next, the case study building

characteristics, test preparation and equipment are presented. It is followed by measurement methodology description and the result presentation. Conclusions are presented in the final section.

2 MULTI-FAMILY BUILDINGS

The residential stock with 75% of floor space is the largest sector of EU building stock.
Within the residential segment 64% of floor space are single family houses, 36 % are apartment buildings. In Poland multi-family houses segment is bigger with 44%, compared with 56% of single family houses (Table 1).

Table 1: Residential floor space, % (Economidou, 2011)

Region	Multi-family houses	Single family houses
EU	36	64
Poland	44	56

IAQ are dependent on ventilation operation and the density of population. Matulska-Bachura (Matulska-Bachura, 2014) gives currently floor space per capita in Poland of 26,3 m²/person, which is quite low in comparison with western Europe countries (Table 2).

Table 2: Floor space per capita, m²/person (Economidou, 2011)

Region	Multi-family houses	Single family houses
North & West EU	36	50
South EU	31	41
Central & East EU	20	26
Poland	average 26,3	

There are more than 13 million dwellings in Poland, 5,3 million in single family houses and 7,7 million in multi-family houses. Almost half population live in multi-family houses. Matulska-Bachura (Matulska-Bachura, 2013) stated that in cities 72% live in multi-family houses. According to Dol (Dol, 2010) Poland is the country with one of the highest share of households living in overcrowded houses in EU.
Almost all multi-family residential buildings in Poland are equipped with natural ventilation driven by chimney stack effect. Last two decades large number of multi-family buildings were retrofitted with new, airtight windows with better heat loss protection. In parallel no new ways of air supply were proposed, only since 2008 the application of trickle vents is mandatory. Since free-market economy and individual heating cost accounting implementation, the IAQ in dwellings deteriorated. It is particularly observed in multi-family buildings, with small floor space per capita.
For the existing polish multi-family stock no reliable airtightness data exists. There are some reasons of such a state. Occupied multi-family residential buildings are extremely difficult or impossible to be measured as a whole. Access to apartments is limited and require cooperation with many building occupants. Testing prior to occupancy requires balancing the construction schedule, coordination with the owner and completion of the building (all leakage influencing components). Sets of blower door units, required for measurements, are not easy to gather in one place. Until last five years there were neither investor interest nor dedicated public funding for such a tests.
Few airtightness results for the multi-family residential buildings exist on European and North America market (Finch, 2009, Ricketts, 2013). Bailly (Bailly, 2013) presents airtightness database developed by the Centre d'Etudes Technique de l'Equipement de Lyon (CETE de Lyon) consist of 31 000 measurements but few entire multi-family buildings are measured. Finch (Finch, 2009) gives the number of 100 000 single houses' measurements documented

and less than 500 for multi-unit buildings worldwide. Building envelopes are often far different in construction and results from different countries are inadequate to compare.

3 MULTI-FAMILY BUILDINGS AIRTIGHTNESS MEASUREMENT

As multi-family buildings are difficult to measure, a guarded zone test method (Figure 1, 2), called also guarded zone pressurization technique or balanced fan pressurization, is used to measure specific single zone air leakages in multi-zone building. To reduce the influence of airflows between zones, both the specific zone (room, dwelling, floor, etc.) and any adjacent conditioned zones (beside, above, below) should be pressurized to the same equal test pressure (Walther, 2009, Ricketts, 2013, Hult, 2014, Finch, 2009).
Some authors (Ricketts, 2013, Finch, 2009) points out that during guarded zone test flow rate and pressure caused by one fan can affect the flow rates of the other fans. In large building and in adverse weather conditions getting multiple fans to operate in equilibrium may be challenging. Automatic fan control helps getting the equilibrium quicker.

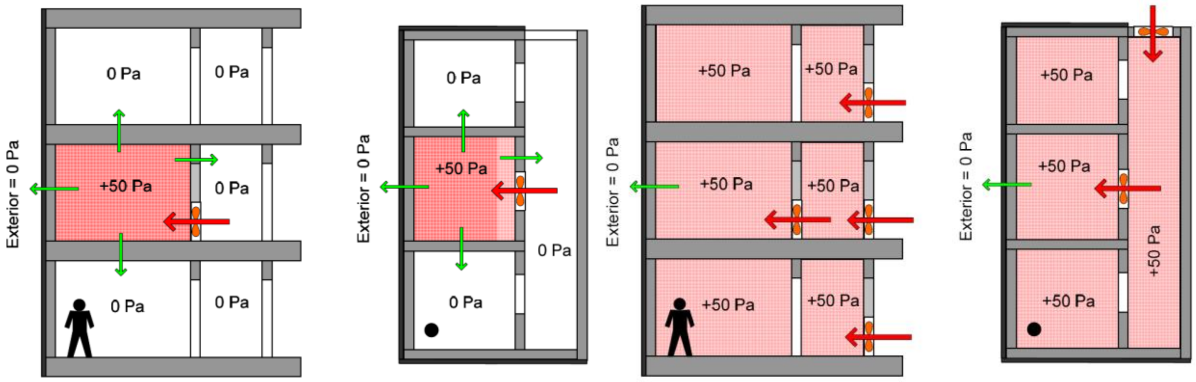


Figure 1: Single zone test with adjacent zones open to exterior (two left figures: section and plan view of test floor) and guarded zone test with adjacent zones pressurised to equal pressure (right figures) (Ricketts, 2013)

It is essential that the larger and leakier zones have the strong influence of a whole building result. For multi-family building wrong results could be obtained when determining building airtightness with use of dwellings results (or even single dwelling result) neglecting the influence of common spaces (which can be very leaky and in practice are impossible to be measured separately).

Figure 2: Simultaneously measurement of 9 apartments in multi-family residential building (NRCERT, 2011)

One may measure specific floor with adjacent floor pressurized to the equal test pressure. Anyway in buildings with large, inaccessible inter-floor leakages that cannot be sealed (such as elevator and mechanical shafts) the sum of the leakages measured for each floor does not give the real total leakage of the building (Bahnfleth, 1999). Proskiw (Proskiw, 2001) provides procedure of measuring and pressurization of single zone, with adjacent zones not pressurized, where the pressures induced in adjacent zones are measured and internal flows

are calculated based on pressure differences. Method is more complicated and requires sophisticated staff and more equipment.
Measuring a zone with adjacent zones not pressurized (Figure 1, left) gives the result which include air leakages to adjacent zones (Genge, 2009, Zhivov 2009). These leakages have no influence on buildings heat losses and should be excluded from the result. Ricketts (Ricketts, 2013) find interior separators often less airtight than the exterior enclosure. Walther (Walther, 2009) describes the FLiB (Fachverband Luftdichtheit im Bauwesen e.V.) method developed for subsidies verification procedure of measuring at least 20% of dwellings in building separately. Apartments on different floors should be chosen, at least one on the top floor, one in-between floor and one on the ground floor. The adjacent zones should be open to outdoors, to minimize the airtight influence of these zones. Weighted average from the results of the separate zones can be calculated, based on the volume (or other basis provided that it is consistent with the airtightness metric used) and is equivalent to the value that would be measured on the building as a whole. Particular zones can be up to 30% leakier than the limit value for the whole building. It is noted that the whole building measurement is more reliable, as the method is based on an assumption of zones' similar airtightness properties.
Due to difficulties in measuring the whole residential buildings the ATTMA standard (ATTMA, 2010) gives also the possibility of testing at least 20% zones, in this case of building's exterior walls area, but concludes that the leakages limit should be 10% smaller than that for the whole building.
As concluded above, the leakages of common spaces play very important role in these assumptions. With very leaky zone not selected for the test (usually common spaces) described methods could lead to wrong results. Lift and technical shafts, halls and stairwells ventilated naturally are often responsible for cause significant leaks. Walther (Walther, 2009) noticed that there is lack of feed-back from the use of mentioned method and it seems that these methods have been derived according to expert intuition but without solid argumentation.
There is the possibility to use single blower door for zone pressurization in multi-zone building but the purpose of the test is different. Armstrong (Armstrong, 1996) describes method of sequence blower door tests with different building component sealed in consecutive measurements. Each change in airflow is attributed to the corresponding set of components that were sealed. Gorzenski (Gorzenski, 2014b) used this method to determine the average leakage through the wall in aquapark.
Performing and interpreting results from air leakage tests are more complicated and more time consuming in multi-zone buildings than in single-zone ones (Hult, 2014, Ricketts, 2013, Finch, 2009). There are no existing standards for multi-zone building airtightness testing, although some standards (ISO 9972, 1996, EN 13829, 2000) allow to measure air leakages to outdoor for single zone in multi-zone building with equal pressure induced in adjacent zones.
Hult (Hult, 2014) examined uncertainty for multi-zone air measurements. Uncertainty in leakage to outside due to pressure fluctuation and calibration error in guarded test result is relatively small. If the interzonal leakage area is small relative to the leakage area to the outside then the uncertainty in the leakage measurement is 4%. If the interzonal leakage area is increased, the uncertainty in the measured leakage is 14%. But leakages in interconnected zones that are not pressurized during the test may have much more substantial impact on the order of 30÷100% of the leakage directly to the outdoors (Hult, 2014). Gorzenski (Gorzenski, 2014a) presented large aquapark (volume of 200.000 m^3) multi-zone airtightness test results doubled in case of no adjacent zones pressurized, compared with the guarded zone test results. Still some authors (Ferdyn-Grygierek, 2012) give examples of measurements made for single dwellings and rooms with adjacent zones not equally pressurized and even suggesting the resulted airflow could be used for calculating the specific window leakages.

4 CASE STUDY BUILDING

An unoccupied multi-family residential building located in Poznan, Poland was chosen for the research (Figure 3). It was under construction, but all envelope components which have the influence on airtightness were completed (roof, windows with windowsills, doors, finished envelope plastered from outside and inside, once painted, floors prepared to be covered, electrical and HVAC plumbing mounted). Tests were performed during autumn and wintertime in late 2013.

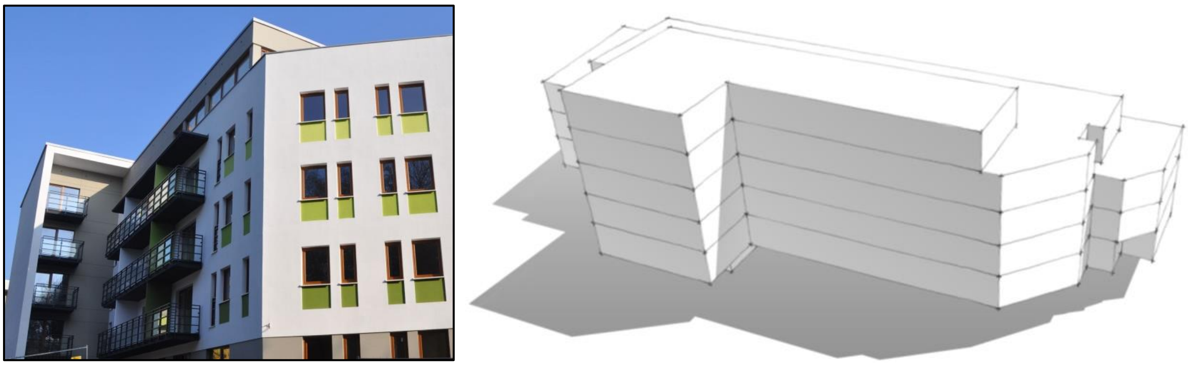


Figure 3: Case study multi-family residential building and its 3D model

4.1 Building characteristics

The object is a multi-family six storey (underground garage and five aboveground floors) residential building built in 2013. Outer walls are made with 24 cm thick silicate bricks and insulated with 15 cm thick expanded polystyrene. Walls are finished with gypsum plaster layer from inside and with silicate thin-layer plaster on polystyrene from outside. The inner walls are made with 8÷24 cm thick silicate brick or concrete. Floors are made with beam-and-block concrete, roof is wooden truss system covered with heat-weldable roofing membrane. Floors are covered with concrete topping.

Building is equipped with natural ventilation system with trickle vents mounted into triple pane window frames and with ceramic blocks extract ducts with diameter of 150 mm dedicated for each apartment (Figure 4).

Figure 4: Trickle vents (left) and ceramic blocks used for extract ducts (right)

The floor area is 33,400 ft^2 (3,100 m^2) and the interior volume is 277,200 ft^3 (7,850 m^3). Total enclosure area including exterior walls, roof and below grade surfaces is 39,300 ft^2 (3,650 m^2).

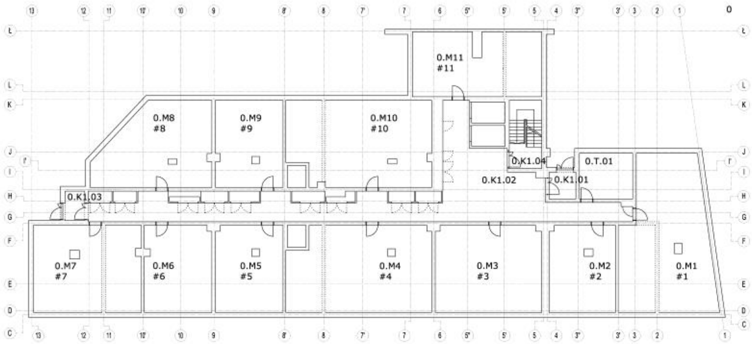


Figure 5: Ground floor layout

There are 47 dwellings with floor area of 25.6÷81.7 m² (average 53.6 m²). Floors consist of a central hallway with apartments on each side (Figure 5). There are one staircase, two elevator shafts and several technical shafts, all running vertically the full height of the building. There are 131 trickle vents mounted into window frames. With high outdoor (~90%) and indoor (~75%) air relative humidity it was assumed and observed that during test trickle vents were fully open (Figure 6).

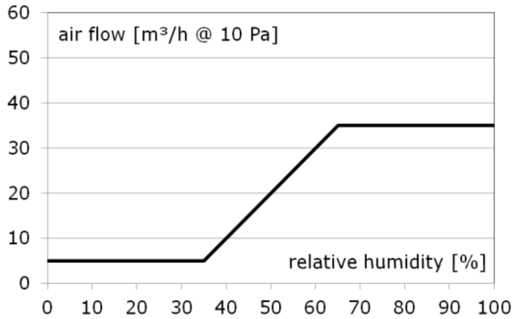


Figure 6: Trickle vents air flow as a function of relative humidity characteristics by manufacturer

4.2 Preparation for testing

All building zones (dwellings and common spaces) except unconditioned garage space on underground floor were included in the whole building test area. On underground floor only staircase and lift shafts are included into conditioned zone and air tightly separated from the garage. Garage is connected to the roof level with some ducts (garage, smoke and trash room ventilation). Ducts are made with steel and equipped with fire dampers, which were closed during the test.
Measurements were conducted with accordance to EN 13829. For the whole building test all ventilation extract ducts were shut with use of 146 inflatable inner rubber bladder (Figure 7).

Figure 7: Extract ducts closed with use of inflatable inner rubber bladder

Figure 8: Chimneys top with gaps between chimney walls and ventilation duct

Windows and doors to outside were closed, inner doors left open. Sewer traps were filled with water or sealed. Trickle vents were set to close and open position in successive tests.
During preparation huge gaps between chimney walls and garage ventilation duct (Figure 8) were observed at roof level. Roof hatch window was found to be leaky one.

4.3 Equipment

A total of 3 Minneapolis Blower Door (BD) were used for the tests. During whole building test set was mounted in the balcony door on the third floor (Figure 9).

Figure 9: Minneapolis Blower Door set used for the tests

5. CASE STUDY MEASUREMENTS AND RESULTS

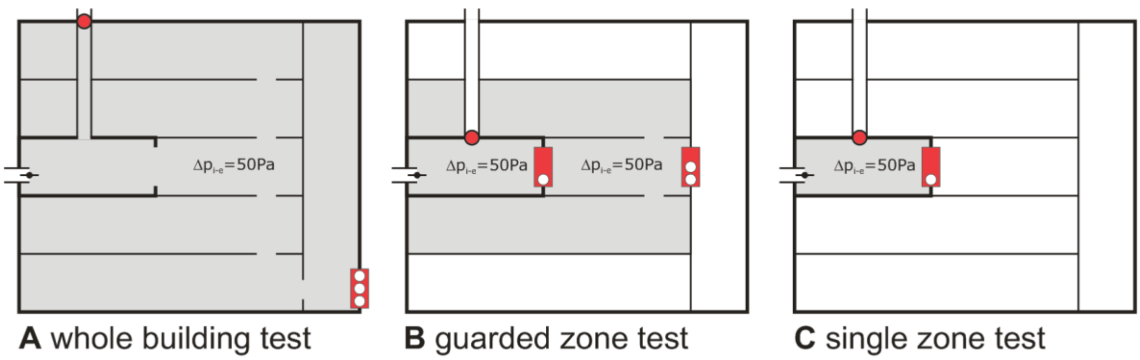


Figure 10: Measurements configurations (pressurized zone marked with grey colour)

Several measurements (Figure 10) were conducted:

- A - whole building airtightness test with use of 3 BD units, extract ducts were sealed from the top on roof level, variants:
  - both pressurisation and under-pressurisation,
  - both closed and open all trickle vents,
- B - guarded zone test of selected 5 different dwellings located on third floor, extract ducts sealed at the dwelling level, trickle vents closed, under-pressurisation only, use of 1 BD for zone pressurising and 2 BD for three floors (above, current and below) pressurising,
- C - single zone test for all 47 apartments in building, no adjacent zones pressurization, extract ducts sealed at the dwelling level, trickle vents closed, under-pressurisation only, 1 BD used.

Whole building airtightness tests (A) were conducted in multi-point sequence according to EN 13829 in two variants: first one with all trickle vents open and second one with all of them closed. n_{50} values were found to be 1,20 h^{-1} for vents closed and 1,55 h^{-1} for open. The difference in V_{50} airflow was only 2800 m³/h, which for 131 trickle vents gives the difference in airflow of 21,4 m³/h between closed and open vent at the pressure difference of 50 Pa. Based on Figure 6 the difference should be 30 m³/h at the pressure difference of 10 Pa and 88 m³/h at the pressure difference of 50 Pa, assuming flow exponent of ⅔. Obtained airflow is more than 4 times smaller than it should be, based on manufacturer data. Further research with trickle vents in pressure/temperature/humidity chamber would be essential.

Table 3: Whole building airtightness test results (A)

Trickle vents	Pressure	V_{50} [m³/h]	n_{50} [h^{-1}]	q_{50} [m³/(h·m²)]
closed	under-pressure	9310	1.19	2.55
	overpressure	9493	1.21	2.60
	average	9402	**1.20**	**2.58**
open	under-pressure	12492	1.59	3.42
	overpressure	11874	1.51	3.25
	average	12183	**1.55**	**3.34**

Building was found to be air tight, even with trickle vents open. One should keep in mind that extract ducts were sealed from the top on roof level during the test.

Results of guarded zone (B) and single zone tests (C) for selected 5 different apartments located on third floor are presented in Table 4. Extract ducts were sealed at the dwelling level, trickle vents were closed. In guarded zone test (B) equal pressure in adjacent zones (three floors: above, current and below) was generated with 2 Blower Doors.

Table 4: Guarded (B) and single (C) zone test results

Dwelling	Floor area [m²]	Volume [m³]	B/C V_{50} [m³/h]	B/C n_{50} [h^{-1}]	C/B diff. [%]
2.M8	75.4	190.8	130/146	0.68/0.77	12
2.M9	65.8	166.4	100/138	0.60/0.83	38
2.M7	36.2	91.6	60/72	0.65/0.79	20
2.M3	57.2	144.6	90/160	0.62/1.11	78
2.M2	60.5	152.9	100/127	0.65/0.83	27
average				**0.64/0.86 h^{-1}**	**35 %**

Weighted (per volume) average air leakages are about 35% higher (C/B factor) in a case of single zone (n_{50}=0.86 h^{-1}) compared to guarded zone method (n_{50}=0.64 h^{-1}). It is due to taken interzonal airflows into account.

Single zone tests (C) were conducted in multi-point sequence according to EN 13829 for all 47 dwellings in building. Single point measurement (50 Pa) and under-pressurisation was used. Adjacent zones were not pressurized. Extract ducts were sealed at the dwellings level, trickle vents were closed. Total floor area of all dwellings is 2520 m² and the inner volume is 6345 m³. Common spaces (staircases, lift shafts, corridors) floor area is 595 m² and the interior volume is 1505 m³.

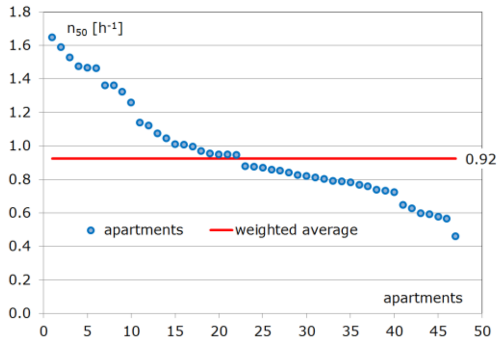


Figure 11: n_{50} values for apartments - single zone test (C)

Results of 47 single zone tests (C) for all dwellings are presented on Figure 11. The weighted (per volume) n_{50} average is 0,92 h^{-1}.

Table 5: Results of single zone tests for 47 dwellings (C) and other estimated (*) values

#	Area	Volume [m³]	Description	V_{50} [m³/h]	n_{50} [h^{-1}]
1	47 dwellings	6345	measured, single zone (C) - FLiB	5893	0,92
2	47 dwellings	6345	#1 reduced by C/B 35% factor	4365*	0,68*
3	whole building	7850	measured, whole building (A)	9310	1,20
4	common spaces	1505	subtraction #3 - #2	4945*	3,28*
5	whole building	7850	ATTMA (#1 +10%)	8016*	1,02*

If one assumes obtained result (#1) is 35% higher (C/B factor), due to internal leakages included in result (Table 4), the corrected V_{50} value for all dwellings is 4365 m³/h (n_{50}=0,68 h^{-1}). Estimated (weighted per volume average) n_{50} value for the whole building, with 100% dwellings tested (more reliable result than required 20%), would be:

- 0,92 h^{-1} - FLiB method (#1),
- 1,02 h^{-1} - ATTMA method (#5).

The most accurate result, for the whole building (A, #3), was n_{50}=1,20 h^{-1}. Both methods would result in underestimation of n_{50} value: 15% for ATTMA and 23% for FLiB method. The common spaces n_{50} value was found to be 3,28 h^{-1}. The main leakages are gaps in ventilation and technical shafts.

6. CONCLUSIONS

Results of airtightness tests of multi-family residential buildings and its 47 dwellings were presented. The whole building (one zone) testing (A) was found to be the only one method resulting in real value for the building. Dwellings were found to be airtight with average of n_{50}=0,64 h^{-1} for 5 dwellings and guarded zone test and n_{50}=0,92 h^{-1} for all dwellings and single zone test (35% higher). Influence of relatively small (19% of internal volume), but very leaky (n_{50}=3,28 h^{-1}) common spaces results in n_{50}=1,20 h^{-1} for the whole building.
Using methods for estimating the whole building airtightness based on single zone test for dwellings results in underestimation of n_{50} value. It was due to very leaky common spaces in comparison with dwellings.

Trickle vents were found to be 4 times more tight than it should be basing on manufacturers data: at 50 Pa pressure difference airflow was found to be 21,4 m³/h per vent instead of 88 m³/h.
Single zone test, with adjacent zones not pressurized, gives the result which contains outdoor and indoor leakages. Determination of zones' outdoor leakages is possible with guarded test method. Anyway it is practically impossible to measure the leakages of common spaces separately, that is why only whole building (one zone) airtightness test only gives the reliable result. On the other hand, measurement of multi-family buildings as a whole is practically possible only if building is unoccupied.

7. REFERENCES

Armstrong, P., Dirks, J., Klevgard, L., et al., (1996). *Infiltration and ventilation in Russian multi-family buildings*, Proceedings of the 1996 Summer Study on Energy Efficiency in Buildings, Washington

ATTMA, (20062010). *ATTMA Technical Standard 1. Measuring Air Permeability of Building Envelopes*, Air Testing and Measurement Association, The British Air Tightness Testing and Measurement Association 2006

Bahnfleth, WP., Yuill, GK., Lee, BW., (1999). *Protocol for field testing of tall buildings to determine envelope air leakage rate*, ASHRAE Transactions, Seattle

Bailly, A., Jiang, Y., Guyot, G., Desfougères, F., (2013). *Preliminary analysis of French buildings airtightness database*. Proceedings of the 34th AIVC Conference, Athens

Dol, K., Haffner, M. (2010). *Housing statistics in the European Union 2010*, Delft University of Technology, Delft

Economidou, M., et al (2011). *Europe's buildings under the microscope. A country-by-country review of the energy performance of buildings*. Buildings Performance Institute Europe (BPIE), Brussel

EN 13829:2000 Thermal performance of buildings - determination of air permeability of buildings - fan pressurization method, European Committee for Standardization,

Ferdyn-Grygierek, J., Baranowski, A. (2012). *Szczelność i wymiana powietrza budynkach mieszkalnych i użyteczności publicznej*, conference proceedings Kompleksowa diagnostyka cieplna in situ budynków, Politechnika Śląska, Gliwice

Finch, G., Straube, J., Genge, (2009). *Air leakage within multi-unit residential buildings: testing and implications for building performance*, 12th Canadian Conference on Building Science and Technology proceedings, Montreal

Genge, C. (2009). *Controlling Air Leakage In Tall Buildings*, ASHRAE Journal 51(4): 50-60

Górzeński, R., Szymański, M., Górka, A., (2014). *Large buildings airtightness measurements using ventilation systems*, Proceedings 35th AIVC Conference, Poznań

Górzeński, R., Szymański, M., Górka, A., Mróz, T. (2014). *Airtightness of Buildings in Poland*, International Journal of Ventilation 12(4): 391-400

Hult, E., Sherman, MH. (2014). *Estimates of uncertainty in multi-zone air leakage measurements*, International Journal of Ventilation, 12(4): 359-368

ISO Standard 9972 Thermal performance of buildings - determination of air permeability of buildings - fan pressurization method, International Organization for Standardization, 1996

Matulska-Bachura, A., et al. (2013). *Zamieszkane budynki. Narodowy spis powszechny ludności i mieszkań 2011*, Główny Urząd Statystyczny, Warszawa

Matulska-Bachura, A., et al. (2014). *Gospodarka mieszkaniowa w 2013r.*, Główny Urząd Statystyczny, Warszawa

NRCERT, (2011). *Multifamily Blower Door Process*, New River Center for Energy Research and Training, proceedings DOE National Weatherization Training Conference, New Orleans 2011

Proskiw, G, Parekh, A. (2001). *A Proposed Test Procedure for Separating Exterior Envelope Air Leakage from interior Partition Air Leakage*. Performance of Exterior Envelopes of Whole Buildings Conference proceedings, Atlanta

Ricketts, L., Finch, G., Bombino, R., (2013). *Air Leakage Control in Multi-Unit Residential Buildings*, RDH Building Engineering Ltd., Vancouver 2013

Walther, W., Rosenthal, B., (2009). Airtightness Testing of Large and Multi-family Buildings in an Energy Performance Regulation Context, ASIEPI

Zhivov, A., Pietilainen, J., Reinikainen, E., Schmidt, F., Woody, A., (2009). *IEA ECBCS ANNEX 46 Energy Process Assessment Protocol*, Energy Branch US Army Corps of Engineers

STATUS OF THE DEVELOPMENT OF THE CEN AND ISO STANDARDS ON ENERGY PERFORMANCE OF BUILDINGS ASSESMENT PROCEDURES

Jaap Hogeling

ISSO, Manager international projects and standards
Chair of CEN TC 371 Program Committee on EPBD; Chair of JWG of ISO TC 163 & 205 advisory group on coordination of ISO and CEN Work on EPB
Fellow of ASHRAE, Fellow of REHVA
ISSO: Dutch Building Services Knowledge Centre
P.O Box 577 | 3000 AN Rotterdam / The Netherlands
j.hogeling@isso.nl

Abstract

The Recast-EPBD[1] requires an update of the current (2007/2008) set of CEN-EPB standards. This update work started in 2012 and will result in a new set of CEN-EPB standards.. Where possible this work will be done parallel with ISO. This project is based on EU-Mandate 480. This mandate accepted by CEN, requires a really out of the box thinking approach of the standard developers. This project is coordinated by CENTC371 the "Program Committee on EPBD" and is considered to be a step forward in progressing towards European Energy Codes for Buildings. This second generation of EPB standards aims on more comprehensive standards, a clear split between informative text in Technical Reports and normative text in Standards, attached excel files to illustrate the calculation procedures etc.. The EPB set of standards and technical reports will support the holistic approach needed for the Nearly Zero Energy Buildings (nZEB) and high performance energy renovation of the existing building stock. CEN proposes a nZEB definition, worked out a common, clear, unambiguous assessment structure and the related standards to calculate the very limited amount of (primary) energy required by nZEB.

The modular structure of EPB standards is flexible in order to take into account national, regional and regional choices. An approach has been introduced, via the so-called Annex A and B in all EPB standards. Annex B is an informative Annex and includes all default values, choices and options needed to use the standard. Normative Annex A includes empty tables for these needed values, choices and options, this empty template shall be used by National Standard Bodies (NSB) (or recognised local, regional or national authorities) to declare these values, choices and options to be followed under their jurisdiction. This approach allows maximal flexibility and transparency in applying the EPB standards. If published by the NSB's These filled in Annexes conform Annex A are indicated as National Annexes.

It is expected that Formal Voting drafts of all EPB standards will be ready before April 2016. After the EPB standards are accepted the publication by the end of 2016 seems possible.

[1]EPBD: DIRECTIVE 2002/91/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 16 December 2002 on the energy performance of buildings.
Recast-EPBD: DIRECTIVE 2010/31/EU OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 19 May 2010 on the energy performance of buildings; (recast).

Keywords:

EU Energy Performance Buildings Directive; CEN ISO Standards EPBD EPB

1. **Introduction**

Analyses regarding the use of the in 2007/2008 published set of CEN-EPB[2] standards and the requirements set out in the recast-EPBD showed the clear need for a second EU mandate to CEN in order to improve these standards. The revision will improve the accessibility, transparency, comparability and objectivity of the energy performance assessment in the Member States, as mentioned in the EPBD.

The "first generation" CEN-EPB standards were implemented in many EU Member States "in a practical way". Typically: partly copied in "all in one" national standards or national legal documents, mixed with national procedures, boundary conditions and input data.

For a more direct implementation of the EPB standards in the national and regional building regulations, it is necessary to reformulate the content of these standards so that they become unambiguous (the actual harmonized procedures), with a clear and explicit overview of the choices, boundary conditions and input data that can or needs to be defined at national or regional level. This implies that the current set of CEN-EPB standards is improved and expanded on the basis of the recast-EPBD.

The standards shall be flexible enough to allow for necessary national and regional differentiation to facilitate Member States implementation. Such national or regional choices remain necessary, due to differences in climate, culture & building tradition and building typologies, policy and/or legal frameworks.

2. Work in progress, the last phase of the on-going work on the EPB standards

The EPB standards have been developed by the following CEN/TC’s:

- TC 089 Thermal performance of buildings and building components;
- TC 156 Ventilation for buildings;
- TC 169 Light and lighting systems;
- TC 228 Heating systems for buildings;
- TC 247 Building automation, control and building management;
- TC 371 Project Committee on Energy Performance of Buildings.

These TC’s are responsible for the technical content of EPB standards to be revised. CEN/TC 371, the overall responsible coordinating committee, also ensuring that the timetable will be met and that the basic principles and rules, the modular approach and the foreseen improvements of the current set of EPB standards, are in line with the targets indicated and meeting the expectations of the end users.

CEN/TC 371 formulated common Basic Principles (CEN/TS 16628:2014) on the required quality, accuracy, usability and consistency and a common format for EPB standards, including a systematic, hierarchic and procedural description of options, input/output variables and relations with other standards and elaborated a unique hierarchic system for the EPB standards.

CEN/TC 371[3] prepared the Basic Principles (BP) and the supporting Detailed Technical Rules (DTR) (CEN/TS 16629:2014), as basis and guidance for the total set.

[2]In this paper EPB stands for “Energy Performance of Buildings” the D for the EU-Directive is intentional deleted in relation to the standards. The EU-directive is of great importance for the EU-member states however these CEN standards could become ISO standards as well and it is more appropriate to use just EPB.

[3]CEN/TS 16628:2014 Energy Performance of Buildings - Basic Principles for the set of EPBD standards
CEN/TS 16629:2014 Energy Performance of Buildings - Detailed Technical Rules for the set of EPB-standards

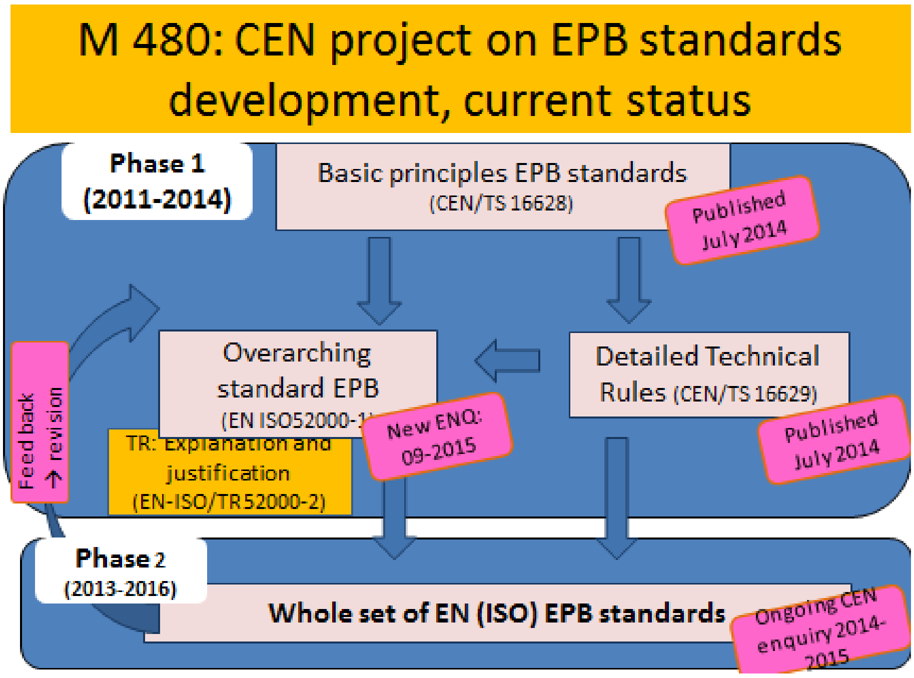


Figure 1 Current status

3. The Process

The mandate M/480 explicitly requests for identification and prioritisation of items for revision and gaps in the current set of standards in consultation with the EU member states (MS).
The expert team working on the program within the CENTC371, the core group of the CEN/TC team-leaders responsible for the EPBD work in these 5 TC's and TC371, here indicated as the CTL, works closely together with experts from Building Regulators side (the Liaison Committee (LC)).
This close cooperation made it possible to focus at the requirements to be fulfilled to make the standards fit for referencing by legislative authorities.
Based on this working structure CEN/TC 371 prepared the general frame for the package of standards. This includes both the standardised calculation structure and the guidance for the drafting the individual EN EPB standards..

The following, general principles are valid for the set of EPB standards :

1. The complexity of the building energy performance calculation requires a good documentation and justification of the procedures. Informative text is required but it will be separated from actual normative procedures to avoid confusion and unpractical heavy documents. Therefore, each EPB standard (or sometimes a close connected set of) shall be accompanied by a Technical Report where all related informative material will be concentrated.[4]
2. The complexity of the building energy performance calculation requires also a very good coordination and testing of each calculation module. Therefore, each EPB standard shall be accompanied by a spread-sheet where the proposed calculation algorithms and data input/output are tested and proved to be consistent. A Software Tool team checks the calculation modules of the total set of EPB standards. With this excel based software it will be possible to assure that the in/output files of the various connected EPB standards are valid. The relation of the above mentioned set of draft documents and the process setup is illustrated in figure 2

[4] Either as a separate TR or if very limited as an informative annex to the standard. It is also possible that a TR will cover more standards.

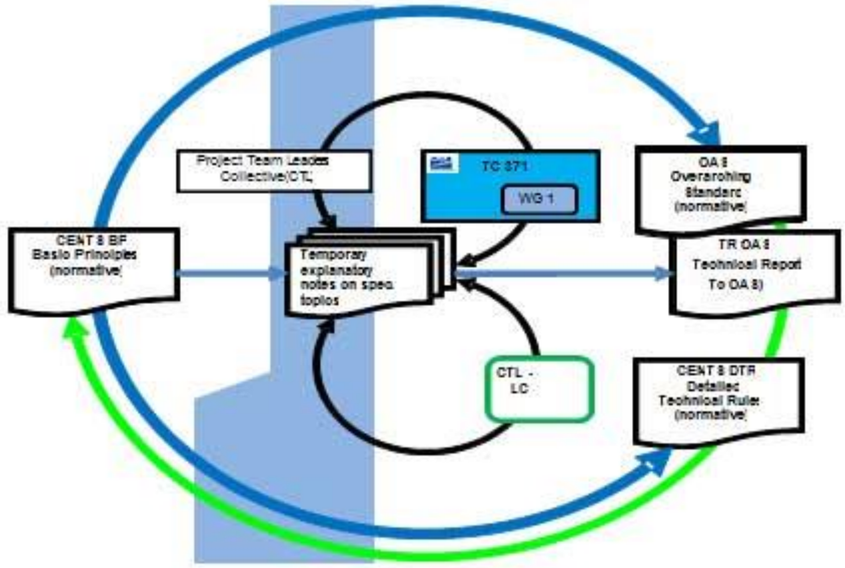


Fig. 2 – Iterative development process around the coordinating CENTC371 and inter-relation between the documents as has been developed.

4. The deliverables of CENTC371

4.1 CEN/TS Basic Principles

CEN/TS 16628:2014 Energy Performance of Buildings - Basic Principles for the set of EPBD standards. This TS provides a record of the rationale, background information and all choices made in designing the EPB package. These basic principles are based on the analysis of the weak points within the first generation EPB package and on an evaluation of requirements by the Regulating Authorities and the outcome of the IEE-project CENSE (see http://www.buildup.eu and http://www.iee-cense.eu/).
The TS Basic Principles provides guidance on the required quality, accuracy, usability and consistency of each standard and the rationalisation of different options given in the standards, providing a balance between the accuracy and level of detail, on one hand, and the simplicity and availability of input data, on the other.

4.2 CEN/TS Detailed Technical Rules

CEN/TS 16629:2014 Energy Performance of Buildings - Detailed Technical Rules for the set of EPB-standards. This TS is based on the CEN/TS BP and provides mandatory detailed technical rules to be followed in the preparation of each individual EPB standard. This is in addition to the CEN drafting rules and complementary to the Overarching Standard (former prEN15603 and current draft-ISO 52000-1) in this article indicated as OAS. The OAS, containing the common terms, definitions and symbols and the overall modular structure for the set of EPB standards.The DTR gives a common format for each standard, including a systematic and hierarchic structure to pinpoint the position of the standard within the framework of EPB standards and procedural description of options, input/output variables. THE CEN/TS DTR includes guidance for:

-a clear separation of the procedures, options and default data to be provided as default CEN option in an annex B but also allowing for national or regional choices conform the normative annex A of each of the EPB standard (where appropriate);

-a specification of the input data, also indicating the source of the data if this is the output calculated according to another EPB standard or related product standard;

-a specification of the intended output that is intended to provide the energy performance assessment results, the related data necessary for their proper interpretation and use, and all relevant information documenting the relevant boundary conditions and calculation or measurement steps.

-an informative CEN Technical Report, accompanying each standard[5], according to a common structure, comprising at least the results of internal validation tests (such as spread sheet calculations for testing and demonstrating the procedures), examples and background information. Almost all informative parts of EPB standards will be in these technical reports.

3.3 Energy performance of buildings-Overarching standard EPB; the former FprEN 15603: 2014 and current ISO DIS 52000-1

This standard (OAS) specifies a general framework for the assessment of the overall energy use of a building, and the calculation of energy ratings in terms of primary energy, using data from other EPB standards, providing methods for calculating the energy use of services within a building (heating, cooling, humidification, dehumidification, domestic hot water, ventilation, and lighting). This assessment is not limited to the building alone, but takes into account the wider environmental impact of the energy supply chain.

The OAS handles the framework of the overall energy performance of a building, covering inter alia:

1. common terms, definitions and symbols;
2. building and system boundaries;
3. building partitioning;
4. unambiguous set of overall equations on energy used, delivered, produced and/or exported at the building site, near-by and distant;
5. unambiguous set of overall equations and input-output relations, linking the various elements relevant for the assessment of the overall energy performance of buildings which are treated in separate standards;
6. general requirements to standards dealing with partial calculation periods;
7. general rules in setting out alternative calculation routes according to the calculation scope and requirements;
8. rules for the combination of different partitioning.

The OAS provides a systematic, clear and comprehensive, continuous and modular overall structure on the integrated energy performance of buildings, unlocking all standards related to the energy performance of buildings.

The overall framework provided by the OAS will work as the "***Backbone***" (see figure 3) of the set of EPB standards, it facilitates a step-by-step implementation by the user, taking also into account the nature of each procedure identifying the typical type of user. More information is given in a Technical Report accompanying the OAS. The justification for the CEN defaults and options are provided in this TR (draft ISO TR 52000-2).

Current (July 2015) status: this ISO DIS 52000-1 will be published for enquiry by September-2015. The Enquiry will close by December 2015, after processing the possible comments it is expected that a Formal Voting draft will be ready before April 2016. After the standard is accepted publication by the end of 2016 seems feasible .

[5]This to significantly reduce the length of the standards and strengthen their focus, thus facilitating the adoption (including translation) in national/regional regulations.

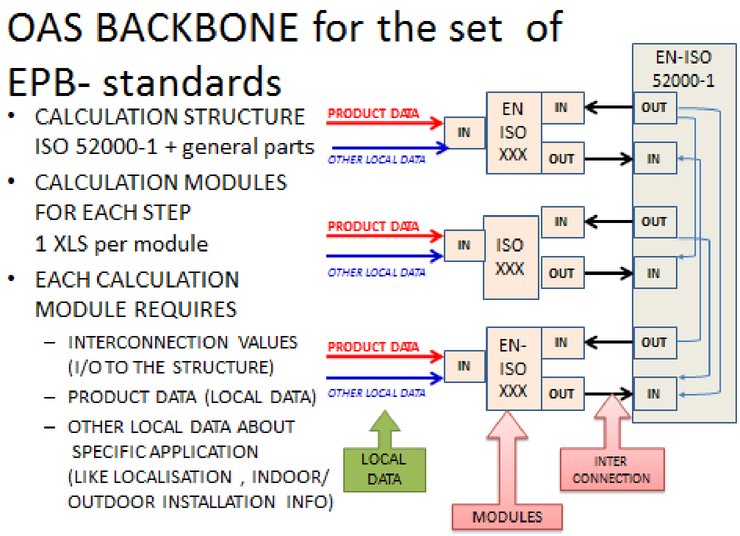


Figure 3 The OAS as backbone for the set of EPB standards

4.4 Draft ISO TR 52000-2 (former prCEN/TR 15615:2014) Energy Performance of buildings - Accompanying Technical Report on draft OAS.

This draft-TR contains information to support the correct understanding, use and national implementation of this standard.
This draft is expected to be published at the same time as the ISO-DIS OAS.

5 Hierarchic numbering system - Modular structure

The setup of a coherent and hierarchically numbered system of EPB standards is a requirement. Given the fact that not all standards will be ready for parallel ISO enquiry or publication and that standard numbering system in CEN doesn't allow this, a modular structure was developed, allowing for addressing documents given hierarchic positioning in that structure. By adding the identification code of a specific cell of the modular structure (see Figure 4 & 5) the purpose of a standard (and/or specific clauses of the standard) can be identified easily.

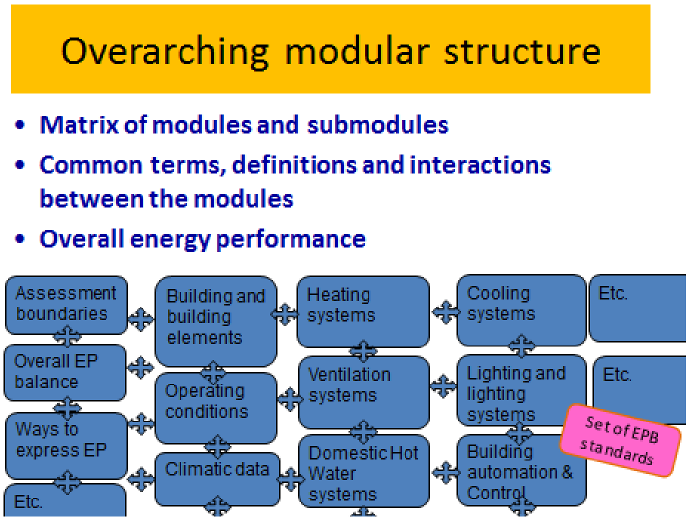


Figure 4 Overarching Modular structure

Overarching		Building (as such)		Technical Building Systems										
	Descriptions		Descriptions		Descriptions	Heating	Cooling	Ventilation	Humidification	Dehumidification	Domestic Hot water	Lighting	Building automation & control	PV, wind, ..
sub	**M1**	**sub**	**M2**	**sub**		**M 3**	**M 4**	**M 5**	**M 6**	**M 7**	**M 8**	**M 9**	**M 10**	**M 11**
1	General	**1**	General	**1**	General									
2	Common terms and definitions; symbols, units and subscripts	**2**	Building Energy Needs	**2**	Needs									
3	Applications	**3**	(Free) Indoor Conditions without Systems	**3**	Maximum Load and Power									
4	Ways to Express Energy Performance	**4**	Ways to Express Energy Performance	**4**	Ways to Express Energy Performanc e									
5	Building Functions and Building Boundaries	**5**	Heat Transfer by Transmission	**5**	Emission & control									
6	Building Occupancy and Operating Conditions	**6**	Heat Transfer by Infiltration and Ventilation	**6**	Distributio n & control									
7	Aggregation of Energy Services and Energy Carriers	**7**	Internal Heat Gains	**7**	Storage & control									
8	Building Partitioning	**8**	Solar Heat Gains	**8**	Generation & control									
9	Calculated Energy Performance	**9**	Building Dynamics (thermal mass)	**9**	Load dispatching and operating conditions									
10	Measured Energy Performance	**10**	Measured Energy Performance	**10**	Measured Energy Performanc e									
11	Inspection	**11**	Inspection	**11**	Inspection									
12	Ways to Express Indoor Comfort			**12**	BMS									
13	External Environment Conditions													
14	Economic Calculation													

Fig. 5 – The overarching modular structure of EPB standards

6 Calculation tool and Module description

The complexity of the building energy performance calculation requires also a very good coordination and testing of each calculation module to ensure coherence and the software-proof of the set of EPB standards. Therefore, each EN EPB standard shall be accompanied by a spread sheet in which the proposed calculation algorithms and data input/output are tested and proved coherent.

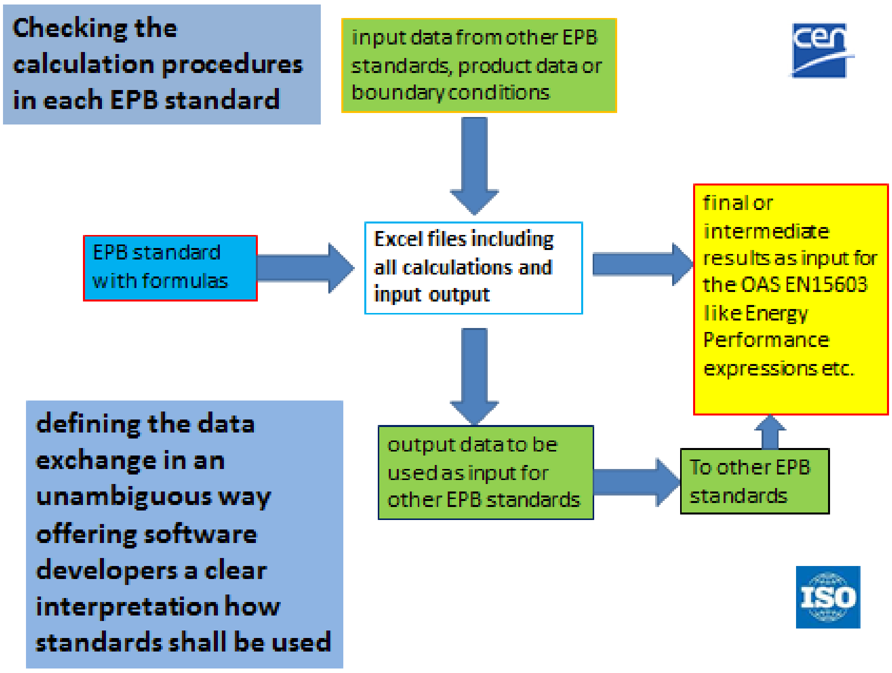


Figure 5 Software check of the excel sheets of the EPB standards

7 How the EPB standards interacts with the relevant product standards

Saving energy in the build environment requires not only that products consuming electricity and fuels are designed to be intrinsically more energy efficient. The interaction of a product with the rest of the system or installation in a building into which it is fitted plays an important role. This appears obvious for a number of product categories such as building equipment for ventilation, heating, cooling, lighting and control and automation. With the increasing application of electronic and communication technologies, this is also increasingly true for many other products, used in buildings but not considered as EPB related, that become 'smart' and 'networked', and can be controlled through wider systems. When EU-policies such as the Ecodesign Directive use a too narrow product-based view, products are considered irrespective if their surroundings and tested in standard conditions. If only their technical efficiency is considered, this approach may look straightforward but misses the savings that can be expected from ensuring that the product is also correctly sized, fitted and controlled to render its service optimally in a well-designed building installation. While it may not be difficult to reach an EU regulation of systems under product policies, it may be possible to find creative ways for tackling at least a part of the energy savings.
On one hand we have the Ecodesign Directive requiring through EU regulation minimal energy performances of energy using products. On the other side we have the EPBD where

the EU Member States are obliged to require minimal target values for the energy performance of buildings, also having specific requirements for the overall thermal performance and the energy performances of the heating, ventilation lighting and cooling systems
The CEN expert teams working on the different EPB-system standards have to check if the product data available on basis of product standards and/or related EU regulations are sufficient as input for their system standards. At the same moment the CEN and ISO product Technical Committees and/or experts have to be convinced that using the EPB system approach, to describe and test the products, is the most efficient way to ensure effective energy performance targets for products, systems and finally the buildings (see figure 6).

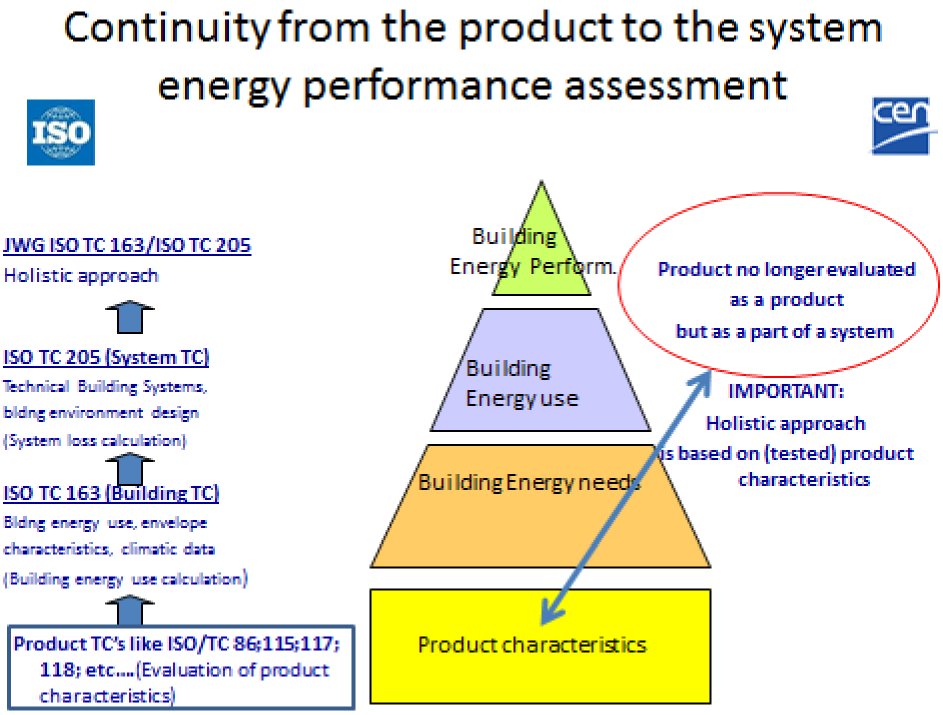


Figure 6 Products not longer evaluated as products but as part of the system.

8 Co-operation with ISO

An active process of interaction for the Overarching Type of standards through the JWG of ISO TC 163 & 205, for the other CEN-EPB standards via the different WG's of ISO TC 163 and 205. Since many years CEN and ISO share early prelim draft texts. Experts in the ISO and CEN teams are working on these standards, with the ultimate goal to agree on EN-ISO standards. A challenge given the geographic and other differences in the building sector and given the very tight time scale at CEN level. For EPB standards under some of the CEN TC's the cooperation with ISO is still informal. This means that for these standards no parallel voting is expected before 2016. However when the EN standards are accepted by the relevant ISO TC's later steps will lead to publication as EN-ISO standards.
In ISO, a series of numbers has been reserved for all EPB standards (52000----52150). Several (11 of the 42) first generation of EBP standards are already EN-ISO standards. They have been developed under the Vienna Agreement. Revision of these standards requires co-operation with the responsible ISO/TC. The central co-ordination of the preparation of a set of international standards on the energy performance of buildings at the ISO level is in the hands

of ISO /TC 163/WG 4, *Joint Working Group of ISO TC 163 and TC 205 on energy performance of buildings using a holistic approach*.
The main leading and active experts in CEN (members of the CTL of CENTC371) and ISO are among the main leading and active members of this ISO Joint Working Group.
In order to co-ordinate revisions of EN-ISO standards required under mandate M/480 and the activities within the responsible ISO/TCs, CEN/TC 371 established a liaison with ISO/TC163/WG4 (see Figure 7). This co-operation with ISO aims to avoid serious duplication of work, to avoid incompatibilities in (input) product data, procedures and (output) energy performance data.

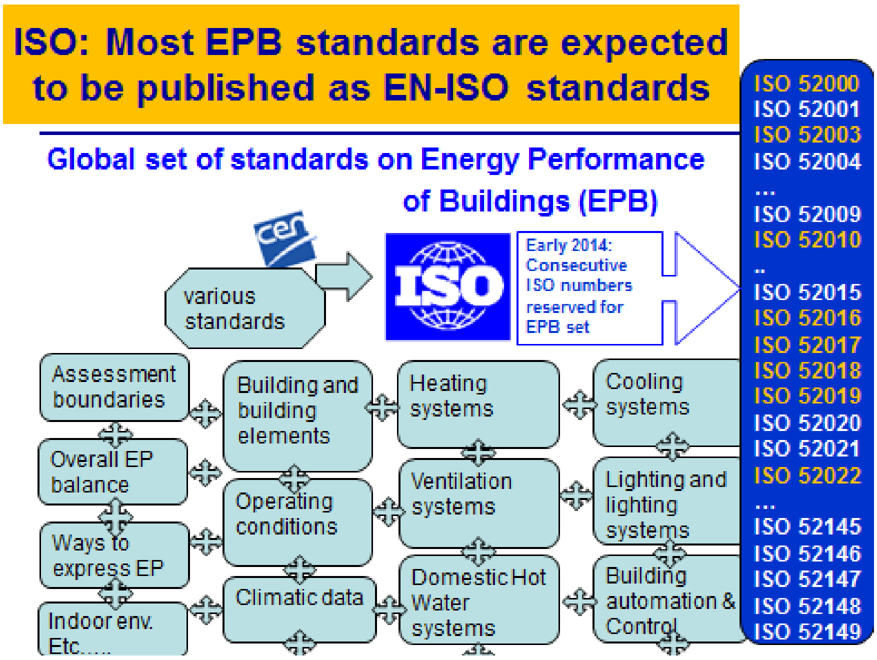


Figure 7 Schematic operational structure.

ANALYSIS OF INDOOR AIR QUALITY & THERMAL COMFORT PARAMETERS IN BUILDING REGULATIONS IN 8 MEMBER STATES

Eleni Kontonasiou[1]

1 BPIE - Buildings Performance Institute Europe
23, Rue de la Science
B – 1040 Brussels, Belgium

ABSTRACT

It is estimated that people spend 60-90% of their life in indoor environments. Therefore, it is obvious that indoor air quality (IAQ) and thermal comfort are of highly importance for the health and wellbeing of the population. Consequently, buildings should be designed to ensure proper indoor conditions. Furthermore, the need to mitigate climate change and to reduce energy import dependency, provides additional challenges for the design and operation of buildings and requires a dramatic reduction in their energy consumption and emissions. Based on that, the Energy Performance of Buildings Directive (EPBD, 2010/31/EU) asks EU Member States (MS) to significantly improve their regulatory and policy framework to ensure that minimum energy performance requirements will be met. At the same time, the EPBD, acknowledging the important role of IAQ, clearly states that minimum energy performance requirements “shall take account of general indoor climate conditions, in order to avoid possible negative effects such as inadequate ventilation”.

Projects and voluntary standards for very low energy buildings already prove that buildings can be energy efficient and at the same time contribute to outstanding IAQ and thermal comfort. But how do today’s building codes address these topics?

The overall aim of the paper is to provide an overview of the regulatory framework for IAQ and thermal comfort and to highlight the importance of having appropriate requirements for the parameters linked with these topics. The parameters that are studied in this paper are: ventilation rates, airtightness, indoor air pollutants, mechanical and natural ventilation, indoor temperatures, humidity and air velocity. The assessment focuses on the respective building codes for new and existing residential buildings in selected MS: Belgium (Brussels Region), Denmark, France, Germany, Italy, Poland, Sweden and the UK (England and Wales).

The results of the analysis show that all studied MS have at least a basic reference to IAQ included in their building codes. Minimum ventilation rates are required or recommended in all 8 MS and precise airtightness requirements are in place in 6 MS (BE, DK, FR, SE, PL, UK). Concerning thermal comfort indicators, indoor temperature requirements or recommendations range between 16°C (PL) and 28°C (FR) and recommendations with regard to humidity are given in 6 MS (DE, PL, IT, SE, UK).

For existing buildings, indoor air quality related requirements (ventilation rates, airtightness, etc.), can hardly be found in the analysed building codes. Regarding thermal comfort, even though it is often considered as a main driver for the decision of an owner/occupier to invest in renovation, it is rarely captured by national and/or European legislations.

Based on the findings of the paper, it can be concluded that indoor health and comfort aspects should be considered to a greater extent in the European and national building codes than it is current practice.

KEYWORDS

Indoor air quality, thermal comfort, residential buildings, building regulations

1 INTRODUCTION

Indoor air quality (IAQ) refers to the quality of the air inside a building. Its importance for people's health, comfort and ability to work is self-evident taking into consideration that people spend 60-90% of their life in indoor environments (homes, offices, schools, etc.) (DG SANCO, 2011). In 2012, 99000 deaths in Europe and 19000 in non-European high income countries were attributable to household (indoor) air pollution (World Health Organization, 2014).

Thermal comfort is described by the British Standard BS EN ISO 7730 as "that condition of mind which expresses satisfaction with the thermal environment" and according to the Health and Safety Executive (Health and Safety Executive) is strongly linked to environmental factors such as air temperature and humidity as well as to personal factors (clothing insulation, metabolic heat). Thermal comfort plays an important role in human health and well-being since, when building occupants feel too warm, this can cause a feeling of tiredness, while when they feel too cold they can be restless and distracted (Green Education Foundation).

Referring to IAQ and thermal comfort aspects, the Energy Performance of Buildings Directive (EPBD, 2010/31/EU) states that minimum energy performance requirements "shall take account of general indoor climate conditions, in order to avoid possible negative effects such as inadequate ventilation". Consequently, holistic planning and all-encompassing building codes are necessary in order to adequately deal with these challenges. But how do today's building codes address these topics?

The aim of this paper is to provide a comprehensive overview of the regulatory framework for IAQ and thermal comfort parameters in residential buildings in selected member states (MS) and to highlight the importance of having appropriate requirements for these parameters.

2 INDOOR AIR QUALITY AND THERMAL COMFORT PARAMETERS

2.1 Ventilation rates

In most standards and guidelines, IAQ is related to a required level of ventilation. Indicators and units to define the air exchange rate vary largely throughout Europe and are not always easy to compare (Table 1).

Table 1: Ventilation standards in dwellings (BPIE, 2015)

Country	Whole Building Ventilation Rates	Living Room	Bedroom	Kitchen	Bathroom + WC	WC only
Brussels (requirements)	3.6 $m^3/(h \cdot m^2)$ floor surface area	Minimum 75 m^3/h (limited to 150 m^3/h)	Minimum 25 m^3/h (limited to 72m^3/h)	Open kitchen Minimum 75 m^3/h (exhaust)	Minimum 50 m^3/h (limited to 75 m^3/h)	Minimu m 25 m^3/h
Denmark (requirements)	Min. 0.3 $l/(s \cdot m^2)$ (supply)	Min. 0.3 $l/(s \cdot m^2)$ (supply)		20 l/s (exhaust)	15 l/s (exhaust)	10 l/s (exhaust)
France (requirements)	10-135 m^3/h (depending on room number and ventilation system)			Continuous: 20 – 45 m^3/h		Minimu m 15 m^3/h
Germany (recommendations)	15-285 m^3/h (from 30 m^2 to 210 m^2)			45m^3/h (nominal exhaust flow)	45 m^3/h (nominal exhaust flow)	25 m^3/h (nominal exhaust flow)
Italy (recommendations)	Naturally ventilated: 0.3 – 0.6 vol/h	0.011 m^3/s per person for an occupancy level of 0.04 persons/m^2			4 vol/h	
Poland (recommendations)	20 m^3/h for each permanent occupant	20 -30 m^3/h for each permanent occupant (for public buildings)		30 m^3/h to 70 m^3/h without windows	50 m^3/h	30 m^3/h
Sweden (requirements)	Supply: min 0.35 $l/(s \cdot m^2)$ floor area					
UK (recommendations)	13-29 l/s (depending on bedrooms)			13-60 l/s (extract)	8-15 l/s (extract)	6 l/s (extract)
EN 15251	0.35 – 0.49 $l/(s \cdot m^2)$	0.6 – 1.4 $l/(s \cdot m^2)$		14 -28 l/s	10- 20 l/s	7-14 l/s

In **Brussels-Capital Region**, in case of new, modified or removed window(s) in a residential unit, air ventilation systems have to be in line with the Belgian Standard NBN D 50-001 "ventilation devices in residential buildings" (Table 1), except some adjustments that are specified in the Decree of 21 December 2007. Bruxelles Environnement, the public administration for environment and energy in the Brussels-Capital Region, recommends that windows should be designed to allow hygienic ventilation and rapid discharge of indoor pollution.

In **Denmark**, the Building Regulation 2010 (BR10) clearly addresses the importance of indoor air quality and ventilation. Apart from the specific ventilation rates (Table 1), the BR10 states that ventilation systems must be designed, built, operated and maintained so that they achieve no less than the intended performance when they are in use. Fresh air must be provided through openings directly to the external air or by ventilation installations with forced air supply.

According to Article R*111-9 of the **French** Code of Construction and Housing, air exchange rates and the expulsion of emissions have to guarantee that air quality does not constitute a danger for the occupants and certain airflow requirements must be followed (Table 1).

The **German** Energy Saving Ordinance (EnEV 2014) requires all new buildings to be built airtight according to the state-of-the-art, in a manner that ensures appropriate air exchange for a healthy and warm indoor environment. The general wording of EnEV 2014 leaves it up to the planner and architect to decide if additional mechanical ventilation is needed. The non-binding standard DIN 1946-6 "Ventilation of dwellings" provides more guidance on minimum ventilation rates (Table 1).

Italy follows a regional approach concerning building regulations: local health agencies and the municipalities are responsible for building requirements according to the Decree of the

President of the Italian Republic No. 380 of 6 June 2001. Based on the Standards listed in Legislative Decree 192/2005 (Table 1), the national implementation of the EPBD, in the case of natural ventilation 0.3 vol/h are used in the design phase. Moreover, the Standards suggest:

- An exchange rate of 4 vol/h for bathrooms
- A flow of external air of 0.011 m^3/s per person for an occupancy level of 0.04 persons/m^2 in dining rooms and bedrooms

In **Poland**, mechanical or natural ventilation must ensure appropriate air exchange for premises designed to accommodate people. According to the legislation Art 149.1, for residential premises, the ventilation rates shall not be lower than 20 m^3/h for each permanent occupant. Moreover, based on the Polish Norms PN-B-03430:1983/Az3:2000 (Table 1), the recommended minimum volumetric flow rate of ventilation air for an apartment is determined as a sum of the respective spaces flow:

- For collective rooms the ventilation rates should not be lower than 20 m^3/h per occupant
- For rooms with air conditioning and ventilation, with no possibility to open the windows, the ventilation rates should not be lower than 30 m^3/h per occupant

The **Swedish** Building Code, BFS 2014:3 - BBR 21, states that "buildings and their installations shall be designed so that air (…) quality, and light, moisture, temperature and hygienic conditions will be satisfactory during the life of the building and thus the damage to people's health can be avoided". Therefore, ventilation systems shall be designed for a minimum airflow of 0.35 $l/(s \cdot m^2)$ in the cases of both new buildings and alternations of buildings, and the outdoor airflow must not be lower than 0.10 $l/(s \cdot m^2)$ of floor area when the space is unoccupied and 0.35 $l/(s \cdot m^2)$ when the space is occupied.

In the **UK (England and Wales)**, according to the Building Regulations 2010, "there shall be adequate means of ventilation provided for people in the building". The "Approved Document F1 – Means of ventilation" provides guidance about compliance and may be accepted as reasonable provision for compliance. Among others, the approved document proposes that the following ventilation rates should be followed:

Table 2: Ventilation rates that should be followed in England and Wales

Extract ventilation rates			
Space type	**Intermittent extract rate / Minimum rate**	**Continuous extract / Minimum high rate**	**Continuous extract / Minimum low rate**
Kitchen (l/s)	30 adjacent to the hob; 60 elsewhere	13	Total extract rate should be at least the whole dwelling ventilation rate given in the following table
Utility rooms (l/s)	30	8	
Bathroom (l/s)	15	8	
Sanitary accommodation (l/s)	6	6	

Whole dwelling ventilation rates					
No of bedrooms (l/s)	1	2	3	4	5
Whole dwelling ventilation rate [a,b] (l/s)	13	17	21	25	29

a. The minimum ventilation rate should not be less than 0.3 l/s per m^2 of internal floor area. b.This is based on two occupants in the main bedroom and a single occupant in all other bedrooms. If a greater level of occupancy is expected, add 4 l/s per occupant.

2.2 Airtightness

Building airtightness, which describes the resistance of the building envelope to inward or outward air leakage, is a crucial aspect for a better energy performance of buildings. Although it is now included in many energy performances related regulations (e.g. in BE, DK, FR, DE, SW, UK) in practice there are major differences in the way it is taken into account (Figure 1).

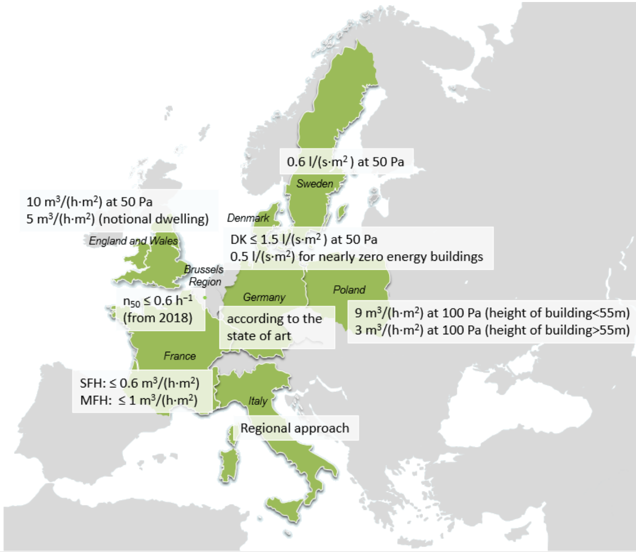


Figure 1: Airtightness requirements in Europe (BPIE, 2015)

According to the Decree of 21 December 2007, in **Brussels-Capital Region** from 2018, the individual dwelling's PEB (Performance Energétique des Bâtiments) units require an airtightness of maximum 0.6 volume per hour, while before 2018, there is no requirement on airtightness.

In **France**, for individual buildings, the airtightness has to be equal or lower than 0.6 $m^3/(h \cdot m^2)$ and for multi-family residential buildings, the airtightness has to be equal or lower than 1 $m^3/(h \cdot m^2)$. The Thermal Regulation RT 2012 requires mandatory airtightness tests for all new dwellings.

In **Germany**, leakages in exterior building elements have to be avoided according to the Energy Saving Ordinance 2014 and airtightness has to be according to the state-of-the-art (DIN 4108-7). Therefore, the air leakage rate (n_{50}) must not exceed 3 h^{-1} in houses with natural ventilation and 1.5 h^{-1} in dwellings using mechanical ventilation, whereas for the Passivhaus standard the limit is set at 0.6 h^{-1} (ETHICS Project, 2008).

Italy has no requirements on airtightness at national level, but some regions do. For example, the Province of Bolzano introduced, on 1 March 2010, mandatory blower door tests (carried out according to EN 13829) in case of energy certification of new dwellings.

In **Poland**, since 2014 there are requirements (Regulation of the Minister of Infrastructure, 12 April 2002) stating that:

- In low or moderately high buildings (up to 55 m), the air permeability of windows and doors (at the pressure of 100 Pa) shall not be higher than 2.25 $m^3/(m^2 \cdot h)$ in relation to the length of the contact line or 9 $m^3/(m^2 \cdot h)$ in relation to the surface area, which corresponds to class 3 of the Polish standard for air permeability
- In high-rise buildings (more than 55 m), the air permeability of windows and doors (at the pressure of 100 Pa) shall not be higher than 0.75 $m^3/(m^2 \cdot h)$ in relation to the length of the contact line or 3 $m^3/(m^2 \cdot h)$ in relation to the surface area, which corresponds to class 4 of the Polish standard for air permeability

In **Sweden**, according to BFS 2011, the air leakage rate through the building envelope shall not be higher than 0.6 $l/(s \cdot m^2)$ (at 50 Pa). Specifically, for single-family homes (<50m^2) if this requirement as well as an average heat transfer coefficient less than 0.33 $W/m^2 \cdot K$ are satisfied, no requirement for maximum energy use ($kWh/m^2/year$) has to be fulfilled.

In the **UK (England and Wales)**, the dwelling complies with the requirements if the measured air permeability is not worse than the limit of 10 $m^3/(h{\cdot}m^2)$ at 50 Pa. At the same time, the notional dwelling specification sets the airtightness level at 5 $m^3/(h{\cdot}m^2)$ at 50 Pa (Approved Document L1A, England & Wales), whereas, as stated in the Approved Document F1 "through good design and execution, domestic and non-domestic buildings can currently achieve an air permeability down to 2 to 4 $m^3/(h{\cdot}m^2)$ at 50 Pa".

2.3 Mechanical and natural ventilation

For a good indoor climate and air exchange in buildings, a ventilation control system is required, for which both natural and mechanical solutions exist. From the studied countries, mandatory mechanical ventilation has been identified for two cases: multi-family buildings in Denmark and high-rise residential buildings in Poland, while for all other cases, recommendations vary from rather pro mechanical ventilation (Brussels-BE and DE) or neutral position (DK, FR, SE, UK) to pro natural ventilation (IT).

In **Brussels-Capital Region**, both mechanical and natural ventilation systems can be installed. Regarding ventilation in dwellings, the Decree of 21 September 2007 refers to the NBN D50-001 standard which defines four approaches: natural ventilation (systems A), single flow controlled mechanical ventilation (CMV) provided for inlet flow (systems B) and outlet flow (systems C), as well as double flow controlled mechanical ventilation (systems D). Furthermore, there is a distinction between continuous ventilation and intermittent/periodic ventilation, which is needed in case of overheating or pollutant activities and requires the presence of openings in kitchens, dining rooms and bedrooms. Moreover, in open kitchens, systems A are not allowed unless a hood with ventilation is installed. The guidelines written by Bruxelles Environnement (Bruxelles Environnement (a), 2014) suggest introducing passive systems for cooling. In order to provide a good quality for indoor air, it is also suggested to introduce a single flow CMV over natural ventilation, although double flow CMV is suggested for an optimum level of indoor air (Bruxelles Environnement (b) , 2014).

In **Denmark**, single-family houses may use natural or mechanical ventilation. It is assumed that people in one-family houses open the windows, have ventilation openings etc., so a good indoor climate can be obtained even without a mechanical system. On the other hand, apartments in multi-storey buildings must be mechanically ventilated.

In **France**, neither regulations for ventilation nor RT 2012 impose a mechanical ventilation system for residential buildings. Generally, natural ventilation, mechanical ventilation and hybrid ventilation are allowed. Specifically, for overseas departments (Guadeloupe, Guyana, etc.) natural ventilation has to be prioritised for dwellings and new parts of dwellings, as mentioned in Article R162-4 of the Code of Construction and Housing.

In **Germany**, as the reference building is a non-binding description, mechanical ventilation systems are not obligatory, but indirectly recommended by the government (EnEV 2009 Anlage 1 Tabelle 1 Zeile 8).

In **Italy**, the Decree of the President of the Republic 59/2009 leaves much freedom for the planner regarding the ventilation design. However, it recommends natural ventilation and - if not sufficient - effective mechanical ventilation to be considered for new buildings and deep refurbishment. Nevertheless, mechanical ventilation is mandatory for new construction and deep renovations in at least 105 regional building regulations all over Italy (ONRE, 2013).

The **Polish** Regulation of the Minister of Infrastructure (April 12, 2002) states that in every occupied room, there is a requirement to use appropriate mechanical or natural ventilation.

The mechanical, exhaust and supply/exhaust ventilation is obligatory in high-rise buildings (>25m, >9 storeys) and in buildings where the adequate quality of the indoor environment is not possible by means of natural ventilation. In all the other buildings, the use of natural and hybrid ventilation is permitted.

In **Sweden**, the required airflow of 0.35 $l/(s \cdot m^2)$ can be ensured via mechanical or natural ventilation. Boverket, the National Board of Housing, Building and Planning, published the handbook “Självdrags-ventilation”, which according to the Swedish Building Code (BFS 2014:3 - BBR 21) can be used for guidance by developers, planners and building committees. The demands are functional, so authorities do not mind hybrid ventilation as long as an airflow of 0.35 $l/(s \cdot m^2)$ can be ensured.

In the **UK (England and Wales)**, the ventilation strategy adopted in Approved Document F suggests natural ventilation, mechanical ventilation system or a combination of both. For mainly naturally ventilated buildings, it is common to use a combination of ventilators. For example, in dwellings it is common to use intermittent extractor fans for extraction ventilation, trickle ventilators for whole dwelling ventilation and windows for purge ventilation. For mechanically ventilated or air conditioned buildings, it is common for the same ventilators to provide both local extraction and whole building/ dwelling ventilation.

2.4 Air velocity

Air velocity is an important factor in thermal comfort because people are sensitive to it. Small air movement in cool environments may be perceived as draught. If the air temperature is less than the skin temperature, it will significantly increase the convective heat loss. Very low levels of air movement can also cause a feeling of discomfort and stuffiness in a room. The following map gives an overview of legal limitations of air velocity throughout Europe.

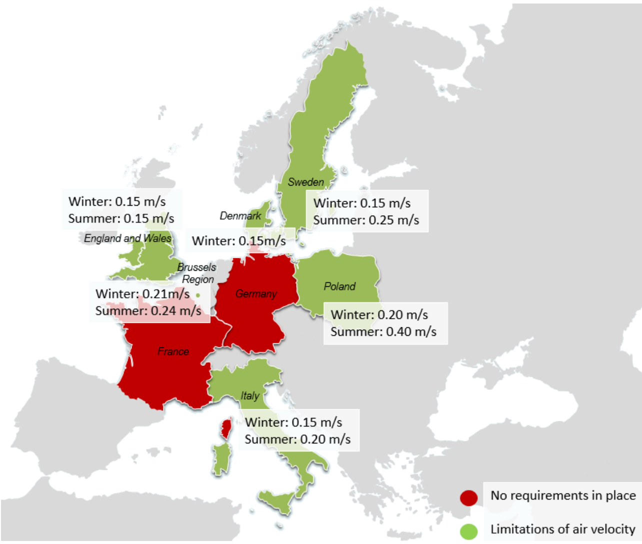


Figure 2: Maximal air velocity in Europe (BPIE, 2015)

According to the recommendations from **Brussels** Environment (Bruxelles Environnement (a), 2014), the air velocity in summer should not be higher than 0.24 m/s, whereas for optimum thermal comfort, air velocity should be limited to 0.12 m/s. In winter conditions, the recommended air velocity limit is set at 0.21 m/s and, for an optimum thermal comfort, it should be no higher than 0.10 m/s.

In **Denmark**, the air velocity should not exceed 0.15 m/s to avoid draughts, except during summertime with temperatures above 24°C.

No requirements regarding air velocity are identified so far in **France and Germany**.

The **Italian** Standard (UNI 10339:1995) foresees that for residential buildings, the air velocity should be between 0.05 and 0.15 m/s during the winter period and 0.05 and 0.20 m/s during the summer period.

The **Polish** Standard PN-B-03421:1978 specifies the comfort indoor parameters (Table 3) taking into account the physical activity of occupants.

Table 3: Indoor comfort parameters in Poland

	Winter	**Summer**
Low metabolic rate		
Indoor temperature	20-22°C	23-26°C
Airflow	0.2 m/s	0.3 m/s
Average rate of metabolism		
Indoor temperature	18-22°C	20-23°C
Airflow	0.2 m/s	0.4 m/s
High rate of metabolism		
Indoor temperature	15-18°C	18-21°C
Airflow	0.3 m/s	0.6 m/s

According to the **Swedish** Buildings Code (BFS 2014:3 – BBR 21), the air velocity in a room is not expected to exceed 0.15 m/s during the heating season and air velocity from the ventilation system shall not exceed 0.25 m/s during other times of the year.

In the **UK (England and Wales)**, the air velocity in a room is not expected to exceed 0.15 m/s during the heating season and air velocity from the ventilation system shall not exceed 0.25 m/s during other times of the year (Brelih & Seppänen, 2011).

2.5 Humidity

Humidity is of particular concern in residential ventilation as most of the adverse health effects and building disorder (condensation, moulds) are related to humidity. From the studied countries recommendations concerning humidity (in order to avoid water condensation or an air too dry) are given in Brussels-Capital Region, Denmark, Germany, Poland, Sweden and the UK (soft reference).

In **Brussels – Capital region**, elementary requirements, defined at Decree of 4 September 2003, related to the healthiness of IAQ have to be fulfilled, for example the limitation of humidity causing mould or damage to the walls. **France** and **Italy** have not adopted specific requirements on humidity levels. In **Denmark**, non-binding recommendations for relative humidity are specified in DS 474, Code for Indoor Thermal Climate. In **Germany**, according to the non-binding DIN EN 13779, the minimum and maximum indoor relative humidity ranges from 30 to 70%. In **Poland**, there are specific requirements to avoid water condensation (Regulation of the Minister of Infrastructure, 12 April 2002) such as the fact that the temperatures in the AC system should be adjusted so there is no condensation on the surfaces. Apart from that, the Polish Standard PN-B-03421:1978 specifies the comfort indoor parameters taking into account the physical activity of the occupants (Table 4).

Table 4: Comfort parameters related to humidity in Poland

Metabolic rate	**Winter**			**Summer**		
	Relative humidity		Max airflow	Relative humidity		Max airflow
	Optimal	Minimal		Optimal	Maximal	
Low	40-60%	30%	0.2 m/s	40-55%	70%	0.3 m/s
Average	40-60%	30%	0.2 m/s	40-60%	70%	0.4 m/s
High	40-60%	30%	0.2 m/s	40-60%	70%	0.6 m/s

Air humidity and moisture safety are major concerns of the **Swedish** Building Code (BFS 2014:3). Maximum moisture conditions are defined as an upper limit for which such negative effects do not happen and in winter, the difference in absolute humidity between indoor and outdoor should not be higher than 3 g/m³. In the **UK**, according to the Approved Document F, the moving average relative humidity in a room during the heating season should be less than the values defined in the following table.

Table 5: Indoor air relative humidity

Moving average period	Room air relative humidity
1 month	65%
1 week	75%
1 day	85%

2.6 Minimal and maximal temperature requirements

Indoor air temperature is an indicator for thermal comfort in all surveyed countries and there are requirements and recommendations in place for lower and upper limit during winter and summer respectively (Figure 3).

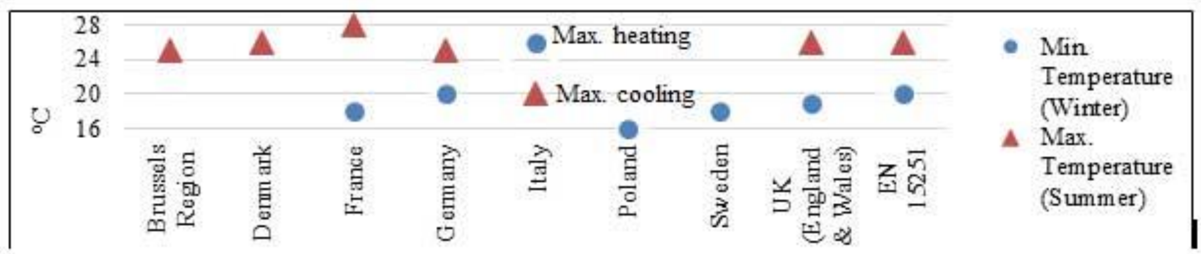


Figure 3: Temperature requirements in Europe (BPIE, 2015)

In **Brussels-Capital Region**, overheating is defined as temperature of more than 25°C and has to be limited to 5% of the time during the year (Decree of 21 December 2007). In **Denmark**, according to the DS 474 Code for Indoor Thermal Climate, appropriate indoor temperatures have to not exceed 26°C for more than 100 hours and 27°C for more than 25 hours. In **France**, heating equipment in all housing must maintain a minimum of 18°C at the centre of the housing parts, while the maximum comfort temperature is 28°C when a mechanical system is used. In **Germany** the recommended indoor air temperature is limited at 25°C, 26°C and 27°C for climatic regions A, B and C respectively and according to DIN 4701-10, the temperature in apartments should be able to reach at least 20°C and 22°C in bathrooms. In **Italy**, minimal and maximal temperatures are required in order to limit the waste of energy for cooling and heating. Therefore, according to Article 3 of the Decree of the President of the Republic 74/2013, cooling systems have to be limited to 26°C and heating systems to 20°C. In **Poland**, the indoor temperature cannot be lower than 16°C and buildings should be designed and constructed in such a way that they reduce the risk of overheating in the summer (Regulation of the Minister of Infrastructure, 12 April 2002). According to the **Swedish** Building Code (BFS 2014-3 – BBR-21), buildings and their installations must be designed to guarantee a satisfactory thermal comfort. Based on that, the recommended minimal operative temperature for the average dwellings is 18°C. In the **UK**, heating systems should be designed to be able to maintain a temperature of 18°C in sleeping rooms and 21°C in living rooms when the temperature outside is at the local design temperature, commonly -1°C. Moreover, according to the Chartered Institution of Building Services Engineers, the operative (maximum) temperature should not exceed 25°C in living areas and 23°C in bedrooms.

2.7 Indoor air pollutants

Beside CO_2 concentration and humidity, there are no other generally accepted criteria and measuring methods for pollutants in EN 15251. Only if specific complaints (e.g. smell, sick building symptoms, etc.) persist and ventilation measurements show that the requirements for fresh air supply are met, should measurements of specific pollutants (e.g. formaldehyde, other Volatile Organic Compounds, fine dust (PM 10 or PM 2.5)) be made. The CO_2 concentration in fully occupied buildings – where inhabitants are the main pollutants – in relation to outdoor concentration is indicated by the European standard EN 15251. Requirements for limiting CO_2 levels in residential buildings are in place in France (<1000ppm), while in the UK there are recommended levels (800 - 1000 ppm). Limitations for nitrogen oxide are also in place such as is the case in Denmark. National implementation of European's construction products directives and further national standards address evaporation of unhealthy chemicals, however, this legislation is not considered for the purpose of this analysis.

2.8 Indoor air quality and thermal comfort requirements in existing buildings

For existing buildings, indoor air quality related requirements, such as minimum ventilation rates, airtightness or limitation of pollutants, can hardly be found in the analysed building codes (BPIE, 2015). Energy efficiency improvements do often apply without mandatory consideration of the influences in terms of building physics or indoor air quality. Among the surveyed countries, the Swedish building codes are unique at the moment by underlining potential conflicts between energy saving requirements and good indoor air quality in existing buildings, stipulating that in such cases priority should be given to the latter. Regarding thermal comfort, even though it is often considered as a main driver for the decision of an owner-occupier to invest in renovation, thermal comfort parameters are rarely captured by national and/or European legislations.

3. CONCLUSIONS

Indoor air quality is recognised as an important aspect in the building codes in all focus countries of this survey. Furthermore, thermal comfort aspects are often captured by national legislation. However, as identified in the eight focus countries of this paper, there are no clear and strict requirements to cover these topics. Therefore, indoor air quality and thermal comfort aspects have to be seriously considered when strengthening the energy performance requirements for buildings and building elements. Towards this goal:

- In the EU and national legislation, stricter energy performance requirements should be complemented with appropriate requirements and recommendations to secure proper indoor air quality and thermal comfort. For instance, requirements for stricter insulation and airtightness should be completed by appropriate minimum requirements for indoor air exchange and ventilation.
- Indoor air quality indicators should be integrated in Energy Performance Certificates as relevant information regarding the actual living conditions in the building.
- The development of a proper cost indicator and a calculation formula to estimate the benefits of a healthy indoor environment should be considered and further integrated in the European methodology to calculate cost-optimal levels at macroeconomic level.
- The co-benefits of a healthy indoor environment should be taken into account when assessing the macroeconomic impact of energy renovation measures (e.g. reduction of health service costs).

4. ACKNOWLEDGEMENTS

This paper is based on BPIE's study "Indoor air quality, thermal comfort and daylight" (2015) and is the product of the collaboration between BPIE's team (S. Kunkel, E. Kontonasiou, A. Arcipowska, F. Mariottini and B. Atanasiu) and key national experts from countries within the scope. BPIE is grateful to all for their contribution which made this report possible.

5. REFERENCES

BPIE. (2015). *Indoor Air Quality, Thermal Comfort and Daylight.* Buildings Performance Institute Europe.

Brelih, N., & Seppänen, O. (2011). *Ventilation rates and IAQ in European Standards and national regulation.*

Bruxelles Environnement (a). (2014). *"Satisfaire les besoins des occupants, tout en minimisant les consommations énergétiques".*

Bruxelles Environnement (b) . (2014). *Assurer la qualité de l'air par son renouvellement et le contrôle éventuel de ses caractéristiques.*

DG SANCO. (2011). *Promoting actions for healthy indoor air (IAIAQ).*

ETHICS Project. (2008). *Report on national and European regulations and requirements regarding airtightness.*

Green Education Foundation. (n.d.). *Thermal Comfort.* Retrieved from http://www.greeneducationfoundation.org/green-building-program-sub/learn-about-green-building/1239-thermal-comfort.html

Health and Safety Executive. (n.d.). *Thermal Comfort, The six basic factors.* Retrieved from http://www.hse.gov.uk/temperature/thermal/factors.htm

ONRE. (2013). *Rapporto ONRE 2013, L'innovazione energetica in edilizia.*

World Health Organization. (2014). *Burden of disease from Household Air Pollution for 2012.*

INDOOR CARBON DIOXIDE CONCENTRATIONS IN VENTILATION AND INDOOR AIR QUALITY STANDARDS

Andrew Persily

National Institute of Standards and Technology
100 Bureau Drive, MS8600
Gaithersburg, Maryland USA
andyp@nist.gov

ABSTRACT

Indoor carbon dioxide (CO_2) concentrations have played a role in discussions of ventilation and indoor air quality (IAQ) since the 18th century. Those discussions have evolved over the years to focus on the impacts of CO_2 concentrations on building occupants, how these concentrations relate to occupant perception of bioeffluents, the use of indoor CO_2 concentrations to estimate ventilation rates, and CO_2–based demand control ventilation. This paper reviews how indoor CO_2 has been dealt with in ventilation and IAQ standards in the context of these issues.

While measured indoor CO_2 concentrations are rarely close to health guidelines, much confusion has resulted regarding CO_2 in ventilation and IAQ standards. For example, an indoor CO_2 concentration of 1800 mg/m^3 (roughly equivalent to 1000 ppm$_v$) has become a de facto standard in many discussions without a sound understanding of its basis or significance. And while there have been anecdotal associations of CO_2 concentrations in this range with occupant symptoms such as stuffiness and discomfort, research results do not support these associations with CO_2 itself. Several studies have shown associations of elevated CO_2 levels with occupant symptoms, but these findings are likely due to lower ventilation rates elevating the concentrations of other more important contaminants along with the CO_2.

The relevance of CO_2 concentrations to ventilation and IAQ standards is based primarily on two factors: their relation to indoor levels of bioeffluents and associated odors, and their relation to ventilation rates per person. Several studies of bioeffluent odor perception in chambers and buildings have shown correlations between dissatisfaction with these odors and both ventilation rate per person and CO_2 level. Also, ventilation rates and indoor CO_2 levels are related based on a single-zone mass balance of CO_2. However, many individuals use CO_2 concentrations to estimate building ventilation rates without understanding the associated mass balance theory and the assumptions on which it is based. This paper reviews these concepts and discusses the role of indoor CO_2 in various ventilation and IAQ standards.

KEYWORDS

Carbon dioxide; contaminant limits; demand control ventilation; indoor air quality; standards; ventilation.

1 INTRODUCTION

Indoor CO_2 concentrations have been prominent in discussions of ventilation and IAQ since the 18th century when Lavoisier suggested that CO_2 build-up rather than oxygen depletion was responsible for "bad air" indoors (Klauss, 1970). About one hundred years later, von Pettenkofer suggested that biological contaminants from human occupants were causing indoor air problems, not CO_2. Discussions of CO_2 in relation to IAQ and ventilation evolved since that time, focusing on the following issues: impacts of CO_2 concentrations on building occupants, how these concentrations relate to occupant perception of bioeffluents, the use of indoor CO_2 concentrations to estimate ventilation rates, and the use of CO_2 to control outdoor air ventilation rates. This paper reviews how these issues have been dealt with in ventilation and IAQ standards. The topic is introduced by discussing CO_2 levels in various standards and guidance documents, followed by identification of the primary issues of interest related to indoor CO_2. Those issues are discussed in more detail in subsequent sections.

1.1 Carbon dioxide concentrations in standards and guidelines

While indoor CO_2 concentrations are rarely close to limits in health-based guidelines, much confusion has resulted regarding CO_2 levels in ventilation and IAQ standards. A 2001 review of ventilation and IAQ criteria in several countries reported that most CO_2 limits are in the range of 1800 mg/m^3, although some are as high as 9000 mg/m^3 or even 30 000 mg/m^3 for short-term exposures (Limb, 2001). ASHRAE Standard 62.1-2013 contains a more recent listing of CO_2 limits in other documents, four of which are equal to 9000 mg/m^3 with one at 6300 mg/m^3 (ASHRAE, 2013a). Short-term exposure limits tend to be higher, ranging from 18 000 mg/m^3 to 54 000 mg/m^3. These concentration limits are typically drawn from values developed for industrial environments, which are not applicable to general populations as discussed below. The basis for limits on the order of 1800 mg/m^3 is also described later.

ASHRAE Standard 62-1981 contained an indoor CO_2 limit of 4500 mg/m^3 for use when applying the performance approach to complying with the standard, i.e., the IAQ Procedure. That limit was changed without explanation to 1800 mg/m^3 in 1989. CEN standard 13779 does not contain an indoor CO_2 limit, but has an informative annex that provides default CO_2 concentrations for its four classes of IAQ (CEN, 2007). The highest IAQ class is associated with concentrations about 700 mg/m^3 above outdoors, with the lowest class 1800 mg/m^3 above outdoors.

Other standards and guidance documents also address indoor CO_2 concentrations. ASTM Standard D6245 does not contain limits on indoor CO_2 concentrations (ASTM, 2012), but notes that adverse health effects from elevated CO_2 have not been observed until concentrations are in the range of 12 600 mg/m^3 to 36 000 mg/m^3 based on exposures of at least 30 days (ECA, 1992; EPA, 1991). LEED v4 includes requirements for naturally ventilated spaces in which one of three different options for monitoring system performance must be employed (USGBC, 2014). One of those options is to monitor CO_2 concentrations in each thermal zone and to issue an audible or visual alert if the concentration exceeds the setpoint by more than 10 %. LEED v4 refers to Standard 62.1 for the determination of these setpoints, but no values are provided in either document. LEED also awards extra points for monitoring CO_2 concentrations in densely occupied spaces, but again no concentration limits are provided. Other building design standards and guidelines allow for the use of CO_2 in demand control ventilation systems, as described below, but do not necessarily provide indoor concentration limits or setpoints.

Based on consideration of existing standards and guidelines, indoor CO_2 limits are either in the range of 1800 mg/m^3 or they are closer to industrial limits on the order of 20 000 mg/m^3 or more. Note that measured indoor values tend to be well below these industrial limits. For example, the U.S. EPA BASE study of 100 randomly selected office buildings had a mean peak indoor CO_2 value of 1260 mg/m^3, a maximum of about 2200 mg/m^3 and a 90th percentile value of 1650 mg/m^3 (EPA, 2006). In a study of 56 office buildings in Europe, the average CO_2 concentration ranged from about 900 mg/m^3 to about 1400 mg/m^3, depending on the country where the buildings were located, with no readings above 2200 mg/m^3 (Bluyssen et al. 1996). A more recent study of 37 small and medium sized commercial buildings in California resulted in a mean of the maximum CO_2 concentration of 3031 mg/m^3, though the overall mean concentration was only 1168 mg/m^3 (Bennett et al. 2011). On the residential side, a study of 108 new homes in California yielded a median indoor CO_2 concentration of 1015 mg/m^3 with a range of about 600 mg/m^3 to 2000 mg/m^3. Higher concentrations have been measured in other residences, including a recent study of 266 dwellings in Tianjin, China, with a median value for the mean CO_2 concentration of about 2500 mg/m^3 (Hou et al., 2015). In that study, concentrations in 41 % of children's bedrooms were below 1800 mg/m^3, 20 % exceeded 3600 mg/m^3, and 6 % exceeded 5400 mg/m^3. In another study of residences in Denmark, about one-quarter of the measurements in the bedrooms were above 3600 mg/m^3 and 6 % were above 5400 mg/m^3 (Bekö et al., 2010). Two studies in school classrooms showed concentrations as high as 7200 mg/m^3, though most were below about 3500 mg/m^3 (Coley and Beisteiner, 2002; Bakó-Biró, 2012). Based on this wide range of studies in different countries and different building types, most measured indoor CO_2 concentration are less than 2000 mg/m^3, with a small fraction above 5000 mg/m^3, but none in the range of health-based industrial exposure limits.

1.2 Questions and issues related to carbon dioxide

In considering indoor CO_2 in the context of ventilation and IAQ standards, a number of issues have been raised and some level of confusion exists. This section outlines those issues as follows, providing context for the discussion what follows:

- Impacts of CO_2 on building occupants
- Relationship of indoor CO_2 and perception of odors from human bioeffluents
- Relationship of indoor CO_2 levels to ventilation rates
- Application of indoor CO_2 levels to controlling outdoor air ventilation

While indoor CO_2 levels are generally far below industrial limits or levels otherwise expected to impact the health and comfort of building occupants, some discussions of indoor CO_2 have referred to the health impacts of exposure to CO_2. It is important for practitioners and other involved with building IAQ to understand how indoor CO_2 levels are associated with occupant health and comfort, and the reasons for any such associations. The second issue listed above concerns the relationship between indoor CO_2 concentrations and the perception of human body odors. This relationship has been studied since the 1930s and has provided the basis for minimum ventilation rates in most ventilation standards and regulations. Those ventilation rates also impact the third issue, the relationship between indoor CO_2 levels and building ventilation rates. In this relationship, CO_2 is simply a tracer gas with a very convenient injection mechanism, i.e., the building occupants. The theory behind tracer gas measurement techniques is well-established, and the assumptions for applying these techniques also apply when using CO_2. Finally, indoor CO_2 levels have been used for many years to control outdoor air ventilation rates based on the actual number of occupants rather than the design occupancy, commonly referred to as demand control ventilation (DCV). This approach has potential benefits for both IAQ and energy efficiency, and the use of DCV in standards is discussed below.

ASTM Standard D6245-12, Standard Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation, which was first issued in 1998, discusses all of these issues (ASTM, 2012). However, per the ASTM definition of a Guide, this document serves as a "… compendium of information or series of options that does not recommend a specific course of action" in order to increase "… awareness of information and approaches in a given subject area" (ASTM, 2014). This is the only standard specific to the interpretation of indoor CO_2 concentrations and addresses all of the issues outlined above, plus some others including the measurement of indoor CO_2 concentrations. However, being a guide, this standard does not contain detailed calculation approaches or specific instructions.

2 IMPACTS OF CO_2 ON BUILDING OCCUPANTS

While indoor CO_2 concentrations in non-industrial buildings are rarely close to any health-based guidelines for industrial environments, much confusion has resulted regarding the health impacts of exposure to CO_2 in buildings. As noted earlier, concentration limits in standards and guidelines for industrial environments are not typically relevant to commercial, institutional and residential buildings. As noted in the ASHRAE Indoor Air Quality Guide, there are large differences between industrial and non-industrial environments in terms of the composition and concentrations of airborne contaminants (ASHRAE, 2009). For example, industrial workers are typically exposed to high concentrations of specific contaminants, while occupants of non-industrial buildings are exposed to a varied mixture of contaminants for long periods of time. Also, the occupants of nonindustrial environments cover a broad range of age, sex, physical condition and pre-existing health conditions. As stated in the IAQ Guide, "For these reasons the use of industrial guideline values, or fractions of these values, is generally considered inappropriate." ASHRAE Standard 62.1-2013 also discusses the application of industrial guideline concentrations, noting that they are intended to limit exposure in order to not interfere with work processes, but not to eliminate effects such as odors or mild irritation (ASHRAE, 2013a). In addition, the standard notes that healthy industrial workers will change jobs if the exposure is intolerable, while occupants of non-industrial building occupants have less choice about where they spend their time and include individuals "…who may be more sensitive, such as children, asthmatics, allergic individuals, and the elderly."

While occupational CO_2 exposure limits are not relevant to non-industrial building environments and indoor concentrations almost never reach those levels, an indoor CO_2 concentration of 1800 mg/m^3 has became a de facto standard in many applications without a sound understanding of its basis (Persily, 1997). This reference notes the existence of anecdotal discussions associating CO_2 concentrations in this range with occupant symptoms such as stuffiness and discomfort, but notes that peer-reviewed studies do not support these associations with CO_2 itself. While several studies have shown associations of elevated CO_2 levels with symptoms, absenteeism and other effects (Apte et al., 2000; Shendell et al., 2004; Gaihre et al., 2014), these associations are likely due to lower ventilation rates elevating the concentrations of other contaminants with health and comfort impacts at the same time they are elevating CO_2.

The 1800 mg/m^3 CO_2 "guideline value" is commonly attributed to ASHRAE Standard 62. As noted earlier, the 1981 version of that standard contained an indoor CO_2 limit of 4500 mg/m^3 for use when applying the performance approach to complying with the standard, i.e., the IAQ Procedure. That limit was changed without explanation to 1800 mg/m^3 in 1989, and that value was removed from the standard in 1999. That and subsequent versions of the standard contained an informative appendix explaining that if a space is ventilated at a nominal rate of 7.5 L/s per person, then at steady-state the CO_2 concentration will equal 1800 mg/m^3 for assumed values of the CO_2 generation rate per person and the outdoor CO_2 concentration. There is nothing in the standard declaring 1800 mg/m^3 or any other CO_2 concentration to be a health or comfort based limit. The 1800 mg/m^3 "limit" simply is based on its association with a nominal ventilation requirement of 7.5 L/s per person for control of body odor perception as discussed below. Also, Standard 62 and other ventilation standards contain a range of ventilation requirements for different types of spaces, which will be associated with CO_2 concentrations other than 1800 mg/m^3.

While indoor CO_2 concentrations are typically well below values of interest based on health concerns, a recent study has shown evidence of impacts on human performance. A chamber study of individuals completing computer-based tests showed statistically significant decreases in decision-making performance at CO_2 concentrations as low as 1800 mg/m^3 (Satish et al., 2012). These experiments were carefully designed to expose the subjects to elevated CO_2 but not to other contaminants. This work has not yet impacted ventilation and IAQ standards, but if the findings are repeated in other studies, it may support future changes.

3 CO_2 AND THE PERCEPTION OF BODY ODOR

There have been many decades of research into outdoor air requirements, which, starting in the second half of the nineteenth century, focused on the control of human body odor associated with the byproducts of human metabolism, often referred to as bioeffluents (Klauss et al. 1970). Building on the seminal work of von Pettenkofer and others, Yaglou et al. (1936) used environmental chambers to investigate ventilation rates required to control the odor from bioeffluents. In these studies, human test subjects rated odor intensity as a function of the ventilation airflow per person. This research found that about 7.5 L/s to 9 L/s per person of ventilation air was needed to dilute body odor to levels judged to be acceptable by individuals entering the room from relatively clean air. Since the time of Yaglou's research, a number of researchers have reported similar results in both laboratory chambers and actual buildings (Cain et al., 1983; Fanger and Berg-Munch, 1983; Iwashita et al., 1990).

Some of these experiments also studied the relationship between CO_2 concentrations and body odor acceptability. The finding that about 7 L/s per person of ventilation controlled human body odor such that about 80 % of unadapted individuals found the odor to be acceptable was accompanied by the result that the same level of acceptability occurred at CO_2 concentrations about 1250 mg/m^3 above outdoors. For an outdoor CO_2 level of 630 mg/m^3, this indoor concentration roughly corresponds to the commonly cited value of 1800 mg/m^3. The relationship between the percentage of subjects dissatisfied with body odor and CO_2 concentrations has been seen experimentally determined in several studies, with the results largely independent of the level of physical activity (Berg-Munch et al., 1986; Fanger and Berg-Munch 1983; Rasmussen et al. 1985). In addition, the relationship did not require that the indoor CO_2 concentration be at steady-state. The lack of a strong dependency on physical activity arises from the fact that humans produce CO_2 and bioeffluents at rates that are roughly proportional to one another. The fact that these relationships do not require the existence of steady-state conditions arises because both CO_2 and odor levels increase at a rate that is primarily dependent on the air change rate of the space in question.

This research supports 1800 mg/m^3 of CO_2 as a reflection of body odor acceptability perceived by unadapted visitors to a building. Of course, there are many other contaminants in indoor air that are not associated with the number of occupants, and CO_2 concentration is not a good indicator for those contaminants.

4 CO_2 AND VENTILATION RATES

As discussed previously (Persily, 1997; ASTM, 2012), per person ventilation rates and indoor CO_2 levels are related based on a single-zone mass balance of CO_2. This relationship has been discussed in Standard 62 since 1981, in which the steady-state equation is presented as follows: the outdoor air ventilation rate per person (Q) equals the ratio of the CO_2 generation rate per person (G) to the difference between the indoor and outdoor CO_2 concentrations (ΔC), or $Q = G/\Delta C$. For a ventilation rate of 7.5 L/s per person and a CO_2 generation rate of 0.3 L/min per person, the indoor CO_2 concentration will be about 1200 mg/m^3 above outdoors. Using slightly different values of the generation rate, one arrives at the indoor CO_2 concentration value of 1800 mg/m^3. This value is often referred to as a CO_2 "limit" erroneously attributed to ventilation standards, but as discussed in earlier, is actually related to recommended ventilation rates for body odor control under idealized, steady state conditions, not to the health or comfort impacts of the CO_2.

This relationship between CO_2 concentrations and ventilation rates is essentially an application of the constant injection tracer gas method, which has been well-understood for decades (Hunt, 1980), and is often used to estimate ventilation rates without an adequate understanding of its basis. The constant injection method, described in ASTM E741 (ASTM, 2011), involves injecting tracer gas at a constant rate into the space being tested and monitoring the concentration response. Since the so-called peak CO_2 approach is a single-zone steady-state tracer technique, it must abide by the following assumptions to yield a valid air change rate: the CO_2 generation rate is known, constant, and uniform throughout the building being tested; the CO_2 concentration is uniform throughout the building and has achieved steady state; the outdoor CO_2 concentration is known and constant; and, the outdoor air ventilation rate is constant. Given that the method is based on a single-zone mass balance, it can only be used to determine the air change rate of an entire building with a uniform concentration. If the CO_2 concentration varies among rooms, the single-zone approach is not valid and one must employ a multizone mass balance of CO_2 that accounts for the airflows between zones and is far more complicated to apply. Similarly, the steady-state CO_2 value in a single room cannot be used to calculate the ventilation rate of that room if adjoining spaces are at different CO_2 concentrations.

In order for the CO_2 generation rate to be known, constant, and uniform throughout the building, the occupancy level must be known, constant, and uniform. Assuming it is, the CO_2 generation rate can be estimated based on the number and characteristics of the building occupants, that is, their age, size and level of physical activity (ASTM, 2012). However, as noted in that standard, CO_2 generation rates can vary significantly among individuals. Also, the requirement for the CO_2 concentration being at steady state translates to conditions being constant for long enough that a steady-state concentration is achieved. As described in Persily (1997) and ASTM (2012), the time required to achieve steady state depends on the air change rate of the building. For a given air change rate, the concentration will be within 95 % of steady state after three time constants, where the time constant is the inverse of the air change rate. For an air change rate of 1 h^{-1}, it will take 3 h to reach 95 % of the steady-state. For an air change rate of 0.5 h^{-1}, it will take 6 h. During this time, the ventilation rate, occupancy, and outdoor concentration must all be constant. Using a CO_2 concentration before steady state has been achieved will overestimate the air change rate and the associated airflow rate per person, in some cases by significant amounts.

Indoor CO_2 concentrations are clearly related to ventilation rates, but that relationship is complicated by the multizone, transient nature of building airflow systems as well as variations in building occupancy (i.e., CO_2 generation) both temporally and spatially. The relationship is simple under only very specific circumstances, making CO_2 a questionable indicator of ventilation rates.

5 DEMAND CONTROL VENTILATION USING CO_2

Ventilation and IAQ standards allow the use of DCV to control outdoor air ventilation rates; in fact it is required under some circumstances in energy efficiency standards (ASHRAE, 2013b). Under ASHRAE Standard 62.1-2013, the ventilation requirement of a space is the sum of a per-person rate multiplied by the number of occupants plus an area rate multiplied by the floor area of the space. The use of DCV is relevant only to control of the ventilation requirement based on the number of occupants, i.e., the rate based on the floor area must always be maintained. In practice, The standard allows the use of CO_2 as means to implement DCV, with the outdoor air intake rate controlled using the indoor CO_2 level such that the air intake rate is increased as the CO_2 levels rises and is decreased as it drops, similar to temperature control using a thermostat The setpoint depends on the ventilation rate requirement of the space of interest, which is a function of the space type, the number of occupants and the floor area. Given the requirements in Standard 62.1, setpoints will tend to be higher in densely occupied spaces. However, Standard 62.1 does not contain requirements specific to the application of CO_2 DCV; it simply allows DCV to be used to dynamically reset outdoor air intake flows and mentions CO_2 as an acceptable means of doing so. The application of CO_2 DCV per the standard is described in detail in the Users Manual for the standard (ASHRAE, 2010). Additional information on the application of CO_2–based DCV, specifically setpoint values for different space types, is available in Lawrence (2008).

More specific requirements for the use of CO_2 DCV are contained in California Title 24 and ANSI/ASHRAE/IES/USGBC Standard 189.1 (CEC, 2013; ASHRAE, 2014). Title 24 contains requirements regarding the number of CO_2 sensors, their location and their accuracy. It also specifies that the setpoint be 1080 mg/m^3 above the outdoor concentration, which can be measured or assumed to equal 720 mg/m^3. Standard 189.1 also contains requirements for sensor location and accuracy, but requires that the outdoor concentration be measured.

6 CONCLUSIONS

This paper summarizes how CO_2 is dealt with in ventilation and IAQ standards, focusing on the impacts CO_2 on building occupants, how CO_2 concentrations relate to occupant perception of bioeffluents, the use of indoor CO_2 to estimate ventilation rates, and the use of CO_2 to control outdoor air ventilation rates. Regarding the impacts of CO_2 on occupants, indoor concentrations are rarely close to limits in health-based guidelines, but much confusion still exists regarding CO_2 levels in ventilation and IAQ standards. Based on studies in different countries and different building types, most measured indoor CO_2 concentrations are less than

2000 mg/m^3, with a small fraction above 5000 mg/m^3, but none are in the range of health-based industrial exposure limits. Nevertheless, discussions of indoor CO_2 have long referred to the health impacts of exposure to CO_2. Similarly, many have referred to 1800 mg/m^3 as a "guideline value" for indoor CO_2 or even a requirement, most commonly attributing it to ASHRAE Standard 62. However, there is nothing in the standard declaring 1800 mg/m^3 or any other CO_2 concentration to be a health or comfort based guideline. The notion of an 1800 mg/m^3 "limit" is based on its association with a nominal ventilation requirement of 7.5 L/s per person under very specific circumstances as discussed in this paper. Research does exist that associates CO_2 concentrations of 1800 mg/m^3 with body odor acceptability of unadapted visitors to a building, but there are many other contaminants in indoor air that are not associated with the occupancy and CO_2 is not a good indicator for those contaminants.

As discussed here, indoor CO_2 concentrations are related to ventilation rates but that relationship is complicated by the multizone, transient nature of building airflow systems as well as variations in building occupancy (i.e., CO_2 generation) both temporally and spatially in buildings. The relationship can be used to estimate ventilation rates from CO_2 concentrations under only very specific and somewhat unusual circumstances, making CO_2 a questionable indicator of ventilation rates. Finally, ventilation and energy efficiency standards allow, in some cases require, the use of CO_2 concentrations to control ventilation rates.

In conclusion, we understand most of the issues related to indoor CO_2 in buildings, with this paper focusing on those issues relevant to ventilation and IAQ standards. Nevertheless, many practitioners and other users of these standards are still confused regarding these issues. Standards, guidelines and technical papers are available to reduce this confusion, but they only have an impact when read and understood. Other mechanisms need to be employed to get this information where it is needed. Training is one such mechanism. It is also important that technical papers which refer to the use of CO_2 concentrations for measuring ventilation rates or which discuss the place of CO_2 in ventilation and IAQ standards be very carefully reviewed by knowledgeable individuals.

7 ACKNOWLEDGEMENTS

The author gratefully acknowledges Stuart Dols, Steven Emmerich and Al Hodgson, for their helpful reviews of this paper.

8 REFERENCES

Apte, M.G., Fisk, W.J., and Daisey, J.M. (2000). Associations Between Indoor CO_2 Concentrations and Sick Building Syndrome Symptoms in US Office Buildings: An Analysis of the 1994-1996 BASE Study Data. Indoor Air, (10) 246-57.

ASHRAE. (2009). Indoor Air Quality Guide. Best Practices for Design, Construction, and Commissioning. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE. (2010). 62.1 User's Manual. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE. (2013a). ANSI/ASHRAE Standard 62.1-2013, Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE. (2013b). ANSI/ASHRAE/IES Standard 90.1-2013, Energy Standard for Buildings Except Low-Rise Residential Buildings. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE. (2014). ANSI/ASHRAE/IES/USGBC Standard 189.1-2014, Standard for the Design of High-Performance Green Buildings. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASTM. (2011). Standard Test Method for Determining Air Change in a Single Zone by Means of a Tracer Gas Dilution. E741-11, ASTM International, West Conshohocken, Pennsylvania.

ASTM. (2012). Standard Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation. D6245-12, ASTM International, West Conshohocken, Pennsylvania.

ASTM. (2014). Form and Style for ASTM Standards. ASTM International, West Conshohocken, Pennsylvania.

Bakó-Biró, Zs., Clements-Croome, D.J., Kochhar, N., Awbi, H.B., and Williams, M.J. (2012). Ventilation Rates in Schools and Pupils' Performance. Building and Environment, (48) 215-223.

Bekö, G., Lund, T., Nors, F., Toftum, J., and Clausen, G. (2010). Ventilation Rates in the Bedrooms of 500 Danish Children. Building and Environment, (45) 2289-2295.

Bennett, D., Wu, X., Trout, A., Apte, M., Faulkner, D., Maddalena, R. and Sullivan, D. (2011). Indoor Environmental Quality and Heating, Ventilating, and Air Conditioning Survey of Small and Medium Size Commercial Buildings. CEC-500-2011-043, California Energy Commission.

Berg-Munch, B., Clausen, G., and Fanger, P.O. (1986). Ventilation Requirements fo rthe Conrol of Body Odor in Spaces Occupied by Woman. Environment International, (12) 195-199.

Bluyssen, P.M., Fernandes, E.O., Groes, L., Clausen, G., Fanger, P.O., Valbjørn, O., Bernhard, C.A., and Roulet, C.A. (1996). European Indoor Air Quality Audit Project in 56 Office Buildings. Indoor Air, (6) 221-38.

Cain, W.S., Leaderer, B.P., Isseroff, R., Berglund, L.G., Huey, R.J., Lipsitt, E.D. and Perlman, D. (1983). Ventilation Requirements in Buildings: I. Control of Occupancy Odor and Tobacco Smoke Odor. Atmospheric Environment, (17) 1183-1197.

CEC. (2013). Building Energy Efficiency Standards for Residential and Nonresidential Buildings, Title 24, Part 6, California Energy Commission.

CEN. (2007). Ventilation for non-residential buildings - Performance requirements for ventilation and room-conditioning systems. EN 13779, European Committee for Standardization.

Coley, D.A. and Beisteiner, A. (2002). Carbon Dioxide Levels and Ventilation Rates in Schools. International Journal of Ventilation, (1) 45-52.

ECA. (1992). Guidelines for Ventilation Requirements in Buildings, European Collaborative Action Report No. 11, EUR1449EN. Luxemborg Office for Official Publications of the European Communities.

EPA. (1991). Introduction to Indoor Air Quality. A Reference Manual, U.S. Environmental Protection Agency, U.S. Public Health Service, National Environmental Health Association, EPA/400/3-91/003.

EPA. (2006). BASE Data on Indoor Air Quailty in Public and Commercial Buildings. 402-C-06-002, United States Environmental Protection Agency.

Fanger, P.O. and Berg-Munch, B. (1983). Ventilation Requirements for the Control of Body Odor. Proceedings of Engineering Foundation Conference on Management of

Atmospheres in Tightly Enclosed Spaces, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 100-106.

Gaihre, S., Semple, S., Miller, J., Fielding, S., and Turner, S. (2014). Classroom Carbon Dioxide Concentration, School Attendance, and Educational Attainment. Journal of School Health, (84) 569-74.

Hou, J., Sun, Y., Kong, X., Wang, P., Zhang, Q., and Sundell, J. (2015). Ventilation Rate in Dwellings And Its Association With Children's Health In Tianjin, China. Proceedings of Healthy Buildings Europe 2015, Eindhoven, The Netherlands, Paper ID473.

Hunt, C.M. (1980). Air Infiltration: A Review of Some Existing Maesurement Techniques and Data. In Building Air Change Rate and Infiltration Measurements, ASTM STP 719. C.M. Hunt, J.C. King and H.R. Treschsel, Eds., American Society for Testing and Materials, 3-23.

Iwashita, G., Kimura, K., Tanabe, S., Yoshizawa, S., and Ikeda, K. (1990). Indoor Air Quality Assessment Based on Human Olfactory Sensation. Journal of Architectural Planning and Environmental Engineering, (410) 9-19.

Klauss, A.K., Tull, R.H., Roots, L.M., and Pfafflin, J.R. (1970).History of the Changing Concepts in Ventilation Requirements. ASHRAE Journal, (12) 51-5.

Lawrence, T. (2008). CO_2 Criteria for Monitoring Outdoor Air. ASHRAE Journal, (50,12) 18-27.

Limb, M.J. (2001) A review of international ventilation, airtightness, thermal insulation and indoor air quality criteria. TN 55. Coventry, Great Britain: Air Infiltration and Ventilation Centre.

Persily, A.K. (1997). Evaluating Building IAQ and Ventilation with Indoor Carbon Dioxide. ASHRAE Transactions, (103, 2) 193-204.

Rasmussen, C., Clausen, G.H., Berg-Munch, B., and Fanger, P.O. (1985). The Influence of Human Activity on Ventilation Requirements for the Control of Body Odor. Proceedings of CLIMA 2000 World Congress on Heating, Ventilating and Air-Conditioning, (4) 357-362.

Satish, U., Mendell, M.J., Shekhar, K., Hotchi, T., Sullivan, D., Streufert, S., and Fisk, W.J. (2012). Is CO_2 an indoor pollutant? Direct effects of low-to-moderate CO_2 concentrations on human decision-making performance. Environmental Health Perspectives, (120) 1671-7.

Shendell, D.G., Prill, R., Fisk, W.J., Apte, M.G., Blake ,D., and Faulkner, D..(2004) Associations Between Classroom CO_2 Concentrations and Student Attendance in Washington and Idaho. Indoor Air, (14) 333-41.

USGBC. (2014). LEED v4 for Building Design and Construction. U.S. Green Building Council.

Yaglou, C.P., Riley, F.C., and Coggins, D.I. (1936). Ventilation requirements. ASHVE Transactions, (42) 133-162.

DEMAND CONTROLLED VENTILATION IN PRACTICE: CASE STUDY

Bart Merema[*1], Maarten Sourbron[2], Johan Verplaetsen[3], Paul Van den Bossche[4], Hilde Breesch[1]

[1] KU Leuven – Technologiecampus Gent
Gebroeders De Smetstraat 1
Gent, Belgium
**bart.merema@kuleuven.be*

[2] KU Leuven- Department of Mechanical Engineering
Celestijnenlaan 300A
3001 Leuven, Belgium

[3] MENERGA
Nieuwlandlaan 16A/B409
3200 Aarschot, Belgium

[4] BBRI
Avenue P. Holoffe 21
B-1342 LIMELETTE

ABSTRACT

Demand controlled ventilation (DCV) can reduce the energy use significantly compared to a constant air volume (CAV) system. However, there is still a large uncertainty about the real energy savings and the ventilation efficiency. Furthermore, control and operation of the system are more complex. To formulate answers to these questions, measurements on a DCV system in a university building in Ghent, Belgium provide insight in the system operation and performance and the air distribution in the classrooms. Monitoring is carried out in March and May 2015.

During the measurements the VAV operation related to the set points was analysed. The active power demand for the fan and the thermal power demand for the heating coil was monitored. Furthermore, the CO_2 concentration was measured at five positions in the classroom to determine the ventilation efficiency at maximum and reduced air flow rate.
The results for the operation of the air handling unit (AHU) show that DCV is able to control the indoor set points for both CO_2 concentration and room temperature. When there is no occupancy in the classroom the DCV system operates at 32% of the time at a minimum air flow rate during operating hours, which decrease the active power demand for the fan.

The CO_2 concentration was measured at five positions in the classroom to characterise the ventilation efficiency. The measurements show that at a high air flow rate the lowest concentrations were found for the position in the middle of the classroom. Comparing a high and low air flow it was found that during a low air flow the difference in CO_2 concentrations for the five positions is at maximum 150 ppm. For a high air flow the position in the middle of the classroom has a maximum difference of 500 ppm lower compared to the front and back of the classroom. Furthermore, it was found that the room sensor did not deliver representative results especially in case of low air flow rates.

KEYWORDS

Demand Controlled Ventilation, Energy savings, Ventilation efficiency, Measurement, Air handling unit

1 INTRODUCTION

Demand controlled ventilation (DCV) is a ventilation system that automatically controls the air flow rate corresponding to the demand in the room, characterized by e.g. occupancy or CO_2-concentration. DCV can operate at reduced air flow rates during a large amount of the occupancy reducing the energy use of the fan and ventilation losses for heating.

This ventilation system can reduce the energy use in buildings. In a study by Mysen et al. (2005) the energy demand was reduced respectively by 38% and 51% for a CO_2 controlled DCV compared to a constant air volume system (CAV) in a school building. Wachenfeldt et al. (2007) studied the energy demand for a CO_2 controlled DCV in a school building with use of both simulations and measurements. The energy demand for heating was reduced by 21% and for the fan by 87% compared to a CAV. The volume flow in this study was reduced by 50% for the simulated time period (11-17 Nov 2002). Maripuu (2009) measured the performance of a DCV system in a university building. With the decreased airflow rates the energy effect for the fans decreases respectively by 50%. This was monitored for a ventilation system with approximately 3500 operating hours/year.

However, there is still a large uncertainty about the ventilation efficiency of the room. Furthermore, control and operation of the mechanical system are more complex. Therefore, this paper evaluates the operation and performances of a DCV and the ventilation efficiency in the room in the new test classrooms at the Technology Campus Ghent of KU Leuven (Belgium). Measurement data of March and May 2015 are discussed.

2 METHOD

First, a description of the case study building and the system is presented. Afterwards, the measurement setup for the evaluation of the operation of the AHU and the ventilation efficiency is shown.

2.1 Case study building and system description

Figure 1 shows the plan and cross section of the new test classrooms at the Technology Campus Ghent of KU Leuven (Belgium). This case study building contains 2 large classrooms with 140 m^2 floor area and a maximum occupancy of 100 students each. The building is built according to the Passive House standard. This means that, amongst others, the building is very airtight.

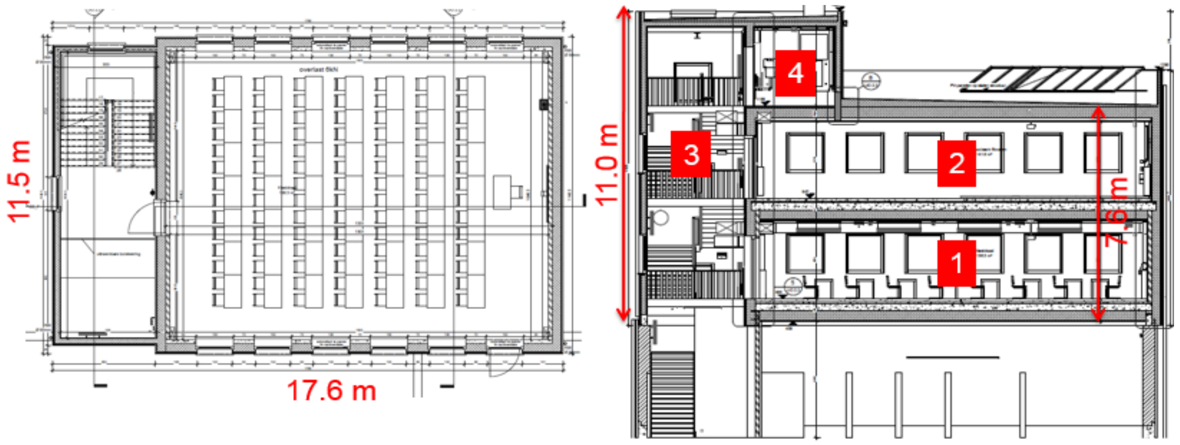


Figure 1 Plan and cross-section of the test classrooms: (1) classroom 1st floor (2) classroom 2nd floor (3) staircase (4) Technical room with AHU

Balanced mechanical ventilation is provided with a total supply airflow of 4400 m³/h and a maximum power demand of 1.57 kW for the supply fan and 1.33 kW for the extraction fan. The AHU regulates the VAV boxes by sending a request signal to control the airflow based on CO_2-concentrations and operative temperature in the classrooms. Furthermore, the AHU controls the return signal of the VAV boxes for the extraction of air. Each classroom is a single zone with a supply and return VAV. For heating purposes, the air is preheated by air-to-air heat recovery, i.e. two cross flow plate heat exchangers connected in series with an efficiency of 78% according to EN 308 (1997). Additionally, a heating coil of 7.9 kW is integrated in the supply ducts of each class room. A modular bypass is included. A scheme of this HVAC system is shown inFigure . For the evaluation, the ventilation efficiency and indoor air quality (IAQ) in the classrooms, the operation of the AHU and the power demand for both the fan and heating batteries are analysed.

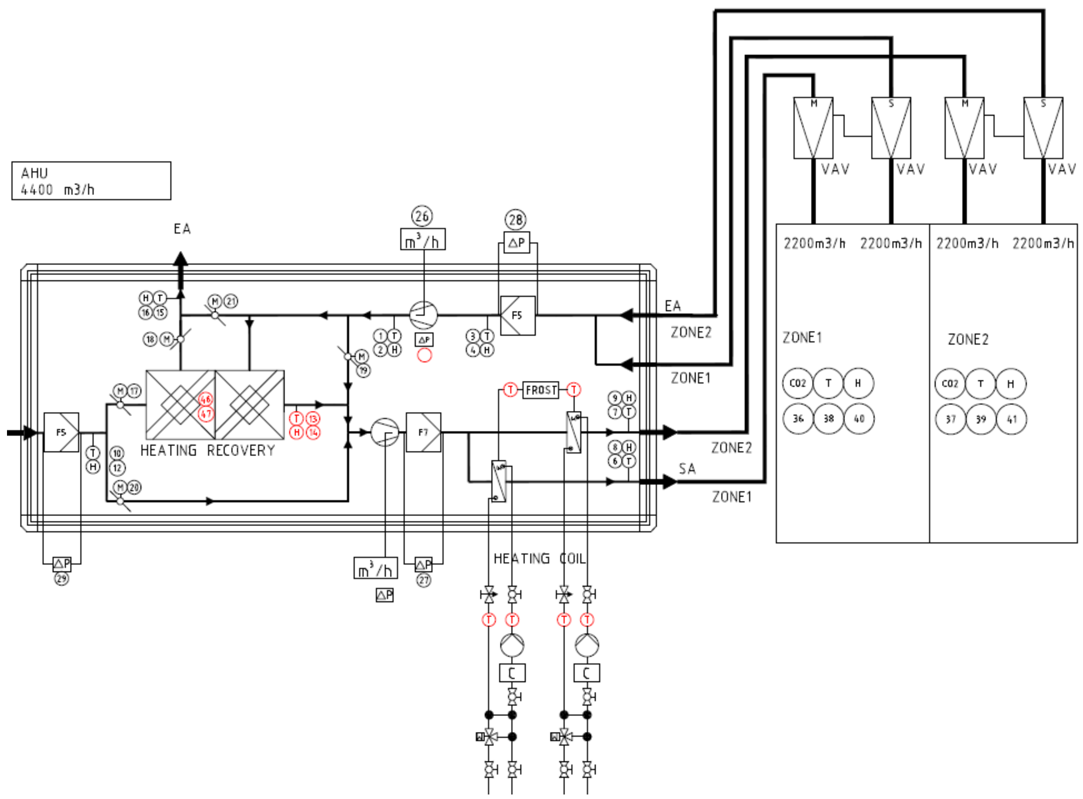


Figure 2 Scheme of air handling unit in the test classrooms of KU Leuven Technology Campus Ghent

2.2 Ventilation efficiency and operation/control of AHU measurement setup

First, the applied control strategy is evaluated. Three different scenarios for system control are evaluated, which are: frequency control, constant pressure and throttling. The control strategy that is used for the current system is constant pressure. To test the frequency control, the frequency of the supply fan is gradually increased from 10 to 60 Hz. The request position of the VAV damper is not adjusted in this scenario and is set to 50%. In the constant pressure scenario the duct pressure is maintained at 150 Pa. For the throttling scenario, the air flow is controlled by adjusting the position of the VAV damper. The frequency of the supply fan is not changed for this scenario and is set to 40 Hz.

To evaluate the detailed operation of the DCV system, the following data of the building monitoring system (BMS) are analysed during occupancy:

- supply and extract air flow rates in each classroom
- electricity use and active power of the fans
- supplied energy to and control of the heating coils
- control and position of VAVs
- position of bypass and recycling valves temperatures in air handling unit
- operative temperature and CO_2-concentration in the classrooms

Data are logged every minute. A typical week from 2 till 5 March is discussed.

The ventilation efficiency is evaluated by comparing the CO_2 concentrations during occupied hours in the classroom at five different measurement positions, shown on Figure . The $CO_2$5 sensor is placed near the CO_2 sensor of the room to verify if the values of the room sensor, used in the control system of the DCV, are reliable. The other sensors are placed near the people sitting in the classroom. The specifications of the sensors used are listed in Table 1.

Two scenarios are discussed: (1) high flow rate: air flow rate is set at 2000 m^3/h and is maintained at this value for the complete measurement period, (2) low flow rate: the frequency of the supply fan is set at 10 Hz which resulted in an air flow of 430 m^3/h. Measurements start from a constant situation in which the air temperature is uniformly distributed in the classroom. In addition, the occupancy of the room is more concentrated around the measurement position in the middle of the room compared to the other positions

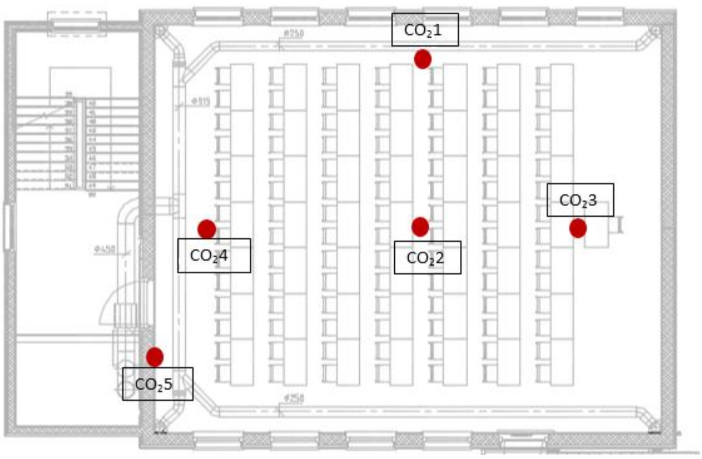


Figure 3 Plan of the classroom with the CO_2 measurement positions at 0.9 m high and 2.0 m high for $CO_2$5.

Table 7 Measurement equipment

Parameter	Type sensor	Accuracy	Sensor n°
CO_2	VAISALA GMW93, CO_2+T	±30 ppm + 2% of reading	$CO_2$4, $CO_2$5
CO_2	Telaire 700LI	±50 ppm or ±5% of reading	$CO_2$1,$CO_2$2, $CO_2$3
CO_2	E+E 80	±50 ppm + 2% of reading	CO_2 room

3 RESULTS

First, the results of the evaluation of the AHU control are presented. Afterwards, the operation of the DCV is analysed in detail. Finally, results of CO_2 measurements in the classroom are shown to evaluate the ventilation efficiency.

3.1 Control of the system

The results of the measurements of the normalised power demand as a function of the normalised ventilation flow rate in the three different scenarios are compared to the theoretical curve in Figure . The results show that frequency control is almost as efficient as the theory regarding the real power demand. The difference with between theory and frequency control is at maximum 5%. For constant pressure, i.e. the used control, the difference with the theory and frequency control real power demand, is for the lower air flows at maximum 150%. With larger air flows the difference in real power demand is reduced compared to the theory and the frequency control.

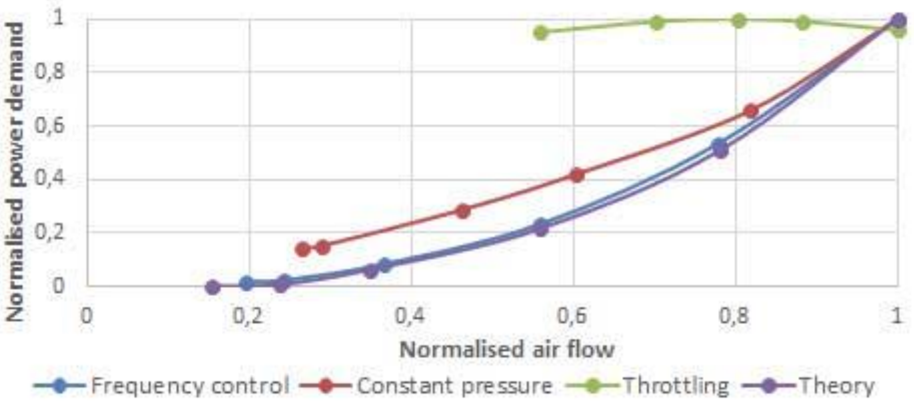


Figure 4 Real normalised power demand for different control strategies for varying air flow rates

3.2 Operation of DCV

In Figure 5 the CO_2 concentration and air flow is shown related to the occupancy and CO_2 set point in the classroom. Figure 6 shows the room, supply and outdoor temperature and the occupancy in the classroom. The results in both figures are from the time period of 2-5 March 2015. During this period the average outdoor temperature was 6.6°C with an average maximum and minimum of respectively 10.3°C and 2.7°C (KMI, 2015). The AHU is switched on from 8:00-18:00h except for the third day, where the AHU was switched off at 14:30h. During operating hours it can be seen that the air flow is approximately 1600 m^3/h during the hours with occupancy. The results also shows that after the end of a lecture the system still operates at an air flow rate of 1600 m^3/h to reduce the CO_2 concentration and the room temperature. When the classroom is not occupied the AHU is operating at a minimum air flow rate of 400 m^3/h.

The air flow in the classroom increases as a result of (1) a high CO_2 concentration or (2) a low room temperature. When the CO_2 set point (1100 ppm) is exceeded during operating hours, the air flow rate increases to a maximum of 1600 m^3/h to reduce the CO_2 concentration. For example, on the second day at 13:30h, the CO_2 concentration exceeds the set point as a result the air flow increases from 400 to 1600 m^3/h. Furthermore, on the fourth day it is noticed that the time switches of the AHU was switched off during the lunch break and switched on at 13:45h. However, the class already started at 13:30h, which causes a peak in CO_2 concentration of 2000 ppm.

Figure 5 clearly shows that when the AHU is switched on in the morning the air flow increases to a maximum of 2000 m^3/h. This is a result of a low room temperature in the morning. At night the system operates in standby mode and the heating set point is decreased from 22°C to 17°C. To increase the room temperature according to the heating set point the supply temperature increases to 40°C for a short time. Furthermore, on the first day of the measurements it can be seen that after the hours with occupancy the air flow rate increases from 1600 to 2200 m^3/h. When the lessons end at 12:30h it can be seen that the room temperature decreases below the heating set point to 21°C. At this time the supply temperature increases to 27°C according to the heating set point of the room. This indicates that still some optimisation is needed in the temperature control.

Moreover, the results show that there are still some problems with the operation of the ventilation system. It is shown that during high CO_2 concentrations the air flow only increases to 1600 m^3/h instead of the maximum of 2200 m^3/h. Furthermore, the response time of the system can be improved, since the air flow rate only increases when the CO_2 set point is exceeded.

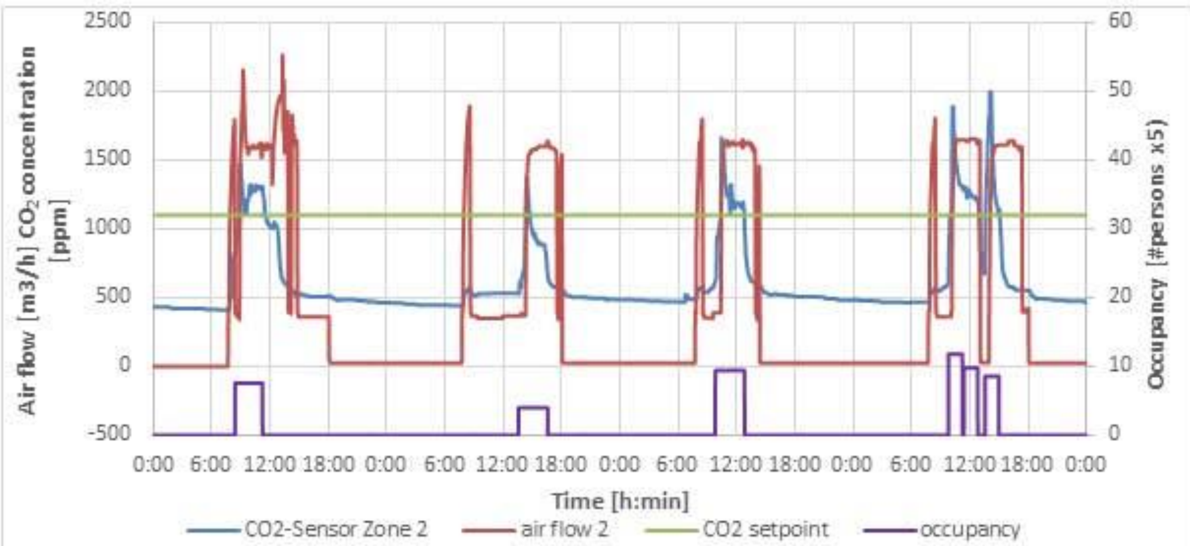


Figure 5 Operation of the flow according to the CO_2 concentration for the classroom (2-5 March 2015)

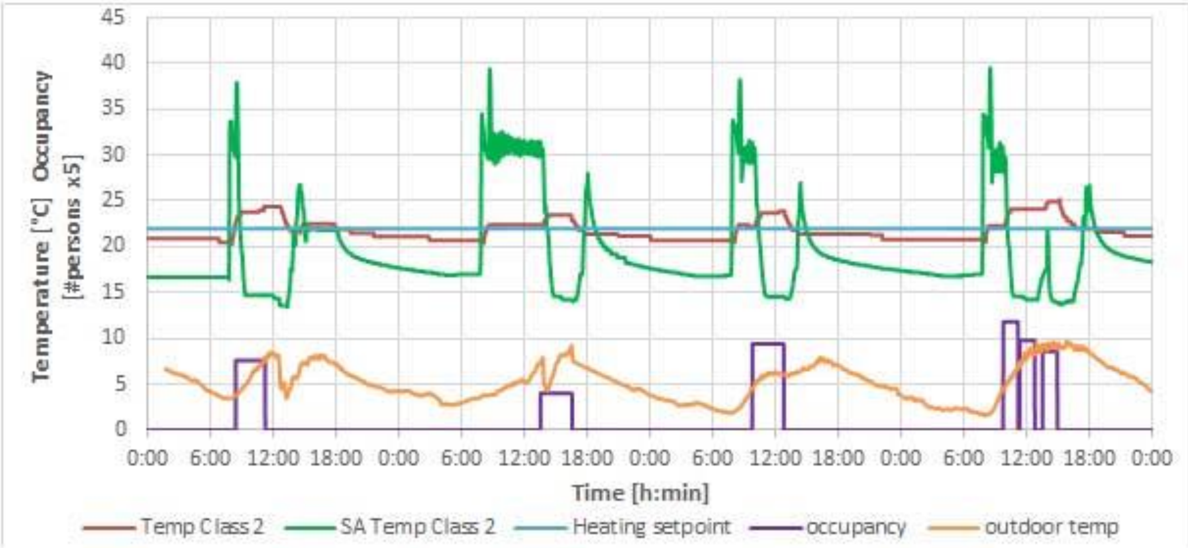


Figure 6 Operation of the supply and room temperatures for the classroom (2-5 March 2015)

In addition, Figure shows the measurements of the active power demand of the fan and the thermal power demand for the heating coil for the classroom during the same period. These measurements are in line with the conclusions from the air flow rate and supply temperature on Figure and 6. The highest peaks of 6 kW in the morning are found for the thermal heating power demand when the AHU is switched on. At this time point it was shown in Figure 6 that the supply temperature increases to 40°C to heat up the room according to the heating set point. Furthermore, there is a second peak of 3 kW in thermal heating power after the lecture on the first three days. When a lecture ends the room temperature rapidly decreases to levels below the heating set point. At this time the supply temperature increases from 15°C to 27°C.

For the fan electrical power it is shown that during high air flow rates the power use increases. When comparing the results for the fan electrical power demand and the air flow rate it can be seen that for the operating hours with no occupancy the fan electrical power decreases to 0.2 kW (13% of max) by an air flow rate of 400 m^3/h (18% of max).

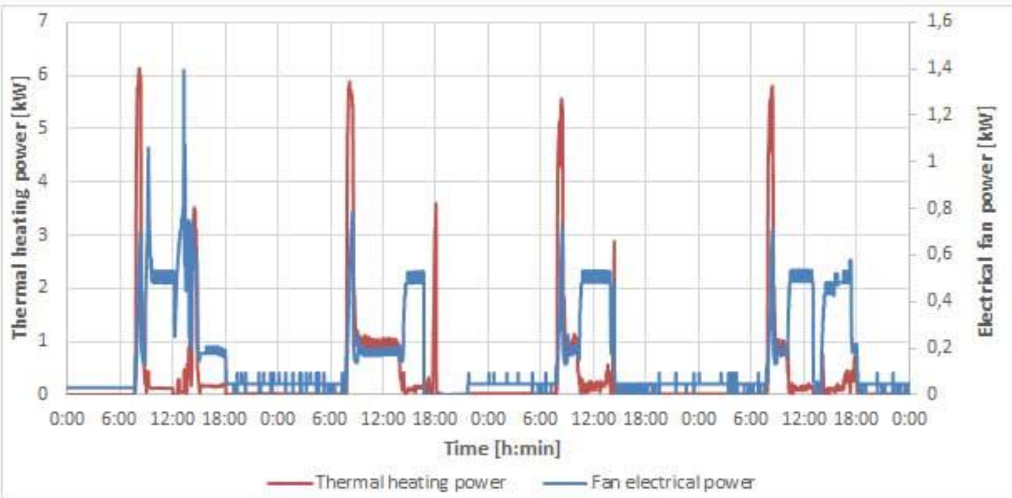


Figure 7 Electrical fan power and thermal heating power for the classroom (2-5 March 2015)

3.3 CO_2 measurements of ventilation efficiency in the classrooms

Figure 8 and 9 compare the CO_2 concentrations at five different locations in the classroom in case of a high respectively a low air flow rate. For the high air flow rate, shown in Figure 8, The difference in CO_2 concentration measured in the room is at some point 600 ppm, measured for the CO_2 2 and CO_2 room sensor. For the positions in the front (CO_2 3) and back (CO_2 4) the difference with the sensor in the middle is at maximum 500 ppm. The maximum values in the classroom near the people is 1200 ppm, which is still acceptable. According to the EN 15251 (2007) the maximum difference between inside and outside CO_2 levels is 800 ppm.

For the low air flow rate shown in Figure 9, the highest concentrations are found for the sensor placed in the front and the back of the classroom. Compared to the sensors placed in the middle of the classroom the concentration is at maximum 100 ppm higher.

The results for both high and low air flow rate indicate that for the position in the middle of the classroom, more fresh air is supplied compared to the measurement positions in the front and back of the classroom. This effect is especially noticed during a high air flow.

Furthermore, these measurements show that the room sensor do not deliver representative results especially in case of low air flow rates. The CO_2-concentration of this sensor was 300 ppm lower compared to the sensor located nearby. In case of a high air flow rate, the room sensor indicates always the highest concentration. The difference between the room sensor and the neighbouring sensor CO_2 5 is around 50 ppm, which is within the accuracy range. This indicates that the accuracy of the sensor is not good for measurements during low air flow rates.

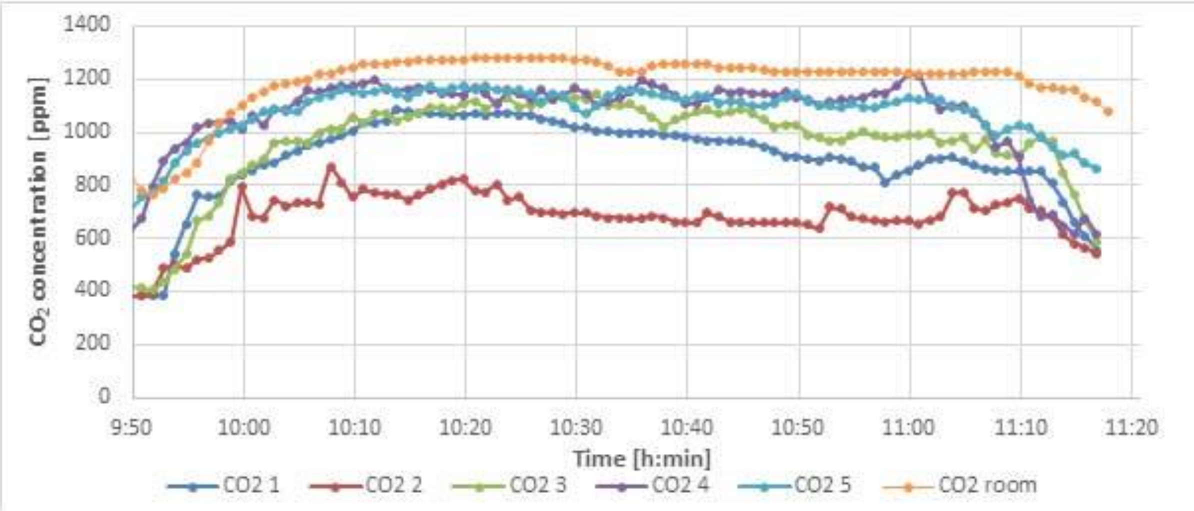


Figure 8 CO_2 concentration in the classroom during lecture (May 18, 9:50-11:10) with an air flow of 2000 m^3/h and an occupancy of 70 persons

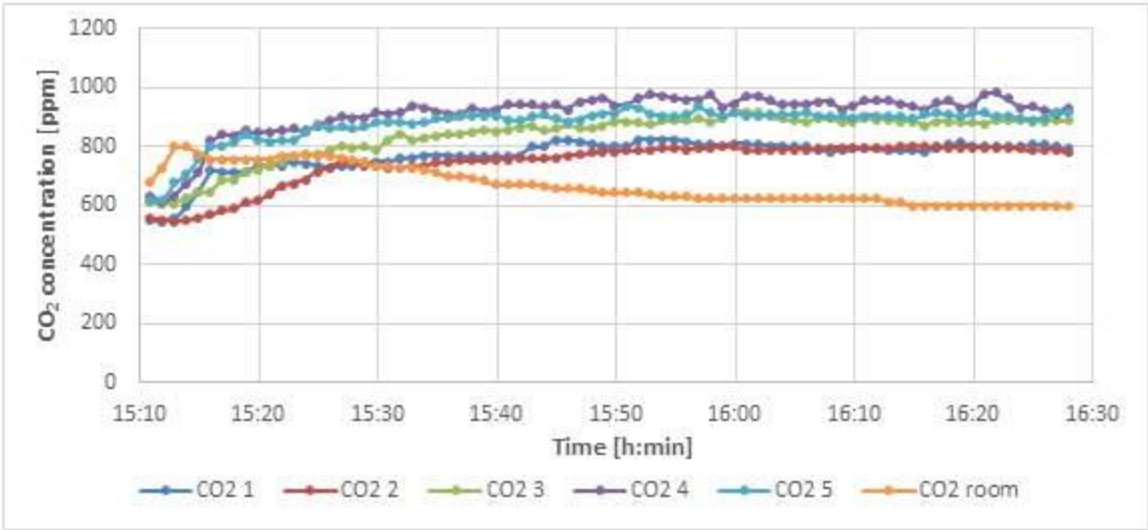


Figure 9 CO_2 concentration in the classroom during lecture (May 12, 15:10-16:30) with an air flow of 430 m^3/h and an occupancy of 30 persons

4 CONCLUSIONS

Operation and performances of a DCV and the ventilation efficiency in the room was evaluated in the new test classrooms at the Technology Campus Ghent of KU Leuven (Belgium). For the monitoring of the DCV system the operation was evaluated for a short time period of four days in March 2015. The control strategy showed that for frequency control the system is almost as efficient as the theory. Ventilation efficiency was evaluated by CO2 measurements in May at a high and low air flow rate.

The operation of the AHU shows that a demand controlled ventilation system responds well to the control parameters, which are the CO_2 concentration and the room temperature. The ventilation system is able to control and maintain the CO_2 and room temperature set points by adjusting the air flow rate to the demand. The active power demand of the fan and the thermal heating power of the heating coil shows that during operating hours the system does not operate at maximum power. During the hours with no occupancy the system runs at a minimum air flow rate of 400m^3/h this is in total 32% of the operating hours.
Results show that during CO_2 measurements with a high air flow rate the highest concentrations were found for the position in the front and back of the classroom, with a maximum of 1200 ppm. The difference with measurement positions in the middle of the classroom was at maximum 500 ppm. At a low air flow rate the average value the maximum CO_2 concentration was 1000 ppm for the position in the back of the classroom the maximum difference was 400 ppm with the room sensor. However, the results showed that the values for the room sensor are not reliable with low air flow rate since the sensor placed next to the room sensor measured values which were on average 300 ppm higher.

This study shows that demand controlled ventilation is able to control the indoor set point parameters. The ventilation system operates most of the time at a reduced air flow rate, which reduces the electrical use for both the AHU and the heating system. However, it was found out that there are still some small problems with the operation.

5 ACKNOWLEDGEMENTS

This research is funded by the Institute for the Promotion of Innovation through Science and Technology in Flanders (IWT-Vlaanderen). This support is gratefully acknowledged.

6 REFERENCES

EN 308 (1997). *Heat exchangers - Test procedures for establishing performance of air to air and flue gases heat recovery devices.* CEN, Brussels.

EN 15251 (2007). *Indoor environmental input parameters for design and assessment of energy performance of buildings- addressing indoor air quality, thermal environment, lighting and acoustics.* CEN, Brussels.

KMI (2015, June 30), *Retrieved from http://www.meteo.be/meteo/view/nl/18905394-maart+2015.html*

Maripuu, M. L. (2009). *Demand controlled Ventilation (DCV) systems in commercial buildings: functional requirements on systems and components*. PhD Göteborg: School of Electrical and Computer Engineering, Chalmers tekniska högskola.

Mysen M., Berntsen S., Nafstad P., Shild P.G. (2005). *Occupancy density and benefits of demand-controlled ventilation in Norwegian primary schools*, Energy and Buildings 37, 1234–1240.

Wachenfeldt B.J., Mysen M., Schild P.G. (2007). *Air flow rates and energy saving potential in schools with demand-controlled displacement ventilation*, Energy and Buildings 39, 1073-1079.

RESIDENTIAL DEMAND CONTROLLED EXTRACT VENTILATION COMBINED WITH HEAT RECOVERY VIA A HEAT PUMP

Ivan Pollet*1,2, Clint Coulier1, Anneleen Vens1, Filip Grillet3

1 Renson Ventilation
Maalbeekstraat 10
Waregem, Belgium

2 Ghent University
Department of Biosystems Engineering
Coupure Links 653
Gent, Belgium
**Corresponding author:*
ivan.pollet@ugent.be

3 3E nv
Kalkkaai 6
Brussel, Belgium

ABSTRACT

In this study the performance of a residential demand controlled (DC) extract ventilation system with an air-to-water heat pump was analysed via dynamic simulations. A real life test case was setup to validate results. The ventilation system controls automatically the extract air in functional as well as habitable rooms, ensuring indoor air quality (IAQ). The total extract rate is mixed with outdoor air as heat source of the air-to-water heat pump (2.5 kW at standard reference conditions). Domestic hot water (DHW) as well as space heating (SH) can be alternatively supplied. A gas boiler as back-up guarantees comfort temperature (SH and DHW) at all time. The ventilation heat losses and fan consumption are reduced by the demand control. The heat recuperation on the extract air by means of the heat pump provides a further reduction of the energy consumption.

Dynamic simulations were performed in Virtual Environment (VE) on an apartment geometry (total heating demand of ±3 kW (at -8°C)) for a 2 and 4 persons family. Results of ventilation performance, heat pump performance (off or DHW or SH and (S)COP), back-up (on/off) and heating demand were analysed. Besides a comparison was made with common mechanical extract ventilation and double flux heat recovery systems.

In the apartment case 6% of the year the back-up is activated, 53% of the year, the heat pump is active. Using mixed air instead of outdoor air as heat pump source has led to an augmentation of the SCOP by 0.4 to 3.64 for SH and by 0.3 to 3.37 for DHW. Compared to a demand controlled mechanical extract ventilation system (MEV) an average reduction of the yearly primary energy demand by 31% is achieved due to the use of a heat pump. Compared to a mechanical ventilation system with heat recovery (MVHR) on half ventilation rate, a reduction of 16 to 23% in primary energy use is achieved mainly because of the demand control and the strongly reduced fan energy consumption.

For the simulated apartment and the test dwelling 18% of the ventilation heat losses are recovered towards SH and DHW. In case of dwellings with a higher heating demand, more of the ventilation heat losses can be recovered, since the operating period of the heat pump for SH increases.

KEYWORDS

Demand controlled ventilation, air-to-water heat pump, heat recovery, hybrid system

1 INTRODUCTION

Energy efficiency is at the heart of the European Union's 2020 strategy for smart, sustainable and inclusive growth and of the transition to a resource efficient economy. Improved energy efficiency is one of the most cost effective ways to enhance the security of energy supply and to reduce emissions from greenhouse gases. In many ways, energy efficiency can be seen as Europe's biggest energy resource. This is why the Union has set itself a target for 2020 to save 20% of its primary energy consumption (Hogeling, 2015). In order to achieve these savings, especially after the adoption of the Directive 2002/91/EC, local authorities have introduced new and more stringent requirements in building regulations. Particularly, the focus consists in reducing the thermal energy need working on the building envelope and using passive techniques and, at the same time, in improving the efficiency of systems such as heating, cooling, ventilation and lighting (Madonna & Bazzocchi, 2013). Regarding systems, heat pumps are one of the most interesting solutions as they represent a valid alternative to conventional systems, due to the reduced primary energy use (Madonna & Bazzocchi, 2013).

Mechanical ventilation is commonly provided by natural air inlets and one extract fan (MEV) or a supply and extract fan with heat recovery (MVHR). MVHR reduces the heat amount needed to condition the supply airflow, however they also require more power to operate than an extract fan (Logue, et al., 2013). Ventilation devices can be made more energy efficient in different ways: use of less fans or more efficient fan(s), modulation of the air flow rate (demand controlled ventilation, DCV) and application of heat recovery (an air-to-air heat exchanger or an exhaust air heat pump) (Laverge, 2013). Exhaust air heat pumps (EAHP) make it possible to recover heat from exhaust air and are often defined as active regenerators (in that the energy consumption takes place and a heat transfer occurs between the two fluid streams, a transfer that is absent in the case of heat exchangers, which is in fact defined as passive recovery) or thermodynamic regenerators (since they operate according to a thermodynamic cycle) (Fracastoro & Serraino, 2010).

As stated by (Hogeling, 2015) products cannot longer be considered as just parts from the shelf. Products are more and more considered as sub-systems, as they are including control-devices, electronics, other auxiliary functions, … Following this philosophy Renson, a Belgian manufacturer of ventilation, ventilative cooling and sun protection solutions, has integrated a demand controlled mechanical extract ventilation device (DCMEV) with an air-to-water heat pump. Renson refers to this system as the E^+ system. As suggested by (Fracastoro & Serraino, 2010) EAHPs are able to cover a relevant fraction of the thermal energy requirements for air-conditioning and allow a remarkable reduction in the primary energy consumption even though they often require an auxiliary plant.

The aim of the present study was to compare the performance of the E^+ system with traditional ventilation systems namely demand controlled mechanical extract ventilation (DCMEV) and mechanical ventilation with heat recovery (MVHR) in combination with a gas boiler. To this end dynamic simulations were carried out in Virtual Environment. The ventilation performance, heat pump performance (off, active for DHW or SH and (S)COP), back-up performance (on/off) and primary energy use were analysed. Besides, first measurements on a test case were analysed.

2 SYSTEM DESCRIPTION

The E^+ system can be described as a demand controlled mechanical extract ventilation system (DCMEV) of which the total extract air flow is mixed with an outdoor air flow as heat source for an air-to-water heat pump (2.5 kW at standard reference conditions). A gas boiler (no electrical resistance) is used as back-up to guarantee comfort temperatures for space heating (SH) and domestic hot water (DHW) at all times. Domestic hot water (DHW) as well as space heating (SH) can be alternatively supplied. For reasons of legionella prevention, the hot water storage is warmed up to 60°C once a week by the gas boiler.

In contrast to other EAHP, the ventilation rate does not augment when the heat pump is activated for space heating (SH) or domestic hot water (DHW), in order not to increase the ventilation losses. The ventilation system controls automatically the extract air in functional as well as habitable rooms, ensuring indoor air quality (IAQ). The total extract flow is mixed with outdoor air in order to reach a constant air flow rate of 350 m³/h as heat source of the air-to-water heat pump.

The ventilation heat losses and fan consumption are reduced by the demand control. The heat recuperation on the extract air by means of the heat pump provides a further reduction of the energy consumption. In order to avoid summer overheating free cooling is integrated in the system. Free cooling means that the ventilation rate is set to its maximal capacity in order to achieve a cooling effect. Conditions to be met in order to activate free cooling were:

T_in > 21°C (to avoid interference with the heating setpoint)
T_out > 14°C
T_in > T_out

3 METHODS

By means of dynamic simulations in Virtual Environment (VE 2014.2.1.0) the performance of three system variants was calculated for an apartment located in Belgium. Simulation time step was 6 minutes. The three considered variants were:

1) Demand controlled mechanical extract ventilation with gas boiler for space heating and domestic hot water (**DCMEV**)
2) Demand controlled mechanical extract ventilation of which the total extract air flow is mixed with outdoor air as heat source of an air-to-water heat pump. A gas boiler as back-up guarantees comfort temperature at all time (E^+ further referred to as **DCMEV hybrid HP**)
3) Mechanical ventilation with heat recovery with gas boiler for space heating and domestic hot water (**MVHR**).

Besides, monitoring data of an installed DCMEV hybrid HP system were available from mid-2014 till mid-2015 for analysis of the real life performance.

3.1 Simulations

The simulated apartment is illustrated on Figure 1, having 90 m² floor area, a volume of 260 m³ and 76 m² exposed surface area. The floor plan is sketched on Figure 2, the large window in the living room with balcony is oriented to the southeast.

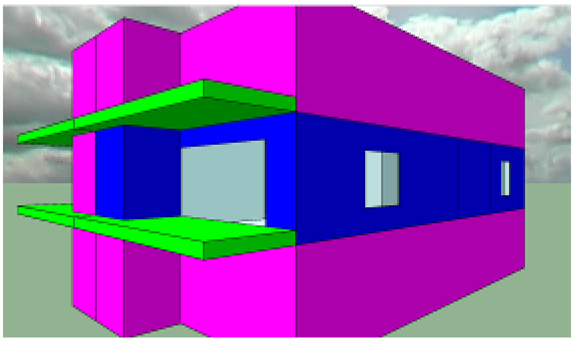


Figure 1: Geometry of the simulated apartment.

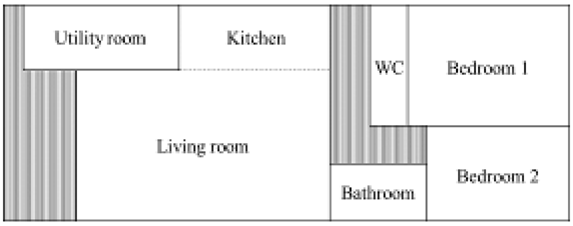


Figure 2: Floor plan of the simulated apartment.

Two occupancy schemes were considered:

- the first one consisting of two working adults and two children, while the heating setpoint was 20°C (referred to as **occ. 2+2**).
- the second considered occupancy scheme consisted of two retired persons which were more at home and had a higher heating setpoint namely 21°C (referred to as **occ. 2**).

The tapping pattern for both occupancy schemes was medium according to EN 16147. Other parameters are listed in Table 1. The apartment had a total heating demand of ±3 kW (at T_{out} = -8°C).

When demand controlled extract ventilation was applied, CO_2 values in the kitchen and bedrooms and relative humidity (RH) in the other wet rooms determined the ventilation rate. The total extract rate of the DCMEV variants varied between 80 and 260 m³/h. When MVHR was applied the total ventilation rate was permanently 230 m³/h which is lower than the maximal DCMEV ventilation rate, since there is no extract in the functional rooms (bedrooms). Weekly thermal disinfection of the water storage tank was simulated by an arbitrary value of 6 kW.

Table 1: Boundary conditions.

	DCMEV	**DCMEV hybrid HP**	**MVHR**
U-value			
wall	0.24 W/m².K	0.24 W/m².K	0.24 W/m².K
glazing	1.1 W/m².K	1.1 W/m².K	1.1 W/m².K
window frame	1.6 W/m².K	1.6 W/m².K	1.6 W/m².K
Solar factor glazing	0.62	0.62	0.62
Solar factor sunshading + glazing	0.101	0.101	0.101
Ventilation			
SFP	1 x 0.15 W/(m³/h)	1 x 0.11 W/(m³/h)	2 x 0.31 W/(m³/h)
Demand controlled	✓	✓	---
Extract in functional rooms	✓	✓	---
Free cooling	✓	✓	---
Heat recovery	---	Via heat pump	Via heat exchanger (η = 75%)
Cooker hood	150 m³/h	150 m³/h	150 m³/h
Floor heating temp. regime	35/30°C	35/30°C	35/30°C
DHW setpoint	45°C ± 5°C	45°C ± 5°C	45°C ± 5°C
Heat source for SH and DHW	Condensing boiler (η = 90%)	Air-to-water heat pump (2.5 kW) + condensing boiler (10 kW) (back-up)	Condensing boiler (η = 90%)

3.2 In-situ measurements

In a test case -situated in Zulte, Belgium- the DCMEV hybrid HP variant was installed and energy consumption (gas and electricity) was recorded. The detached dwelling had 172 m² floor area, a volume of 543 m³ and 414 m² exposed surface area. Weekly output was available for electricity use of the heat pump and gas consumption of the gas boiler (back-up) for a whole year (12 May 2014 till 10 May 2015). Due to a system fault no data was available from 10 till 23 November 2014. The data of the Renson weather station (< 10 km) was used to calculate the weekly mean outdoor temperature.

4 RESULTS – DYNAMIC SIMULATIONS

For the occupancy scheme referred to as 2+2, the ventilation system, the heat pump and the back-up performance were analysed. For both occupancy schemes the primary energy use was calculated and compared for the three considered systems.

4.1 Ventilation performance

The resulting hourly extract ventilation rate due to demand control and free cooling of the DCMEV system is plotted next to the constant extract ventilation rate of the MVHR system as a function of the outdoor temperature on Figure 3. Different ventilation levels due to demand control are visible as horizontal lines (minimum of 80 m³/h). The free cooling zone can clearly be distinguished in the range of 14-25°C, with varying mean ventilation rates between 80 and 260 m³/h. This has an impact on the indoor temperature as shown on Figure 4. When the outdoor temperature is high, indoor temperatures are clearly lower than when MVHR (without bypass) was applied. On colder days, this also occured due to the absence of a heat exchanger in a DCMEV system.

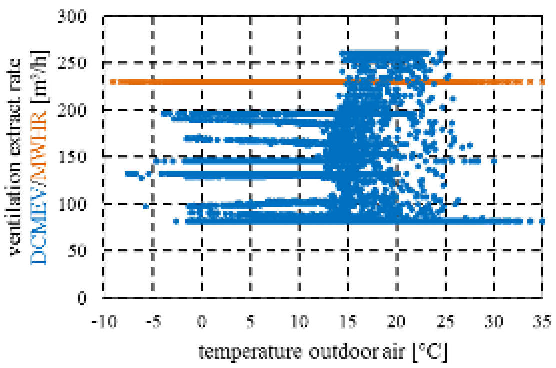


Figure 3: Difference on hourly ventilation extract rate using DCMEV with free cooling vs. MVHR (occ. 2+2).

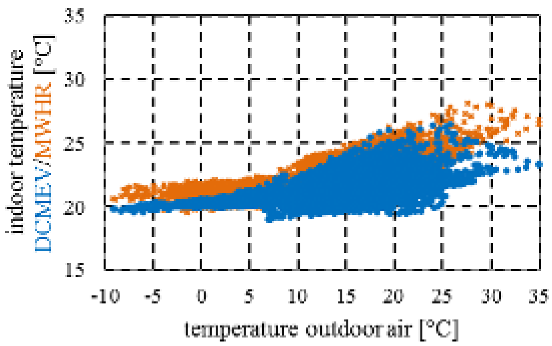


Figure 4: Difference on hourly indoor temperature using DCMEV with free cooling vs. MVHR without free cooling (occ. 2+2).

4.2 Heat pump performance

On Figure 5 the resulting hourly source temperature (temperature of the mixed air) is plotted as a function of the outdoor temperature. Using mixed air instead of outdoor air as heat source led to a mean increase of the supply temperature of the heat pump of 5°C (dotted line). When the outdoor temperature is low the source temperature has increased the most. This source temperature increase gives rise to an increased heat pump COP as illustrated on Figure 6. The mean SCOP for SH has increased by 0.4 to 3.64. The mean SCOP for DHW has increased by

0.3 to 3.37. This augmentation can be used to calculate the equivalent heat recovery effectiveness of the system (ε) according to formula (1) with Φ_{HP} the electricity demand of the heat pump during the heating season (1 October - 30 April; 1097 kWh for SH and 309 kWh for DHW) and Φ_{vent} the ventilation losses (3000 kWh). For the apartment 18% of the ventilation heat losses were recovered towards SH and DHW.

$$\varepsilon = \Phi_{HP} \,.\, \Delta COP / \Phi_{vent} \tag{1}$$

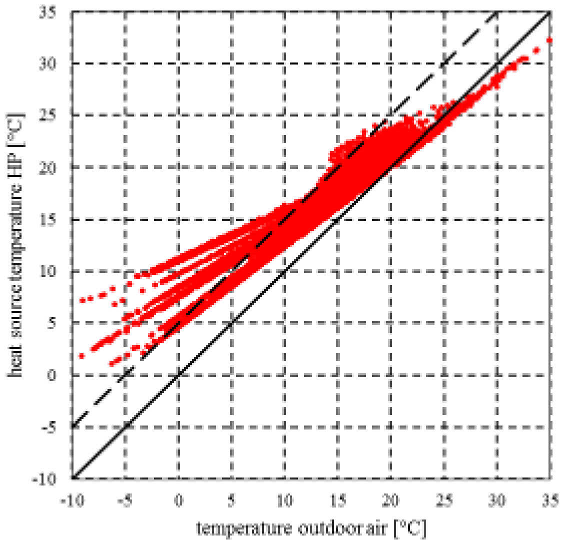


Figure 5: Effect of using mixed air instead of outdoor air as heat pump source.

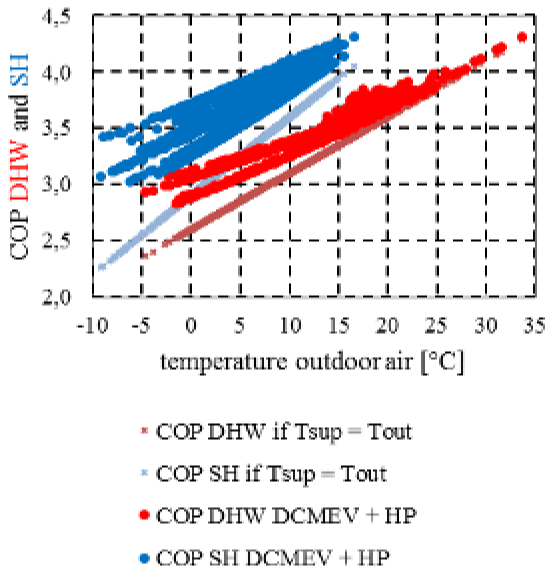


Figure 6: Effect of using mixed air instead of outdoor air as heat pump source on COP.

4.3 Back-up performance (gas boiler)

On Figure 7 the net heating demand of the heat pump and back-up is plotted on a weekly basis. The heating season is marked in green. The whole year the back-up is weekly used for thermal disinfection of the DHW at a temperature of 60°C. During the heating season the back-up was also active for SH when the heat pump was heating the storage tank (priority) and there was a demand for space heating at the same time. Overall the back-up was active for 6% of the whole year, mainly for disinfection. The heat pump was active 53% of the year meaning the rest of the year the heat pump fan was only used for ventilation purposes. On Figure 8 the cumulative net heating demand is plotted as a function of the hourly outdoor temperature. Since Belgium has a temperate maritime climate with moderate winters cumulative net heating demand is largest when the outdoor temperature is between 0 and 10°C what is the typical range of the Belgian heating season (Figure 12). Outdoor temperatures under -2°C are rarely and only few DHW demand occurred during that simulation period. The total final energy consumption and heating demand are summarized in Table 2 (system efficiency for SH of 90% and 77% for DHW). The simulation results indicated the back-up was used to cover 11% of the net heating demand (10% of total space heating demand and 16% of total domestic hot water demand).

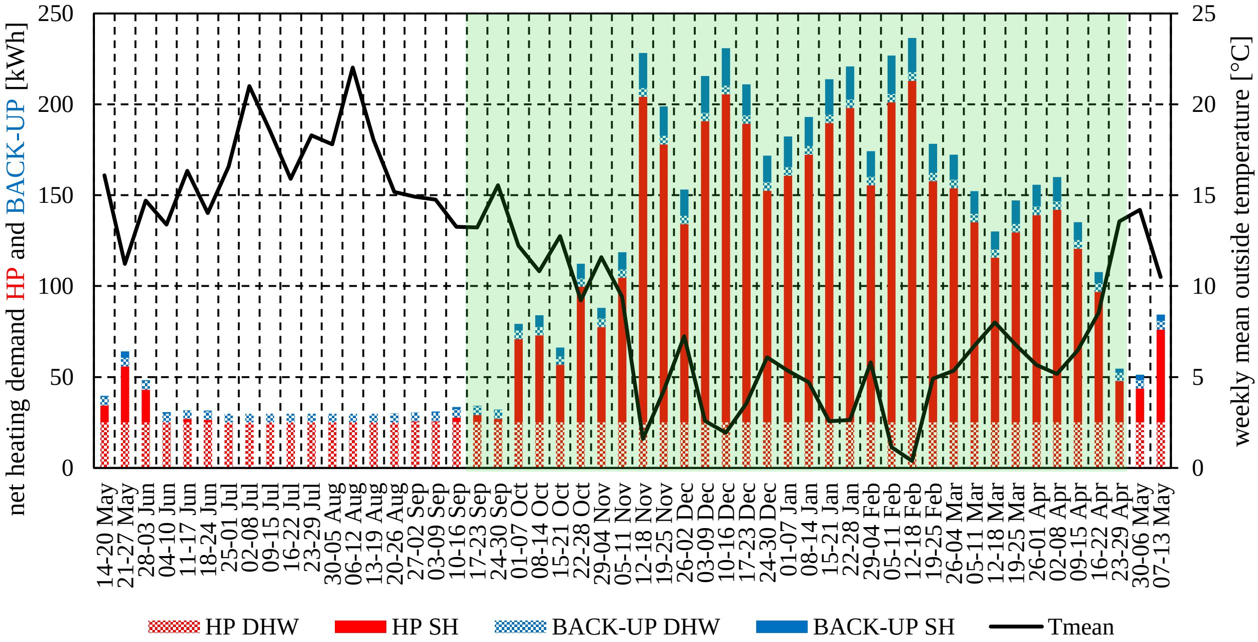


Figure 7: Simulated weekly net heating demand of the heat pump (red) and the gas-boiler (back-up, bleu). Typical Belgian heating season marked in green.

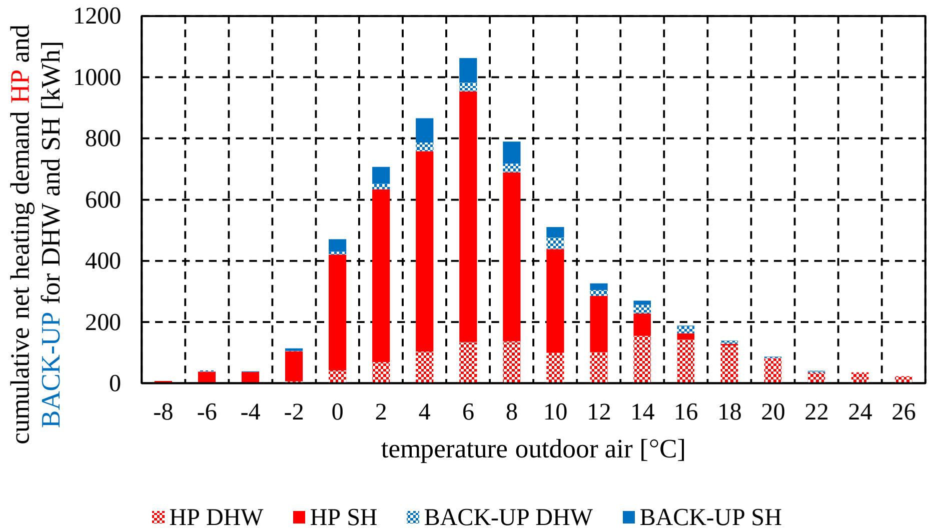


Figure 8: Cumulative net heating demand heat pump (HP) and gas back-up for DHW and SH as a function of hourly outdoor temperature.

4.4 Primary energy use

The performance on primary energy use for the different system variants and both occupancy schemes is presented on Figure 9. Since MVHR systems are frequently set at a lower rate by the occupants, a fourth variant with only 50% of the maximal ventilation rate is added (MVHR(50%)). Calculations were performed with a primary energy factor (PEF) for electricity of 2.5 and for natural gas of 1. The DCMEV hybrid HP variant has the lowest primary energy use: 31% lower than the DCMEV with gas boiler variant (for both occupancy schemes), respectively for occ. 2+2 and occ. 2 41% and 36% lower than the MVHR variant and 23% and 16% for the MVHR(50%) variant. The high primary energy use of the MVHR with gas boiler variant is mainly caused by the MVHR without demand control and the rather energy inefficient fans (high SFP cf. Table 1, even though the input data was based on a frequently used ventilation system) whereas the DCMEV hybrid HP variant uses only one fan for both ventilation and heat pump performance.

The impact of the occupancy scheme (from occ. 2+2 to occ. 2) was a reduction of 6% to 7% for respectively the DCMEV with gas boiler and the DCMEV hybrid HP variant. For the MVHR and MVHR(50%) with gas boiler the impact was higher namely 16%. The hot water consumption of the two people occupancy scheme was lower than for the two adults with two children but the comfort temperature was higher (21°C instead of 20°C). The increased comfort temperature for space heating had a larger impact on the system variants with natural air supply resulting in an overall smaller impact on the difference in primary energy consumption.

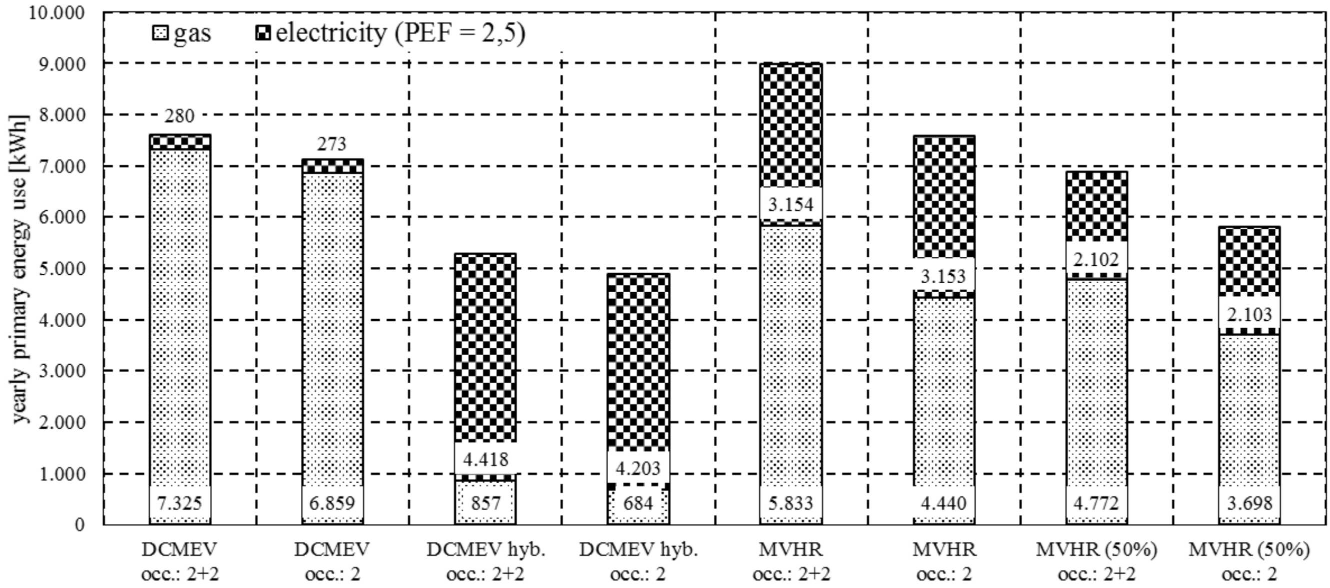


Figure 9: Primary energy use for different systems and for different occupancy schemes.

5. RESULTS - IN-SITU MEASUREMENTS

Since the actual COP of the heat pump was not logged, an average COP of 3.5 and system efficiency of 85% was assumed in order to analyse the results. The weekly net heating demand is plotted in Figure 10. Clearly during the heating season the back-up was more active as was also seen in the simulations (Figure 7). But in this test case the back-up share was larger than in the simulated case. In week 16-22 March the back-up was even used to cover more than 50% of the heating demand. Also in the following weeks the back-up share is close to 50%.

Same as in the simulated case the back-up is still necessary in summer for thermal disinfection purposes. In this summer period weekly heating demand (for DHW purposes) is quite constant as was assumed in the simulations. In the test dwelling an increase in DHW demand occurred from week 21-27 July on. Probably because of a higher setpoint for DHW.

On Figure 11 the net heating demand is plotted as a function of the outdoor temperature. In contrast to Figure 8 the back-up has a larger share in the total energy use in the zone from 0 to 10°C. This was also visible when comparing Figure 10 with Figure 7. From the data collected in Table 2 the back-up share was calculated. The back-up was used to cover 33% of the total heating demand where in the simulated apartment this was only 11%.

Assuming that the application of mixed air leads to an increase of the COP with 0.4 the equivalent heat recovery effectiveness of this installation could be calculated according to formula (1). Based on the measurements results Φ_{HP} = 1100 kWh during the heating season (29 September - 3 May). The mean outdoor temperature during the heating season was 7,4°C. Assuming the average extract air rate is 100 m³/h, the ventilation losses heat losses are 2400 kWh resulting in ε = 18% which is the same as in the simulated case.

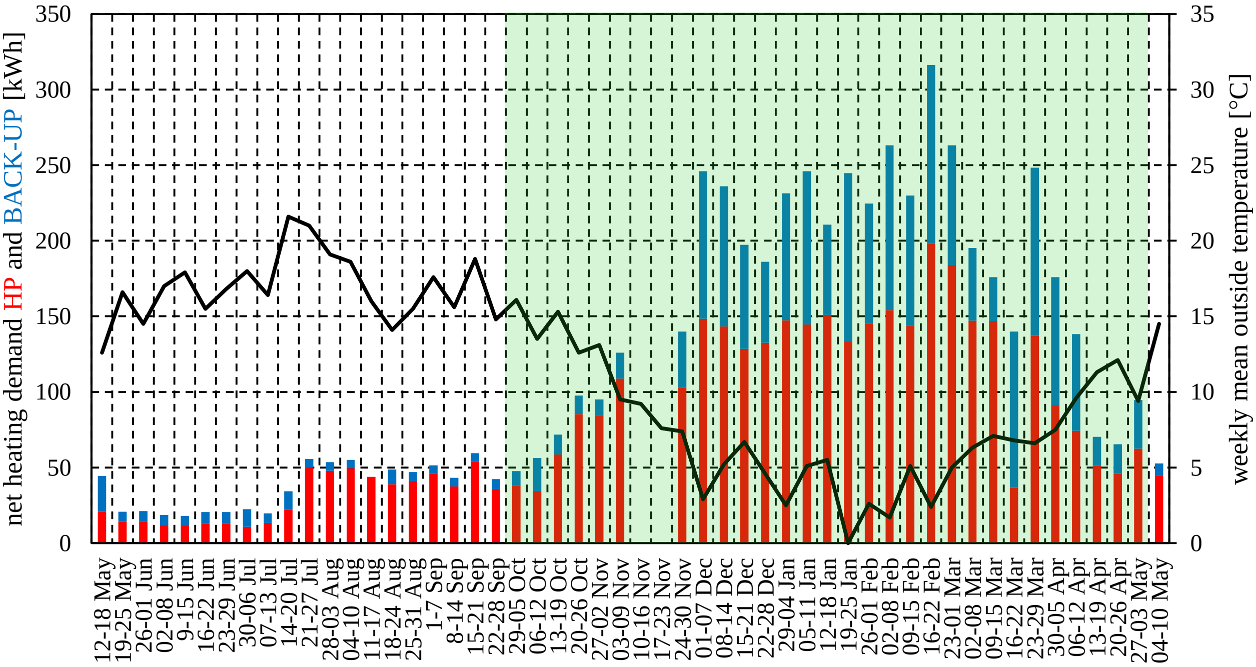


Figure 10: Measured weekly net heating demand of the heat pump (red) and the gas-boiler (back-up, bleu). Typical Belgian heating season marked in green.

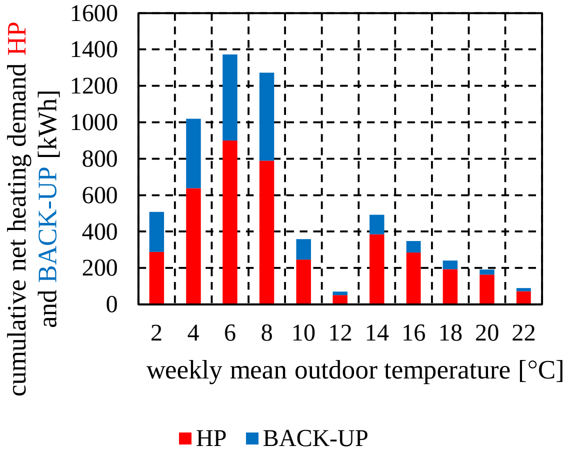


Figure 11: Cumulative net heating demand heat pump (HP) and gas back-up for DHW and SH from 12 May 2014 till 10 May 2015.

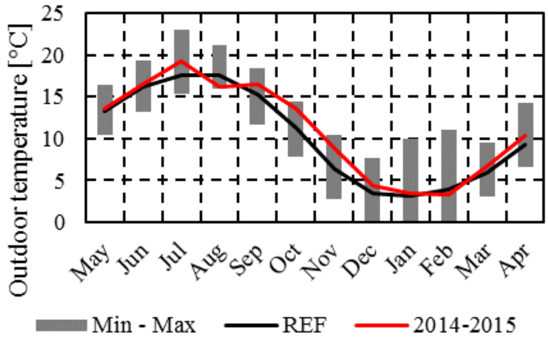


Figure 12: Belgian climate. Reference mean outdoor temperature (black) vs. period 2014-2015 (red) vs. spread in mean temperature (grey).

Table 2: Results of the dynamic simulations for the apartment case and the measurements in the test dwelling for DCMEV hybrid HP variant.

	Case 1: Apartment Dyn. Sim.	Case 2: Test dwelling Meas. 2014-2015
Final energy consumption [kWh]		
SH - HP	1150	---
SH - back-up	508	---
DHW - HP	507	---
DHW - back-up	349	---
Total HP	1657	1309
Total back-up	857	2528
Net heating demand [kWh]		
SH - HP	3771	---
SH - back-up	411	---
DHW - HP	1314	---
DHW - back-up	242	---
Total HP	5086	3894
Total back-up	653	1934
TOTAL	5739	5828

6. CONCLUSIONS

Using mixed air instead of outdoor air as heat pump source leads to an augmentation of the SCOP by 0.4 for SH and 0.3 for DHW. Calculated equivalent heat recovery effectiveness for the DCMEV hybrid HP variant was 18% in both the simulated case and the test dwelling which is lower than traditional efficiencies of MVHR systems. But in contrast to these systems heat recovery occurs during the whole year and not only during the heating season. Moreover the coupling of ventilation extract air and heat pumps makes sure the dwelling is partly heated by means of renewable energy. In case of dwellings with a higher heating demand, more of the ventilation heat losses can be recovered, since the operating period of the heat pump for SH increases.

The back-up is used to cover 11% of the total heating demand in the dynamic simulations whereas in the test dwelling this is 36%. To be able to understand the cause of these differences further research will focus on more real life performances of the DCMEV hybrid HP variant. Detailed logging (ventilation rate, temperatures, energy, COP, …) will be available for further investigation.

7. REFERENCES

Fracastoro, G. V. & Serraino, M., 2010. Energy analyses of buildings equipped with exhaust air heat pumps (EAHP). *Energy and Buildings,* Volume 42, pp. 1283-1289.

Hogeling, J., 2015. From product to system and building performance declaration and certification. *REHVA Journal,* March.p. 5.

Laverge, J., 2013. *Design strategies for residential ventilation systems,* PhD: Universiteit Gent.

Logue, J. M., Sherman, M. H., Walker, I. S. & Singer, B. C., 2013. Energy impacts of envelope tightening and mechanical ventilation for U.S. residential sector. *Energy and Buildings*, Issue 65, pp. 281-291.

Madonna, F. & Bazzocchi, F., 2013. Annual performance of reversible air-to-water heat pumps in small residential buildings. *Energy and Buildings*, Issue 65, pp. 299-309.

SIMPLIFIED METHODS FOR COMBINING NATURAL AND MECHANICAL VENTILATION

Nolwenn Hurel[*1,2], Max Sherman[2], and Iain Walker[2]

1 LOCIE - Université de Savoie,
Savoie Technolac
73376 Le Bourget-Du-Lac, France
**Corresponding author: nolwenn.hurel@gmail.com*

2 Lawrence Berkeley National Laboratory
1 Cyclotron Rd MS 90-3074
Berkeley CA 94720, USA

ABSTRACT

In determining ventilation rates, it is often necessary to combine naturally-driven ventilation, such as infiltration, with mechanical systems. Modern calculation methods are sufficiently powerful that this can be done from first principles with time varying flows, but for some purposes simplified methods of combining the mechanical and natural ventilation are required—we call this "superposition". An example of superposition would be ventilation standards that may pre-calculate some quantities within the body of the standard. When there are balanced mechanical systems, the solution is simple additivity because a balanced system does not impact the internal pressure of the space. Unbalanced systems, however, change internal pressures and therefore can impact natural ventilation in such a way as to make it sub-additive. Several sub-additive superposition models are found in the literature. This paper presentments the results of millions of hours of simulations of the physically correct solution, which span a broad range of weather, leakage and structural conditions. This wide range of data allows for the comparison of three superposition models from the literature and two new ones. The results showed that superposition errors can be reduced significantly by using the appropriate model(s), from the 20% over-prediction from simple linear addition to 1%, or less.

KEYWORDS

Unbalanced ventilation, infiltration, REGCAP simulation, empirical models, superposition

1 INTRODUCTION

Most homes are ventilated by the form of natural ventilation known as "infiltration", which is defined in ASHRAE Standard 62.2 (ASHRAE 2013) as the "uncontrolled inward leakage of air through cracks and interstices in any building element and around windows and doors of a building". In order to decrease energy consumption, house envelopes are getting tighter. Combined with potential increases in pollutant sources in indoor living environments, this raises concerns for indoor air quality (IAQ). Since people spend an average of 90% of their time inside, more houses are using a mechanical ventilation system to maintain a good air quality.

Infiltration is caused by two driving forces, namely the wind and stack effects. The wind raises the pressure on the windward side of the building, and lowers the pressure on the other sides in proportion to the square of wind speed. The stack effect is due to density differences between indoor and outdoor air. In winter the heated air inside the building is less dense than the cold air outside resulting in pressure differences across the envelope with higher inside pressure at the top of the building and lower inside pressure at the bottom of the building. The reverse happens in summer when the outside temperature is greater than the inside.

If a balanced ventilation system is installed, the impact on the infiltration will not be significant because the balanced system does not change the pressures across the building leaks. As a result the total ventilation rate (Q_t) is simply the addition of the fan flow (Q_f) and the natural infiltration (Q_{inf}).

Unbalanced mechanical ventilation systems modify the indoor pressure of the building, which is interacting with the wind and stack induced flows, making the combination of the flows sub-additive. Exhaust fans depressurize the building which increases the air flow in through the building envelope. The greater the fan flow, the higher proportion of the building envelope experiences inflow. The opposite effect occurs with supply-only systems.

In order to avoid both excessive energy consumption and poor IAQ, it is necessary to predict the total flow rate resulting from the combined natural and mechanical ventilation. This can be done using mass flow balance physical and mathematical models to find the internal pressure that balances the incoming and outgoing mass flows. This approach is powerful but requires many computational inputs and can be too time consuming for some purposes such as ventilation standards or simplified parametric modelling. An alternative is to use a simple empirical model for estimating the total ventilation rate Q_t from Q_f and Q_{inf}. These models are generically called "superposition" models. A few models were suggested and tested a few decades ago but the results are sometimes contradictory and there is no consensus on the best one to be used.

In this study we used the REGCAP air flow mass balance model to simulate millions of hours of the physically correct solution, with a broad range of weather, leakage and structural conditions. Then we compared this data with three superposition models from the literature as well as two new empirical models based on the simulation results. The objective was to determine the uncertainty of existing models and to develop improved models that retain the ideal of simplicity.

2 BACKGROUND

2.1 Previous work on superposition

In the eighties and early nineties a number of models for empirically combining the natural infiltration flow and unbalanced mechanical ventilation were suggested. A summary is presented in the Appendix. However, many of these were optimized for limited situations, such as the Palmiter and Bond (Palmiter & Bond, 1991) method, referred here as the half-fan model, which was developed for stack only natural infiltration.

Li (Li, 1990) tested ten models by comparing them with a flow model over a range of wind speeds (0 to 8 m/s) and temperature differences (-20 to 20°C) with open and closed exterior doors and two different exhaust fan speeds. His conclusion was that the quadrature combination of natural and mechanical ventilation worked best. This result is in agreement with the earlier work of Modera and Peterson (Modera & Peterson, 1985) who also used a mass balance ventilation model.

Field tests with tracer gas measurements by Kiel and Wilson (Kiel & Wilson, 1987) and later by Wilson and Walker (Wilson & Walker, 1990) found that for strong exhaust mechanical ventilation (four times the natural rate), simple linear addition was the most acceptable model. Unlike Li, these studies showed large under-predictions using quadrature. This could be due to different building envelope leakage, weather conditions, leakage distributions and strength of mechanical ventilation but it mainly underlines the necessity of new studies.

2.2 REGCAP model

REGCAP is a two zone ventilation model combined with a heat transfer model and a simple moisture transfer model. The two zones are the house and the attic above it and interact through the ceiling. The ventilation rate is found by determining for each zone the internal pressure required to balance the incoming and outgoing mass flows resulting from the natural and mechanical ventilation driving forces.

The model uses an envelope airtightness measurement (ACH_{50}) and a description of the leakage distribution. The leakage for the home is split between walls, floor, ceiling and open flues/chimneys. In this study the leakage distribution was varied with the number of storeys and the type of foundation. Each leak is defined by its flow coefficient, pressure exponent, height above grade, wind shelter and wind pressure coefficient taken from wind tunnel tests. An iterative method is used to solve the non-linear mass balance equations. The attic temperature is not regulated and will therefore both be affected by the ventilation rate and affect the infiltration flow due to the stack effect. In addition, REGCAP includes models for the HVAC equipment in the home and operates on one-minute time steps. The ventilation and heat transfer models are coupled and the combined solution is also found iteratively. A more detailed discussion of REGCAP, including validation compared to measured field data, was done by Walker et al. (Walker, Forest, & Wilson, 2005).

2.3 Applications

Each simplified model can either be used for forward or inverse calculations. The forward model predicts the total ventilation airflow (Q_t) as a function of the natural infiltration (Q_{inf}) and the fan flow (Q_f), whereas the inverse model gives Q_f as a function of Q_t and Q_{inf}. They can be applied to hourly or annual calculations, which results in four different cases:

- Hourly, Forward Case: for the hourly air change rate prediction; useful for estimating energy loads and needed for relative pollutant exposure calculations.
- Annual, Forward Case: predicting the annual effective ventilation given the effective infiltration and a fixed (or effective) fan flow; for indoor air quality (IAQ) purposes.
- Hourly, Inverse Case: when one wants to vary the fan size each hour to compensate for varying hourly infiltration in order to keep the total ventilation constant.
- Annual, Inverse Case: for finding the fixed fan size that will combine with effective infiltration to produce a desired total ventilation; useful for standards such as the 62.2.

3 APPROACH

3.1 REGCAP simulations

We used REGCAP to create a data based on a wide range of weather and housing conditions. The range of inputs is presented in Table 1 and results in 720 combinations. The number of storey changes but the floor area is constant and equals to 1900 ft² (176 m^2). For each set of inputs, we ran the model for the 525 600 minutes of a year and we hourly averaged the outputs, which results in more than 6.3 million of points of comparison for the superposition models.

Table 1: Range of inputs for the REGCAP simulations

Parameters	Values
Envelope airtightness (ACH_{50})	0.6; 3; 5; 7; 10
Mechanical ventilation type	Exhaust ; supply
Number of storey	1; 2; 3
Foundation type	Slab on-grade; crawlspace; basements
Climate zones	Miami; Houston; Memphis; Baltimore; Chicago; Burlington; Duluth; Fairbanks

We calculated the flow through the exhaust or supply fan (Q_f) according to ASHRAE Standard 62.2 which includes the infiltration credit. We used REGCAP to calculate first the infiltration flow through the envelope (Q_{inf}) due to the stack and wind effect, with no mechanical ventilation operating. We repeated the simulations with supply or exhaust fans operating to obtain the total flow (Q_t). Then we compared the results for each superposition method for combining Q_f and Q_{inf} to Q_t.

For the annual calculations, the fan flow is still the same as it is a constant over the year, but Q_{inf} and Q_t are effective annual average infiltration rates. The effective values differ from the averaged ones. As defined in ASHRAE Standard 62.2, they correspond to "the constant air infiltration rate that would result in the same average indoor pollutant concentration over the annual period as actually occurs under varying conditions". This annual approach can be particularly useful when one wants to size the fan to the total ventilation required by ventilation standards.

3.2 Simplified models

The equations describing the five simplified superposition models are presented in Table 2.

The three first models come from the literature described earlier. The **additivity** model, which is a simple addition of the flows, is in the current ASHRAE 62.2 Standard, and has been experimentally verified by Kiel and Wilson. **Simple quadrature** is the current model in the ASHRAE Handbook of Fundamentals, and has been verified by both Modera and Peterson and Li. The **Half Fan** model was used in earlier editions of ASHRAE Handbook of Fundamental and has been established experimentally by Palmiter and Bond. For each of these models the forward and inverse forms are equivalent. For all three models, verification was for a narrow range of conditions and the current study aims to investigate their performance over a much wider range of homes and conditions.

In addition we developed other models for this study in order to reduce the uncertainties associated with the existing models. We used the simulation results to approximate a sub-additivity coefficient (Φ) weighting the infiltration contribution to either the total ventilation (forward) or the fan sizing (inverse):

$$Q_t = Q_f + \Phi_{\mathrm{fw}} Q_{inf} \quad (7)$$

$$Q_f = Q_t - \Phi_{\mathrm{inv}} Q_{inf} \quad (2)$$

Table 2: Forward and inverse equations of the simplified models compared to the REGCAP results

Model	**Forward**	**Inverse**
Additivity	$Q_t = Q_f + Q_{inf}$	$Q_f = Q_t - Q_{inf}$
Simple quadrature	$Q_t = \sqrt{{Q_f}^2 + {Q_{inf}}^2}$	$Q_f = \sqrt{{Q_t}^2 - {Q_{inf}}^2}$
Half-fan	$Q_t = \max\left(Q_f\ , Q_{inf} + \frac{Q_f}{2}\right)$	$Q_f = \min\left(Q_t\ , 2(\mathrm{Q_t} - \mathrm{Q_{inf}})\right)$
Exponential sub-additivity	$Q_t = Q_f + \exp\left(-k_{fw}\frac{Q_f}{Q_{inf}}\right) Q_{inf}$	$Q_f = \mathrm{Q_t} - \exp\left(-k_{inv}\left(\frac{Q_t}{Q_{inf}} - 1\right)\right) Q_{inf}$
Simple inverse sub-additivity (SISA)	$Q_t = \frac{Q_f}{2} + \sqrt{\frac{{Q_f}^2}{4} + {Q_{inf}}^2}$	$Q_f = \mathrm{Q_t} - \frac{{Q_{inf}}^2}{Q_t}$

Table 3: Values of the k coefficient for the exponential model

	Forward	**Inverse**
Hourly	$k_{fw,h} = \frac{2}{3}$	$k_{inv,h} = 1$
Annual	$k_{fw,a} = \frac{4}{9}$	$k_{inv,a} = \frac{2}{3}$

We empirically developed an **exponential** form for Φ that worked well to reduce errors. The resulting forward and inverse models are not equivalent but have the same limits and trends. We optimized the coefficients k_{fw} and k_{inv} to best approximate the simulation results and we found different values for the annual and hourly data, as shown in Table 3.
We developed a fifth model, SISA, in order to avoid the complications of the exponential function. For the SISA model, Φ is the ratio of Q_{inf} to Q_t for the inverse approach. This ratio is referred to as α, as defined in Equation 3. α is a useful normalization to use for all the models when examining the trends and limits of the modelling errors. The forward version of SISA uses a semi-quadratic formulation:

$$\Phi_{\mathrm{inv}} = \frac{Q_{inf}}{Q_t} = \alpha \quad (3)$$

4 RESULTS

We evaluated the models by comparing the air flow prediction to the one obtained with the simulation. The forward model aims at predicting the total airflow so the error, E_{fw}, is given by:

$$E_{fw} = \frac{Q_{t,model} - Q_{t,sim}}{Q_{t,sim}} \quad (4)$$

In the same way, the error for the inverse model, E_{inv}, is the difference between the predicted and simulated fan flows. It is still divided by the total airflow since a division by a fan flow close to zero would give a significant error but the impact on the actual ventilation rates would be very small.

$$E_{inv} = \frac{Q_{f,model} - Q_{f,sim}}{Q_{t,sim}} \quad (5)$$

4.1 Hourly results

For the hourly data, the high number of points (over six million) requires the use of summary statistics, represented by box-and-whisker plots. We sorted the data into 20 bins by infiltration fraction (α) and each bin is represented by a box. The box widths are proportional to the square-root of the number of observations in the bin. The bottom and top of the box are the first and third quartiles, and the black band inside is the median. The ends of the whiskers represent the minimum and maximum of the data. In our case a box represents in average more than 300 000 points which explains why these values can be quite far from the median. When several parameters are plotted, each of them is identified by a color and the horizontal offset in αbetween them is only for the sake of clarity.

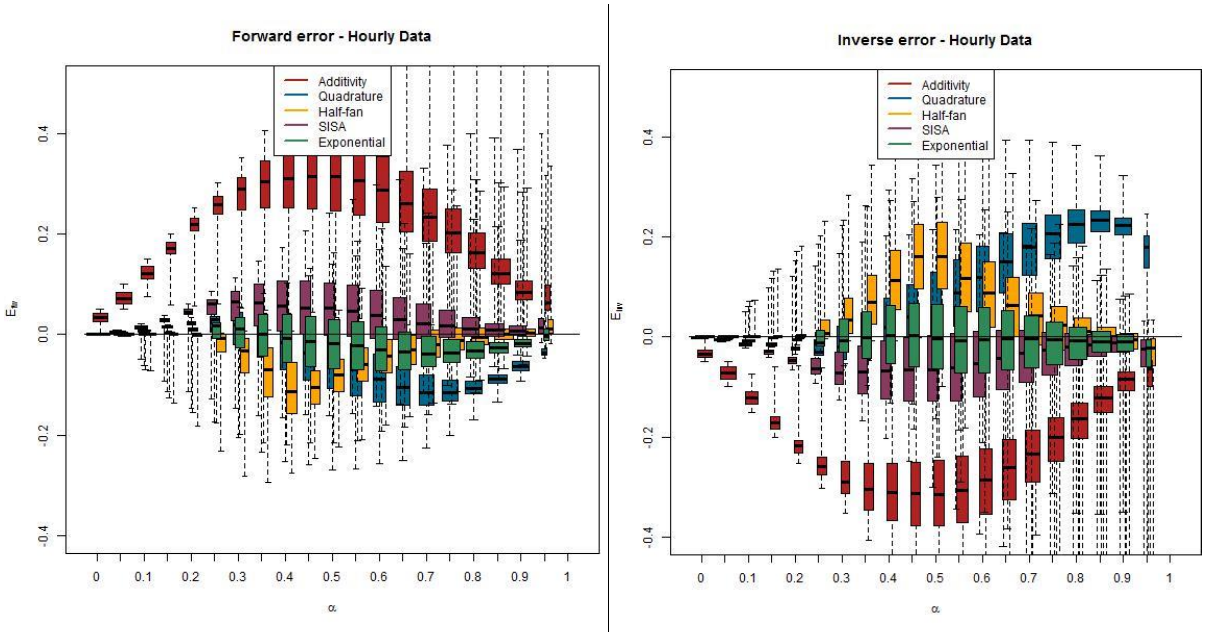


Figure 36: Forward and inverse hourly errors for every model

As shown in Figure 1, an over-prediction of the total ventilation (E_{fw}>0) results in an under-prediction of the fan flow with the equivalent inverse model (E_{inv}<0). We don't observe this for the exponential model since the forward and inverse models are not equivalent. For the additivity model the two errors have simply opposing values, but there is no such symmetry for the other models. The inverse error for the quadrature model reaches higher values than the forward error for high infiltration fractions. In the same way the half-fan model gives a higher peak in the inverse error than the forward one.

4.2 Annual results

For the annual analysis, there is a single result for each of the parameter combinations in Table 1. This reduced number of points (720) allows all the individual results to be shown. Compared to the hourly data, there are less extreme values and no point with α above 0.9. We can observe a gap around α = 0.15, which can be explained by the lack of an intermediate value between 0.6 ACH and 3 ACH in the airtightness levels of the simulated houses.

As shown in Figure 2, the trends are similar to those of the hourly errors. However since they are effective values, Q_t and Q_{inf} are smaller than a simple annual average, unlike Q_f that is constant over the year. As a result we can observe smaller over-predictions but greater under-predictions for the forward models, and the opposite for the inverse models.

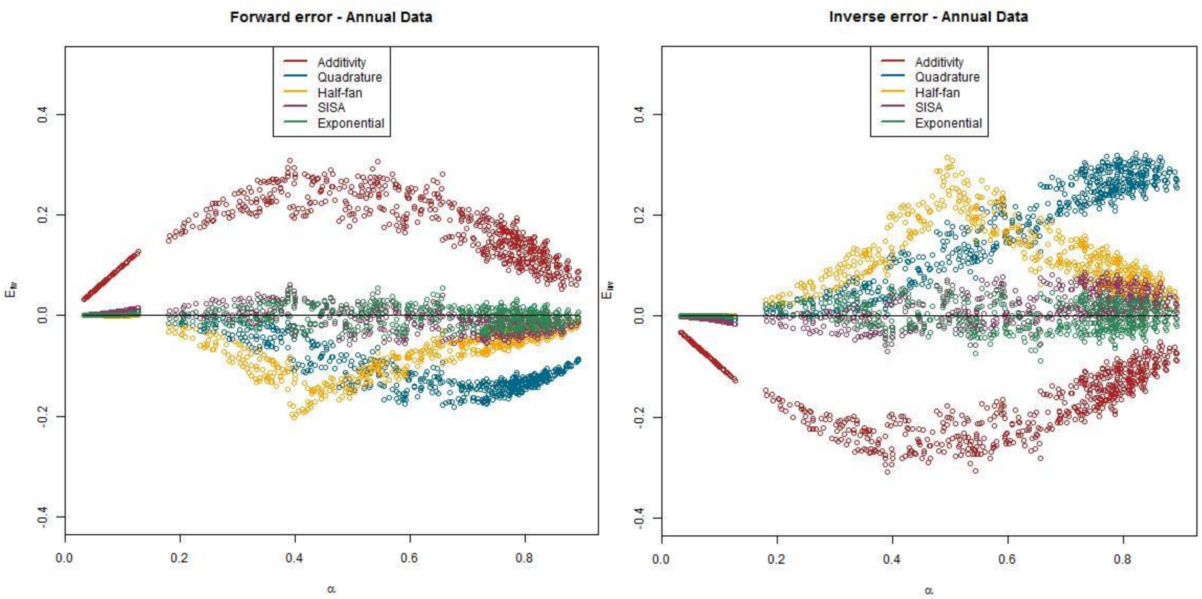


Figure 2: Forward and inverse annual errors for every model

4.3 Discussion

The characteristics of the hourly and annual errors are presented respectively in Tables 4 and 5. The various simulations give results covering a wide range of infiltration ratio (α) but not evenly dispersed, with for example fewer points around α=0.15. In order to compensate for this disparity, we calculated the bias and RMS of the errors for 20 bins of α values, and then we equally weighted them. The bias is the error over the full range of house and weather parameters exercised in this study. The RMS is representative of the error for an individual home and is most useful for most applications – such as sizing fans for an individual home to meet a ventilation standard, such as ASHRAE 62.2. Because of the high number of points, the maximum error is not meaningful for the hourly error. We use instead the maximum median among the 20 groups of data, and the maximum of 90% of the data.

The exponential models always give the best predictions with biases around or below 1%, RMS ranging from 1.5% to 5.5% and maximums around or under 10%. The additivity model is always the worst, with bias and RMS errors of around 20% and maximums above 30%. The quadrature and the half-fan are much better than the additivity, especially for the forward models applied to the hourly data, but with a significant difference compared to the exponential models.

The SISA model is the second best model, and is almost as good as the exponential for the annual data with biases below 1%, RMS around 2% and maximums at 6% and 8.7%. There is no reason to prefer this model to the exponential one for the forward prediction but it has a simpler expression for the inverse prediction. It also has the advantage of not having Q_t as a denominator, which, unlike Q_{inf}, can never equal to zero and may therefore be a good option for calculations determining fan size requirements to meet total ventilation rates.

Table 4: Error on the model predictions for the hourly data

Model	Forward error				Inverse error			
	Bias	RMS	Max. Med.	Max. 90%	Bias	RMS	Max. Med.	Max. 90%
Additivity	20.9%	21.8%	31.5%	34.1%	-20.9%	21.8%	31.5%	34.1%
Quadrature	-3.95%	6.65%	11.6%	13.1%	7.99%	11.0%	23.3%	25.2%
Half-fan	-2.85%	4.78%	11.3%	9.22%	4.20%	7.65%	16.1%	15.8%
Exponential	-1.15%	4.01%	3.78%	8.03%	-0.61%	5.57%	2.31%	11.3%
SISA	3.29%	5.39%	6.57%	9.43%	-4.12%	7.02%	7.21%	12.7%

Table 5: Error on the model predictions for the annual data

Model	Forward error			Inverse error		
	Bias	RMS	Max.	Bias	RMS	Max.
Additivity	17.4%	17.5%	30.9%	-17.4%	17.5%	30.9%
Quadrature	-7.51%	7.82%	18.1%	11.72%	12.1%	32.2%
Half-fan	-6.43%	6.58%	20.3%	9.86%	10.1%	31.6%
Exponential	-0.15%	1.57%	5.55%	0.18%	2.22%	8.85%
SISA	0.32%	1.95%	6.18%	0.68%	2.60%	8.65%

Each of the models have the same physical limits with Q_t equals to Q_f when α tends towards 0 (no infiltration) and Q_t equals to Q_{inf} when α tends towards 1 (no mechanical ventilation). That is the reason why the errors tend to 0 at the extreme values of α. One can notice that the additivity and the half-fan models have their maximum errors for α close to 0.5 whereas the quadrature model has its maximum error around 0.7 for the forward calculation and 0.8 for the inverse one. It means that depending on the airtightness of the building, the ranking of the best models from the literature is different, and could be one of the reasons why the previous studies did not agree on which model to recommend.

5 CONCLUSIONS

A superposition model with an exponential sub-additivity coefficient gives very satisfying results, always better than the models used in the literature. It takes different forms for the forward and inverse calculations, and has different optimized coefficient for the hourly and annual ones:

Table 6: Exponential sub-additivity model

	Forward	Inverse
Hourly	$Q_t = Q_f + \exp\left(-\frac{2}{3}\frac{Q_f}{Q_{inf}}\right) Q_{inf}$	$Q_f = \mathrm{Q_t} - \exp\left(-\left(\frac{Q_t}{Q_{inf}} - 1\right)\right) Q_{inf}$
Annual	$Q_t = Q_f + \exp\left(-\frac{4}{9}\frac{Q_f}{Q_{inf}}\right) Q_{inf}$	$Q_f = \mathrm{Q_t} - \exp\left(-\frac{2}{3}\left(\frac{Q_t}{Q_{inf}} - 1\right)\right) Q_{inf}$

In the case of inverse annual calculations, the SISA model is almost as good as the exponential one. Since it has a simpler expression, it is a good alternative and could be used to improve ventilation standards such as the ASHRAE 62.2.

6 ACKNOWLEDGEMENTS

This work is financially supported by the French National Research Agency (ANR) through its Sustainable Cities and Buildings programme (MOBAIR project n°ANR-12-VBDU-0009); by the Région Rhône-Alpes; by the U.S. Dept. of Energy Building Technologies Program, Office of Energy Efficiency and Renewable Energy under DOE Contract No. DE-AC02-05CH11231; by the U.S. Dept. of Housing and Urban Development Office of Healthy Homes and Lead Hazard Control through Interagency Agreement I-PHI-01070; by the U.S. Environmental Protection Agency Office of Air and Radiation through Interagency Agreement DW-89-92322201-0 and by the California Energy Commission through Contract 500-08-061.

7 REFERENCES

ASHRAE Handbook of Fundamentals, ASHRAE. (2013). Atlanta, GA

ASHRAE Standard 62.2-2013, Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings. (2013). Atlanta, GA

Kiel, D., & Wilson, D. (1987). Influence of natural infiltration on total building ventilation dominated by strong fan exhaust. ASHRAE Transactions vol. 93 part 2.

Levins, W. P. (1982). Measured Effect of Forced Ventilation on House Infiltration Rate. ASHRAE/DOE Thermal Performance of the Exterior Envelopes of Buildings II Conference. Las Vegas.

Li, Y. (1990). Simplified Method of Combining Natural and Exhaust Ventilation. Climate and Buildings vol.2, 29-35.

Modera, M., & Peterson, F. (1985). Simplified Methods for Combining Mechanical Ventilation and Natural Infiltration. Lawrence Berkeley National Laobratory report no. LBL-18955.

Palmiter, L., & Bond, T. (1991). Interaction of mechanical systems and natural infiltration. 12th AIVC Conference - Air Movement and Ventilation Control Within Buildings, (pp. 285-295). Ottawa, Canada.

Shaw, C. Y. (1985). Methods for Estimating Air Change Rates and Sizing Mechanical Ventilation Systems for Houses. Division of Building Research, National Research Council of Canada, Building Research Note 237.

Walker, I. S., Forest, T. W., & Wilson, D. J. (2005). An Attic-Interior Infiltration and Interzone Transport Model of a House. Building and Environment, 701-718.

Wilson, D., & Walker, I. S. (1990). Combining Air Infiltration and Exhaust Ventilation. IndoorAir '90, (pp. 467-472). Toronto, Canada.

Appendix: Summary of previous superposition models and the results of the simulation/experimental comparison studies carried out on them

Name/Ref	Model	Range	Comparison		
			Ref.	Sim/Exp	Results
Additivity	$Q_t = Q_f + Q_{inf}$	All	Kiel & Wilson	Exp.	best agreement
			Wilson & Walker	Exp.	overpredicts Qt by 7%
			Li	Sim.	average error: 33%; maximum error: 64%
Quadrature	$Q_t = \sqrt{Q_f{}^2 + Q_{inf}{}^2}$	All	Modera & Peterson	Sim.	good agreement: error on Qt < 10%
			Kiel & Wilson	Exp.	underpredicts Qt by 15-30%
			Wilson & Walker	Exp.	underpredicts Qt by 20%
			Li	Sim.	good agreement: average error: 5%; maximum error: 17%
			Palmiter & Bond	Exp.	underpredicts the fan efficiency for Qinf<Qf; overpredicts the fan efficiency for Qinf>Qf
Levins (Levins, 1982)	$Q_t = Q_{inf} + Q_f.\exp\left(-\frac{Q_{inf}}{Q_f}\right)$	All	Kiel & Wilson	Exp.	underpredicts Qt by 15-30%
			Li	Sim.	good agreement: average error: 5%; maximum error: 20%
	$Q_t = \left(Q_f{}^{\frac{1}{n}} + Q_{inf}{}^{\frac{1}{n}}\right)^n$	All	Modera & Peterson	Sim.	bigger errors on Qt than the quadrature model
			Kiel & Wilson	Exp.	underpredicts Qt by 10-25%
			Li	Sim.	average error: 11%; maximum error: 30%
Shaw (Shaw, 1985)	$Q_t = \begin{cases} Q_f & \text{for } h_0 > H \\ F\left(Q_{w-f}{}^{\frac{1}{n}} + Q_w{}^{\frac{1}{n}}\right)^n & \text{for } h_0 < H \end{cases}$		Shaw	Exp.	in general within 25% of the measured values
			Kiel & Wilson	Exp.	underpredicts Qt by 15-30%
	$Q_t = \sqrt{Q_f{}^2 + (2Q_{inf})^2}$	$Q_f \gg Q_{inf}$	Kiel & Wilson	Exp.	very spread data; overpredicts Qt when Qf<0.7Qt ; mostly underpredicts Qt when Qf>0.7Qt
			Li	Sim.	average error: 56%; maximum error: 100%
Li	$Q_t = \left(Q_f{}^{\frac{1}{n}} + (2Q_{inf})^{\frac{1}{n}}\right)^n$	$Q_f \gg Q_{inf}$	Li	Sim.	average error: 98%; maximum error: 160%
Kiel & Wilson	$Q_t = \sqrt{\left(\frac{Q_f}{2}\right)^2 + Q_{inf}{}^2} + \frac{Q_f}{2}$	All (Exhaust fan)	Kiel & Wilson	Exp.	underpredicts Qt by 10-30%
			Li	Sim.	average error: 12%; maximum error: 35%
			Palmiter & Bond	Exp.	overpredicts the fan efficiency
Wilson & Walker	$Q_t = \left(\left(\frac{Q_f}{2}\right)^{\frac{1}{n}} + Q_{inf}{}^{\frac{1}{n}}\right)^n + \frac{Q_f}{2}$	All (Exhaust fan)	Wilson & Walker	Exp.	underpredicts Qt by 7%
			Li	Sim.	average error: 18%; maximum error: 42%
Li	$Q_t = \frac{1}{2}\sqrt{Q_f{}^2 + (2Q_{inf})^2}$	$Q_f < Q_{inf}$	Li	Sim.	average error: 22%; maximum error: 50%
Li	$Q_t = \frac{1}{2}\left(Q_f{}^{\frac{1}{n}} + (2Q_{inf})^{\frac{1}{n}}\right)^n$	$Q_f < Q_{inf}$	Li	Sim.	average error: 21%; maximum error: 50%
Half-fan - Palmiter & Bond	$Q_t = \begin{cases} \frac{Q_f}{2} + Q_{inf} & \text{for } Q_f < 2Q_{inf} \\ Q_f & \text{for } Q_f \geq 2Q_{inf} \end{cases}$		Palmiter & Bond	Exp.	good agreement

DEVELOPMENT OF A SEASONAL SMART VENTILATION CONTROLLER TO REDUCE INDOOR HUMIDITY IN HOT-HUMID CLIMATE HOMES

Sara Ticci[*1,2], Brennan Less[2], Iain Walker[2] , and Max Sherman[2]

1 University of Florence,
Department of Civil and Environmental Engineering
50139 Florence, Italy
**Corresponding author: sara.ticci@dicea.unifi.it*

2 Lawrence Berkeley National Laboratory
1 Cyclotron Rd MS 90-3074
Berkeley CA 94720, USA

ABSTRACT

Controlling indoor humidity is important in homes, because high indoor humidity is associated with occupant health and building durability issues. Ventilation is often used to avoid peaks of moisture in homes, such as in kitchens and bathrooms. However, in hot-humid climates, outdoor air can have higher humidity than indoors, and continuous whole house ventilation can lead to increases in indoor humidity levels. This problem is exacerbated in high performance homes, because their efficient building envelopes limit the operation of the cooling system, which also provides incidental dehumidification. This paper analyzes a time-shifting approach to smart ventilation control that takes advantage of changes in outdoor and indoor humidity, essentially venting more when dryer outside and less when more humid outside. The changes in whole house airflow are controlled in such a way that exposure to air pollutants is equivalent to a continuously operating, fixed airflow system. Specifically, this paper presents the development and initial testing of a seasonal smart ventilation controller, based simply on the month of the year and the net-humidity balance for the month between inside and outside. We assessed high performance test homes with varying floor areas and moisture generation rates across a variety of hot- and mixed-humid climates in the south-eastern U.S., using ventilation simulation software specially adapted for this analysis. We present the results from baseline simulations, with continuous fans sized at 0%, 50% and 100% of the ASHRAE 62.2-2013 fan flow rates. From this analysis, we develop a seasonal control strategy. Finally, we present preliminary results from use of the proposed seasonal controller. The results include comparisons of IAQ, energy use and indoor humidity levels.

KEYWORDS

Humidity, ventilation, IAQ, homes, hot-humid climates, high performance homes, smart ventilation, controls

1 INTRODUCTION

Elevated indoor humidity levels in homes represent a risk to occupant thermal comfort and health, as well as building durability. Indoor relative humidity is commonly controlled between 40% and 60% for comfort, health and building durability reasons. In most homes, the indoor humidity is kept within this range by high levels of natural air exchange and by operation of the central cooling system and its associated moisture removal. High performance homes are more efficient from the thermal point of view, and their cooling system operates less, which reduces the associated moisture removal. Ventilation is often cited as a secondary contributor to high indoor relative humidity in high performance homes in hot humid climates, and questions have been asked about lowering mechanical ventilation rates. Several studies have assessed the costs and effectiveness of strategies to reduce indoor humidity levels in high performance, humid climate homes (Kerrigan & Norton, 2014; Moyer, Chasar, Hoak, & Chandra, 2004; Rudd, 2013b; Rudd & Henderson, Jr., 2007; Rudd et al., 2005). The main goal of these efforts was to reduce the number of hours above 60% RH to an unspecified, “acceptable” level. Strategies have included dehumifiers, energy recovery ventilators (ERV) and enhanced cooling system control strategies.

This paper investigates the use of smart ventilation controls to reduce the impact of ventilation on indoor moisture in high performance, low sensible load homes in humid climates. Our smart ventilation controller is designed to reduce the number of hours of high indoor humidity, without worsening occupant exposure to indoor pollutants. To do this, we introduce the concept of *relative exposure*, used to quantify indoor air quality (IAQ) (Sherman, Mortensen, & Walker, 2011; Sherman, Walker, & Logue, 2012). Relative exposure is represented as a fractional value, comparing the pollutant concentration in the control case against a base case continuously vented to the ASHRAE 62.2-2013 Total Ventilation Rate (ANSI/ASHRAE, 2013). If the annual average relative exposure is equal to one, the occupants have received the equivalent exposure to indoor contaminants. An annual average relative exposure below one means exposure is lower than the standard, and values greater than one indicate exposure is higher than the standard. Our smart controller uses this concept to ensure equivalent annual exposure while time-shifting ventilation to periods that are advantageous from a humidity perspective.

2 APPROACH

To assess the humidity impacts of mechanical ventilation and of smart controls, we used the REGCAP simulation software to simulate high performance homes in hot humid U.S. climates. REGCAP is a physics-based model with mass, moisture and heat balance modules, simulated on a one-minute time-step in C++. The REGCAP model is described in detail in Appendix 1 of (Walker & Sherman, 2006). The REGCAP model was first used to run baseline simulations, with continuous fans sized to ASHRAE 62.2-2013. Analysis of these baseline simulations led to the development and testing of the seasonal smart control strategy.

2.1 Test house and parameters of interest

In our simulations, we varied the climate zones, home size, fan size, internal moisture generation rates and sensible heat gains for a total of 162 combinations (see Table 1). A variety of locations were chosen in hot- and mixed-humid climate zones to assess the effectiveness of humidity control by smart ventilation control. These locations were chosen because past simulations have shown them to have high indoor humidity (Martin, 2014), or they were the representative cities in humid U.S. climate zones 1A-4A. Three one-story house geometries were chosen with varying conditioned floor areas (Small-Size: $100m^2$, Medium-Size: $200m^2$, Large-Size: $300m^2$). The building envelopes and equipment performance specifications are based on the requirements[1] of the U.S. DOE Zero Energy Ready home (U.S. Department of Energy, 2013).

[1]DOE Zero Energy Ready Home air-tightness requirements: Climate-Zones1-2:ACH_{50}=3, Climate-Zones3-4:ACH_{50}=2.5.

All test cases are representative of very high performance, efficient homes, with airtight envelopes (2.5-3 ACH_{50}), insulation levels compliant with the International Energy Conservation Code 2012, efficient windows, HVAC ducts located in conditioned space, etc.

Three moisture generation rates were used to represent low, medium and high occupancy homes, with 3, 6.5, 11.8 kg/day, respectively. Sensible internal heat gains also varied with occupancy and were calculated using the formula for the reference home in the Home Energy Rating System (HERS) Standards (RESNET, 2006) Table 303.4.1(3). Values ranged from 180 to 928 watts. We assumed that the moisture and sensible loads are generated evenly throughout the day.

The whole house ventilation fan was sized to meet ASHRAE 62.2-2013 requirements, which varies fan flow with floor area and occupancy. An infiltration credit was deducted from the total required airflow, depending on the airtightness and climate zone using the infiltration credit calculations in the standard. One type of common mechanical ventilation system was simulated, the central fan integrated supply (CFIS). CFIS is a duct from outside to the return of the central air handler, and when the central fan operates, outside air is introduced. The CFIS system has the potential to provide the most efficient humidity removal, because the air flowing over the cooling coil including the air from outside can be at a higher humidity than if the outdoor air first mixed with indoor air and then entered the cooling system. We took into account three fan sizes: the value 0 to indicate no continuous ventilation, 0.5 to represent the 50% of the rate required by ASHRAE 62.2-2013 and the value 1 to accomplish the 100% of the required rate. Local exhausts simulated in REGCAP included bathroom and kitchen fans, as well as vented clothes dryer.

Table 1: Synthesis of all the parameters for the simulations.

CLIMATE ZONES	Miami, FL (1A)	72 HDD	
	Orlando, FL (2A)	302 HDD	
	Houston, TX (2A)	311 HDD	
	Charleston, SC (3A)	1049 HDD	
	Memphis, TN (3A)	1631 HDD	
	Baltimore, MD (4A)	2537 HDD	
HOME SIZE	LARGE (300sqm)	MEDIUM (200sqm)	SMALL (100sqm)
FAN SIZE	0	0.5	1
MOISTURE GENERATION RATES	HIGH (11.8 kg/day)	MEDIUM (6.5 kg/day)	LOW (3 kg/day)
VENTILATION TYPE	CFIS		

2.2 Outdoor Humidity

In order to understand the potential for smart ventilation strategies to mitigate moisture issues in high performance homes, it is first essential to understand patterns in outdoor humidity, as represented in Typical Meteorological Year (TMY3) weather data files. We assessed these patterns by performing seasonal decomposition on the outdoor humidity data, breaking the variability down into three components: the monthly, the daily and the hourly trends. As pictured in Figure 1, the variability in outdoor humidity values decreases going from monthly, to daily and hourly periods. The monthly variation in outdoor humidity is fairly predictable across locations, with higher values of outdoor humidity during summer months and lower values during the rest of the year. Daily patterns are not predictable, because they are not driven by diurnal or annual seasonal patterns. Control strategies that take advantage of daily humidity variability would have to be sensor-based. Hourly patterns are diurnally driven and therefore more predictable, but variation within hours of a day is small, so the value of hourly control is limited. From these considerations, control based on the month of the year is both possible and will have the most benefit in controlling indoor humidity.

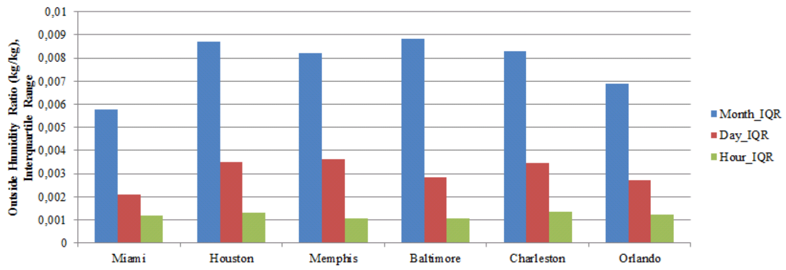


Figure 1: Interquartile ranges of outdoor humidity in select hot-humid cities based on monthly, daily and hourly decomposed trends.

2.3 Initial Simulations to investigate impact of ASHRAE 62.2 ventilation on indoor humidity – Baseline Simulations

2.3.1 Indoor Humidity

For all baseline cases, we evaluated the average indoor relative humidity and the proportion of the year with indoor relative humidity above 60% and 70%, as this is the primary metric used in the literature to assess humidity problems and interventions. Table 2 shows summary statistics for each location aggregated across varying floor areas and occupancy rates. All locations had substantial periods of the year above 60% and 70% RH in all locations. The indoor humidity generally worsens as climates become both hotter and more humid.

Table 2: Proportion of the year with indoor relative humidity above 60% and 70%, values averaged across all parameters (house size, occupancy rate, flow rates)

Climate Zone	Median fraction of the year (%)		Annual Average of Median values of RH (%)
	>60%	>70%	
Miami	87	41	67
Orlando	70	28	65
Houston	64	34	64
Charleston	64	32	66
Memphis	38	13	57
Baltimore	31	14	52

Given that indoor humidity issues vary by climate zone, smart ventilation control strategies may also need to differ based on location. Indeed cities such as Miami and Orlando present yearlong humidity problems, with periods of high humidity spread throughout the year. The highest indoor humidity periods tend to occur in "shoulder seasons", when outdoor humidity is increasing towards its summer peak, but sensible loads are not yet driving cooling system operation and dehumidification. Such shoulder season humidity peaks are evident in the month of February in Figure 2, which shows the annual pattern of indoor humidity in Orlando. Also evident in Figure 2 is that indoor humidity is above 60% RH nearly all months of the year. Conversely, for Baltimore (see Figure 3), issues occur only during the summer period.

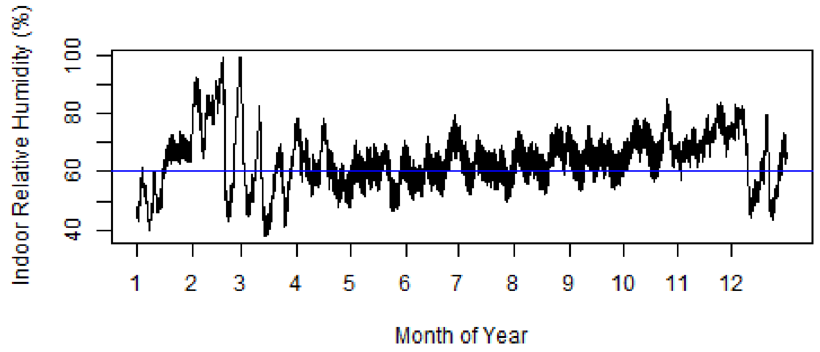


Figure 2: Indoor Humidity trend in Orlando.

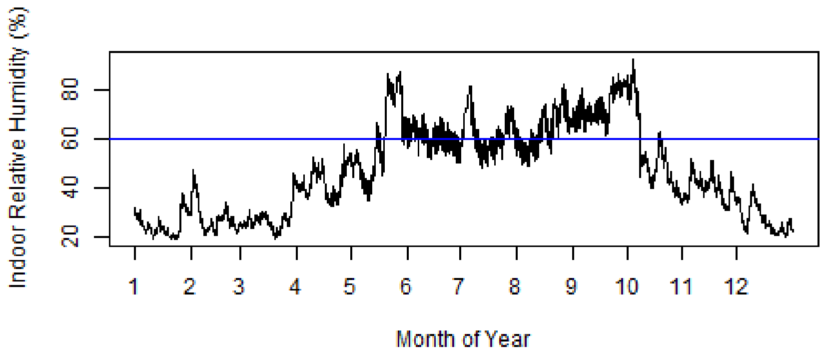


Figure 3: Indoor Humidity trend in Baltimore.

2.3.2 Net-moisture balance and effects of house parameters

From first principles, the simple mass balance for moisture suggests that ventilation will either transport water vapour into or out of the home, depending on the sign of the humidity ratio difference ($w_{house} - w_{outside}$). Positive values lead to moisture removal from the house and negative values lead to moisture transport into the house.

$$\dot{m}_{water} = \dot{m}_{air} \times (w_{house} - w_{outside}) \tag{1}$$

m_{water} = mass flow of moisture, *kg*
m_{air} = mass flow of air, *kg*
w_{house} = humidity ratio of house, *kg/kg*
$w_{outside}$ = humidity ratio outside, *kg/kg*

For each simulated test case, we calculated the humidity ratio difference (HRdiff) between the house and outside for every hour of the year. This HRdiff is shown for an example case in Figure 4. We then averaged these values over different time periods of interest, namely annually and monthly. We refer to this annual average as the net-humidity balance. For each combination of house size, occupancy rate and climate zone, there is an annual net-humidity balance (average of all HRdiff values for the year), which is either positive or negative. Positive means that on average for the year, it is more humid inside than outside, and more ventilation will provide net-moisture removal. Negative means that on average, it is more humid outside than inside, and more ventilation will provide net-humidification. These same principles function on a monthly basis as well, which we explore below in section 2.3.3 in development of our seasonal control algorithm.

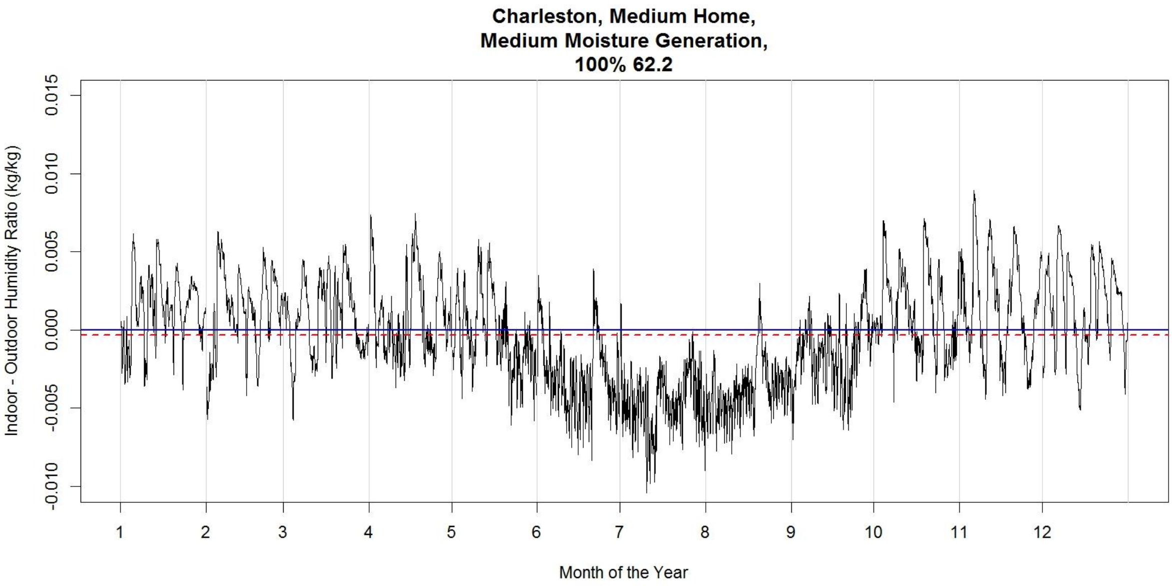


Figure 4: Time series plot of hourly humidity ratio differences (HRdiff) for a medium size Charleston home with medium moisture generation rate, and ventilation sized at 100% of 62.2-2013. Solid blue line represents value of zero, and the dashed red line is the annual average for this case.

We expect that the value of ventilation over the course of the year will vary by climate zone, and that this will depend on the sign and magnitude of the net-humidity balance. It will also vary by home size and moisture generation rates. In order to understand these effects independently, we developed a linear multiple regression model to predict the net-annual humidity balance using three factor variables—climate zone, home size and occupant density (p-value = < 2.2e-16; Adjusted R^2 = 0.9811). The resulting coefficients are provided in Table 3, with a reference intercept value of -0.0017 kg/kg in a large Miami home with high occupant density.

Table 3: Coefficient estimates from multiple linear regression model of annual humidity ratio difference on climate zone, house size and occupant density.

Parameter	Coefficients	Parameter	Coefficients
Miami_HouseLarge_HighDensity	-0.00171	HouseSize_Med	0.000389
Houston	0.000999	HouseSize_Small	0.001225
Orlando	0.001026	OccDensity_Med	-0.00075
Charleston	0.001752	OccDensity_Low	-0.00142
Memphis	0.001947		
Baltimore	0.002667		

From these regression coefficients, it is clear that climate zone, home size and occupant density have substantial impacts on the net-annual humidity balance. Annual net-humidity balances become more positive as we progress from Miami through Baltimore, with positive model coefficients of increasing size as climates become progressively less humid. Similarly, for any given moisture generation rate and climate zone, the smaller sized home will have higher humidity than the larger size home. This is reflected in positive model coefficients for small and medium size homes that get larger as the home gets smaller (Medium = 0.000389 and Small = 0.00123). Similarly, for any home size and climate zone, a higher moisture generation rate (represented by occupant density in this model) will lead to higher levels of indoor humidity compared with a lower generation rate. This is reflected in negative model coefficients for medium and low occupant densities that get more negative as occupancy decreases (Medium = -0.00075 and Low = -0.00142). To illustrate, a large home in Charleston goes from a slightly positive humidity balance with high moisture generation (0.000039 kg/kg) to increasingly negative values when decreasing the moisture generation rate to medium (-0.00072 kg/kg) and low (-0.0014 kg/kg). From this, we can see generally that large homes with low occupancy will experience the most increase of indoor humidity when ventilating (largest negative values of net-humidity balance), and that small homes with high occupancy will experience the most decrease of indoor humidity when ventilating.

We tested fans sized at 0%, 50% and 100% of the 62.2-2013 ventilation rates to assess the impacts of changing the ventilation rate. The impact of smaller or larger fans depended on the net-humidity balance. As expected, with a positive net-humidity balance, ventilation dehumidified the space, and more ventilation dehumidified more. With a negative net-humidity balance, ventilation humidified the space, and more ventilation humidified more. We see this illustrated in Figure 5, particularly in the “high” and “low” moisture generation categories. The net-humidity balance is positive and large in the “high” cases (left figure), and increasing ventilation from 0% to 50% to 100% of 62.2 rates reduces periods of high indoor humidity from approximately 50% of the year down to 30% (right figure). In “low” cases, the net-humidity balance is negative (left figure), and increasing ventilation from 0% to 100% of 62.2 rates leads to increased periods of high humidity from approximately 10% to 20% of the year (right figure). These effects are non-linear, and they mostly indicate the direction, but not magnitude, of the changes in indoor humidity as ventilation rates are varied.

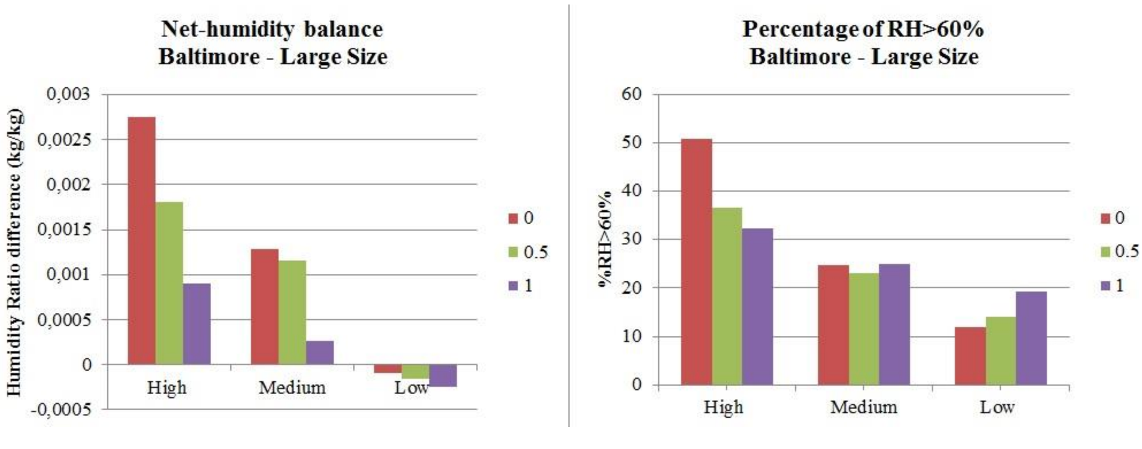


Figure 5: Relation between net-humidity balance and periods of high humidity in an example case of a large sized home in Baltimore with varying moisture generation rates and whole house ventilation rates (0, 50%, 100% 62.2)

2.3.3 Seasonal and monthly variation in simulated humidity problems

We have discussed how the effect of ventilation on indoor humidity depends on the annual net-humidity balance between inside and outside. This same effect occurs on a monthly basis.

In Table 4, we present the monthly net-humidity balances for the 100%[2] of ASHRAE 62.2-2013 flow rates, averaging across home sizes and occupancy rates. As before, positive values (green cells) indicate the ventilation will provide a net-humidity benefit, and negative values (red cells) indicate a net-penalty. Light-green and pink cells indicate months with marginal smaller net-humidity differences. Each climate zone has a clear seasonal pattern, with negative values in the summer and positive values during the other months of the year.

Table 4: Monthly mean differences between indoor and outdoor humidity

1	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Miami	0.0019	0.0014	0.0006	-0.0008	-0.0028	-0.0049	-0.0055	-0.0053	-0.0046	-0.0031	-0.0006	0.0011
Houston	0.0012	0.0012	0.0014	0.0009	-0.0012	-0.0044	-0.0057	-0.0057	-0.0030	0.0013	0.0019	0.0018
Memphis	0.0019	0.0014	0.0018	0.0012	0.0003	-0.0024	-0.0040	-0.0032	-0.0014	0.0021	0.0023	0.0020
Baltimore	0.0015	0.0017	0.0014	0.0016	0.0009	0.0001	-0.0018	-0.0010	0.0005	0.0027	0.0018	0.0017
Charleston	0.0018	0.0017	0.0012	0.0013	-0.0001	-0.0027	-0.0045	-0.0037	-0.0012	0.0017	0.0020	0.0020
Orlando	0.0012	0.0018	0.0011	0.0012	-0.0012	-0.0031	-0.0056	-0.0045	-0.0029	-0.0011	0.0015	0.0017

When comparing between the house size and occupancy rate parameters, the seasonal monthly patterns do not change in most cases, but the magnitude of the humidity differences shift up or down. In cases where the seasonal pattern does shift, the mean values of the humidity differences during those months tend to be small (i.e., an order of magnitude smaller than the differences found in non-shifting months). This limits the overall impact of any given month going from a positive to negative humidity difference. To account for this shifting on the margins, we design our seasonal control to be based on the dark red months.

This average seasonal pattern is not the best fit in all cases. For example, one of the worst-cases is a large, low-occupancy home in Miami when compared to the average in Table 4. In this case, two months are added to the dark red category, with fairly large negative humidity differences, and our control based on the average will over-ventilate during these months, leading to higher humidity. But in most cases, these shifts are smaller and less important.

2.4 Proposed seasonal control strategy

Based on the previous analysis, we present our proposed seasonal ventilation control strategy in Table 5. The red months are the periods when ventilation is a liability, and the green months when ventilation is a benefit. Therefore a seasonal controller should increase the ventilation rate during green months and decrease it during red months. The magnitude of these increases and decreases must be designed so that annual exposure to pollutants is equivalent with a continuous fan. For our control strategy, this means targeting different relative exposure values in different months (see Table 5). In red months, we target a fixed exposure of 1.5, which reflects a reduction of approximately 33% in the ventilation rate. There are differing numbers of red months depending on the climate zone. So, for each climate, we calculated the required exposure target for the green months, such that the annual average would be equal to one. A real-time ventilation (RTV) controller was then used to achieve these exposure targets. The RTV controller used the equivalence approach (Sherman, M. H., Walker, I. S., & Logue, J. M. (2012) to operate the CFIS system to achieve the target exposure rates.

[2]In this work we will show only the results for a fan sized at 100% of ASHRAE 62.2-2013.

Table 5: Values of the Exposure Targets set for different months of the year.

	Green Months	Exposure Target	Red Months	Exposure Target	Green Months	Exposure Target
Miami	Jan-Apr	0.46	May-Oct	1.5	Nov-Dec	0.46
Orlando	Jan-Apr	0.46	May-Oct	1.5	Nov-Dec	0.46
Houston	Jan-Apr	0.608	May-Sep	1.5	Oct-Dec	0.608
Charleston	Jan-May	0.72	June-Sep	1.5	Oct-Dec	0.72
Memphis	Jan-May	0.72	June-Sep	1.5	Oct-Dec	0.72
Baltimore	Jan-June	0.876	July-Aug	1.5	Sep-Dec	0.876

2.4.1 Preliminary results and discussion

We implemented this seasonal control strategy using the REGCAP simulation tool, and here we show preliminary results for the medium sized home with medium occupancy rate using a CFIS ventilation system (see Table 6). The proportion of the year over 60% and 70% humidity is presented, along with the maximum time periods continuously above 60% and 70% RH. These values are presented and compared for the baseline and control cases, and the changes are summarized for each, along with energy use for the control strategy.

Table 6: Performance summary of the seasonal smart control strategy.

	BASELINE					SEASONAL CONTROL					**COMPARISON**				
	R60 %	R70 %	Max R60 days	Max R70 days	Total KWh	R60 %	R70 %	Max R60 days	Max R70 days	Total KWh	R60 %	R70 %	Reduc% R60 days	Reduc% R70 days	Total KWh
Miami	88	41	26	10	3700	80	29	21	5	4007	**-9**	**-30**	**-16**	**-50**	**+338**
Orlando	68	26	25	23	5358	58	18	12	9	5737	**-14**	**-30**	**-67**	**-23**	**+379**
Houston	64	32	29	16	10649	57	27	23	12	10917	**-10**	**18**	**-21**	**-28**	**+267**
Charles.	64	31	0	47	10994	62	23	0	6	11337	**-4**	**-24**	**0**	**-47**	**+343**
Memph.	38	12	16	5	16172	33	10	14	5	16710	**-11**	**-21**	**-17**	**-4**	**+538**
Baltim.	30	14	45	0	22463	29	14	30	0	29691	**-3**	**-1**	**-34**	**0**	**+228**

The control strategy increased HVAC energy use in all cases. The controller was most effective in the most humid climates, but was not effective in Baltimore. In general, the controller was more effective at reducing hours above 70% than those above 60%. These higher humidity hours are the most important to eliminate, so the strategy was having some success. Nevertheless, our maximum impact is only a 30% reduction in annual hours above 70% RH, which leaves substantial periods of high humidity despite smart control. However, the controller substantially reduced the longest continuous time periods of high humidity. These sustained periods of continuously high humidity may be the most important to interrupt and shorten, and the controller succeeded at doing this.

3 CONCLUSIONS

The smart seasonal control of ventilation rates leads to improved indoor humidity control in the more humid climates, with modest increases in annual energy use. It is less effective in intermediate climates. Given that the maximum reduction in high humidity hours was around 30% in Miami and Orlando, other smart control strategies are worth developing. We are in the process of testing more advanced seasonal control strategies, as well as real-time strategies using indoor and outdoor humidity sensors. Yet, ventilation is a secondary factor affecting indoor humidity, and smart ventilation controls are unlikely to be a fully sufficient solution on their own. They may need to be implemented alongside mechanical dehumidifiers or other solutions, which we are also exploring in future work.

4 ACKNOWLEDGEMENTS

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program, of the U.S. Department of Energy under Contract No. DE-AC02-05CH11231.

5 REFERENCES

ANSI/ASHRAE. (2013). Standard 62.2-2013 Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings. Atlanta, GA: ASHRAE.

Kerrigan, P., & Norton, P. (2014). Evaluation of the performance of houses with and without supplemental dehumidification in a hot-humid climate. Golden, CO: National Renewable Energy Laboratory. Retrieved from http://apps1.eere.energy.gov/buildings/publications/pdfs/building_america/houses_supplemental_dehumidification.pdf

Moyer, N., Chasar, D., Hoak, D., & Chandra, S. (2004). Assessing Six Residential Ventilation Techniques in Hot and Humid Climates. Presented at the ACEEE 2004 Summer Study on Energy Efficiency in Buildings, Pacific Grove, CA: American Council for an Energy-Efficient Economy. Retrieved from http://www.fsec.ucf.edu/en/publications/pdf/FSEC-PF-378-04.pdf

Rudd, A. (2013). Supplemental Dehumidification in Warm-Humid Climates (Building America Report No. 1310). Somerville, MA: Building Science Corporation. Retrieved from http://www.buildingscience.com/documents/bareports/ba-1310-supplemental-dehumidification-warm-humid-climates

Rudd, A., & Henderson, Jr., H. I. (2007). Monitored Indoor Moisture and Temperatue Conditions in Humid-Climate US Residences. ASHRAE Transactions, 113(1), 435–449.

Rudd, A., Listiburek, J. W., & Ueno, K. (2005). Residential Dehumidification Systems Research for Hot-Humid Climates. Septebmer 1, 2001-December 30th, 2003 (No. NREL/SR-550-36643). Golden, CO: National Renewable Energy Laboratory. Retrieved from http://www.nrel.gov/docs/fy05osti/36643.pdf

Sherman, M. H., Mortensen, D. K., & Walker, I. S. (2011). Derivation of Equivalent Continuous Dilution for Cyclic, Unsteady Driving Forces. International Journal of Heat and Mass Transfer, 54(11-12), 2696–2702.

Sherman, M. H., Walker, I. S., & Logue, J. M. (2012). Equivalence in ventilation and indoor air quality. HVAC&R Research, 18(4), 760–773. http://doi.org/10.1080/10789669.2012.667038

Walker, I. S., & Sherman, M. H. (2006). Evaluation of Existing Technologies for Meeting Residential Ventilation Requirements (No. LBNL-59998). Berkeley, CA: Lawrence Berkeley National Lab. Retrieved from http://epb.lbl.gov/publications/pdf/lbnl-59998.pdf

Martin, E. (2014). Impact of Residential Mechanical Ventilation on Energy Cost and Humidity Control (No. NREL-60675). Golden, CO: National Renewable Energy

Laboratory. Retrieved from http://www.fsec.ucf.edu/en/publications/pdf/NREL-60675.pdf

U.S. Department of Energy. (2013, April 17). DOE Challenge Home National Program Requirements (Rev. 03). U.S. Department of Energy. Retrieved from http://www1.eere.energy.gov/buildings/residential/pdfs/doe_challenge_home_requirementsv3.pdf

RESNET. (2006). 2006 Mortgage Industry National Home Energy Rating Systems Standards. Residential Energy Services Network. Retrieved from http://www.resnet.us/professional/standards/mortgage

THERMODYNAMIC ANALYSIS OF BUILDINGS WITH NATURAL VENTILATION AND INDOOR AIR QUALITY

Víctor Moreno[1,*], Amalia Roca[1]

[1] *ISOLANA Ahorro Energético SL*
Carrer Cobalto, 95
L´Hospitalet de Llobregat, Spain

** Corresponding author´s email: vmoreno@isolana.es*

ABSTRACT

The aim of this study is to analyse the behaviour of natural ventilation techniques in low-rise commercial buildings in terms of Indoor Air Quality (IAQ). Verifying how the outside air flow can enter a building using natural ventilation techniques to check if they are suitable to be bound by the regulations, thus validating passive techniques for ventilating buildings.

With the emergence of the regulation of thermal installations in buildings (RITE) in Spain, the basic regulatory framework that regulates the requirements in energy efficiency and security is set up, thermal installations in buildings need to meet the demand of welfare. It is compulsory to vent all conditioned spaces of a building using fans, which involves installing ducts and equipment. With the RITE approval is compulsory to install them in buildings that do not have cooling demand. This implies an increase in the cost of ventilation and operation.

An analysis of different foreign regulations related to natural ventilation is performed, the UK regulations Building Bulletin 101 - Ventilation of School Buildings and the ASHRAE Standard 62.1-2007 - Ventilation for acceptable indoor air quality. The design requirements that these regulations prescribe are collected. In these guidelines, it is possible to introduce the necessary outside air in buildings using natural ventilation techniques, in order to maintain IAQ, healthy environments and avoid potential pathologies related to comfort inside buildings.

A case study is carried out, designing a low-rise commercial building prototype with natural ventilation systems. It is calculated the size, distances and orientations of the openings necessary in each occupied zone. Subsequently it is checked the design made by computational thermodynamics simulation. The thermodynamic simulation tool used to test the feasibility of natural ventilation techniques is Energy Plus, which by Airflow Network module enables the modelling of natural ventilation in buildings. Additionally, it is also used Computational Fluid Dynamics (CFD) simulations to properly design natural ventilation systems and to validate inside thermal comfort of the proposed building.

The study work scheme has the following parts:

1. Calculation of the intake outside air needs in the model building, according to the Spanish regulation.
2. Design of Natural Ventilation systems to implement into the model building.
3. Thermodynamic simulations are performed to verify the compliance with the requirements of the Spanish normative.

4. Analysis of results.

Finally, it is verified that in normal weather conditions and proper design of natural ventilation systems, indoor air quality meets the requirements of regulations. In conclusion, thanks to the implementation of this technology we can achieve considerable savings in the implementation and operation, along with a decrease in CO_2 emissions to the atmosphere.

KEYWORDS

natural ventilation, indoor air quality, cfd, energy savings, thermodinamic modeling

1 INTRODUCTION

The reason of this study is the need to know the advantages of implementing natural ventilation methodologies in commercial buildings. The system will be analyzed to find out if the regulatory requirements in terms of Welfare and Health inside buildings are met, compared to other ventilation systems in buildings and analyze its energy, economic and environmental impact. The ultimate goal is to show that the IAQ analyzed and CO_2 concentration inside buildings are within the ranges established by law using natural ventilation systems as well as using mechanical ventilation.

To verify that the regulations are met using natural ventilation techniques, a commercial building prototype is modeled using a thermodynamic simulator. Several viable options for the implementation of natural ventilation systems are studied, the necessary outdoor openings are sized and thermodynamic and CFD simulations are performed, the results give us the amount of external air introduced in each internal area of the building, evaluating whether the airflows meet the requirements of the regulations. The energy simulation tools used for the calculation of natural ventilation and internal comfort is Energy Plus[1].

A building prototype is developed without incorporating the design measures necessary for natural ventilation. This building will have heating system and mechanical ventilation system and will serve as a reference when comparing the results obtained in the proposed building using natural ventilation systems.

Energy consumption and outside air flows supplied to each zone are compared, verifying the feasibility of using natural ventilation systems for ventilating commercial buildings.

2 REGULATIONS

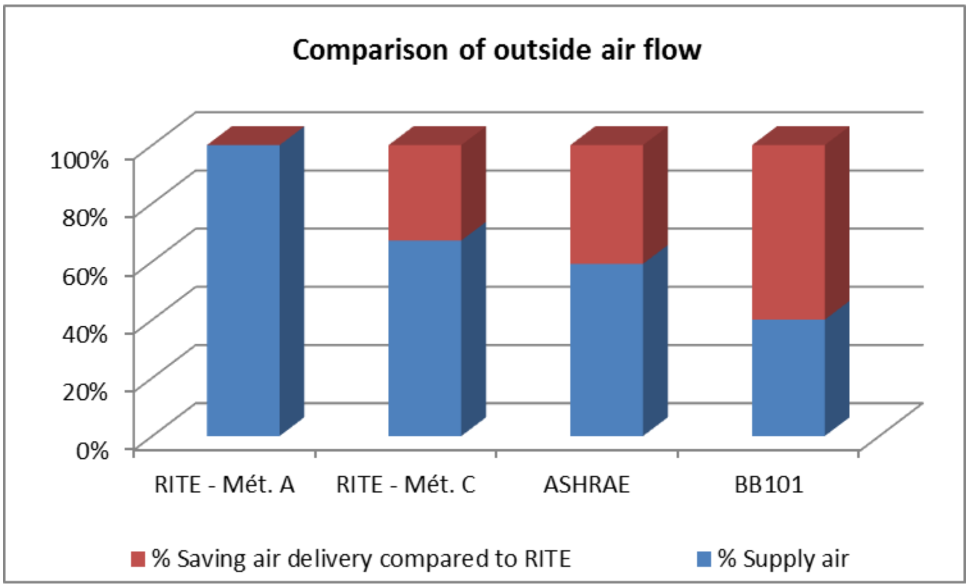
The Regulation of Thermal Installations in Buildings (RITE) in Spain establishes the conditions that ventilation, heating, cooling and domestic hot water installations must met to facilitate the demand for thermal and health wellness through a rational use of energy. RITE does not prescribe the implementation of natural ventilation systems in the design, in comparison to regulations in others developed countries where these systems are allowed. But RITE allows the possibility of alternative solutions using a simplified procedure, allowing building ventilation techniques using natural techniques, as long as its technical feasibility is justified.

[1]EnergyPlus - U.S. Department of Energy. http://apps1.eere.energy.gov/buildings/energyplus/

The requirements to be viable as required by RITE by designs using natural ventilation systems in this study are explained. Other foreign policies as the ASHRAE Standard 60.1 or UK regulations allow natural ventilation as a viable strategy to maintain a good IAQ in buildings.

If we compare outdoor air requirements necessary to enter in commercial buildings to maintain good IAQ, we can see how the Spanish regulations (RITE) requires a greater supply flow of outside air against regulations in USA and UK. According to the method of calculating the flow of outside air that enables RITE, ventilation flows vary, even more if we rely on the regulations that RITE referred to, as for example the UNE-EN 13779.

Using Method C. RITE direct method for CO_2 concentration and the different calculation methods described in the regulations: *UNE-EN 13779 - Ventilation for non-residential buildings, NTP 742 - General ventilation of buildings and NTP 549 - Carbon dioxide in assessing of indoor air quality*; a value of 8.4 l/s·person as exterior air flow is specified as necessary to introduce in the occupied areas of the building assessed, since RITE specifies the maximum values of the ranges of comfort described in the regulations specified above, specifying a outside air flow using Method A. RITE indirect method of outside air flow which is 12.5 l/s·person.



2.1 REQUIREMENTS FOR INDOOR AIR QUALITY (RITE) IN A TERTIARY BUILDING PROTOTYPE

The first point of the analysis is the calculations of the outside air flows required to enter in each area of the prototype building for tertiary use at low altitudes to ensure that IAQ is suitable as required by regulations. For this reason, the tertiary building prototype is designed and requirements are analyzed. According to the table surfaces of the building prototype we calculate the occupation of each area using the designated occupancy rates. In this way, we can calculate the requirements for the admission of outside air for each zone of the Prototype. The most significant areas of the building to be naturally ventilated are the office areas, because these areas have the greatest occupation in terms of number of people and hours of use per year. The building has three types of offices easily distinguishable by their size and location in the building. The calculations required by the regulations specified above are performed, obtaining the following needs of air changes per hour (ACH) depending on the type of office analyzed

- Offices type 1: 5.0 ACH
- Offices type 2: 5.0 ACH
- Offices type 3: 3.3 ACH

3 APPLIED SOLUTIONS TO THE TERTIARY PROTOTYPE TO BUILDING AND ITS TECHNICAL DEVELOPMENT

Natural ventilation is produced mainly by two mechanisms: the chimney effect and wind pressure ("wind effect"). The chimney effect arises from the decrease in air density when its temperature rises. Wind pressure also influences the ventilation of a building by creating pressure variations around the outside of the building. The pressure variations are highly dependent on the shape of the building, speed and wind direction.

The design of the openings and their arrangement in the façade is a key factor, besides the opening surface, to ensure that the external air flow entering each zone is adequate. To be able to analyze the impact that has the arrangement of the openings in the exterior walls on the ventilation in zones, different simulations were performed using CFD techniques. CFD simulations are used, because the thermodynamic simulations are not feasible for analyzing openings design. This is because the thermodynamic and energy simulation programs are able to model and give results of the air flows coming into the building, the flows leaving the building and internal paths that perform the air flows between different areas inside the building. But do not discriminate between the situation of openings in the façade, neither inside air movements by natural convection effects. For this reason, CFD techniques are used.

Firstly, a pre-dimensioning of the openings to the outside surface is performed to achieve introduce the external air quantity specified in regulations. This analysis is performed using different methods of empirical calculation as *AM10CalcToolv5 (CIBSE), Florida Solar Energy Method I and II, ASHRAE, Aynsley and British Standard*. Later we validate the sizing of the openings to the outside via thermodynamic simulation using Energy Plus, specifically using its *AirFlowNetwork* module that allows the simulation of multi-nodal natural ventilation models, considering differences in internal pressures and external air flows between zones.

Once the outside necessary openings are specified, different design options in the openings in the façade are analyzed, distributing the opening surface and its height in the façade to take advantage of natural convection inside the areas to be ventilated naturally. Creating cross ventilation, and designing ventilation shunts with expulsion of air through chimneys favoring the chimney effect of the inside air flow, helping to evacuate the inside air and improving ventilation of the area. 6 different distribution openings (glazing) designs were developed on the facade, keeping the opening area constant, because this value of opening area ensures the required air flow. The design options analyzed are described as follows:

- <u>1 openable window</u>: The air intake is performed by opening the bottom of the window (openable windows).

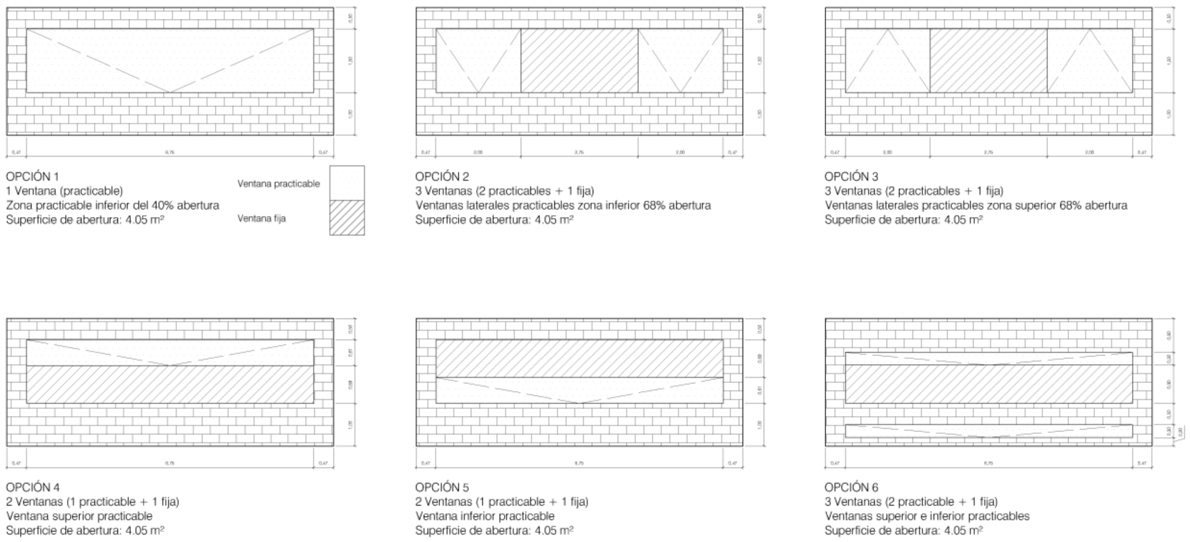
- <u>3 windows (2 opening + 1 fixed)</u>: The air intake is performed by opening the bottom of the 2 opening windows (openable windows).

- <u>3 windows (2 opening + 1 fixed)</u>: The air intake is performed by opening the top of the 2 opening windows (openable windows).

- <u>2 windows (one practicable + 1 fixed)</u>: The air intake is performed by opening the upper window (longitudinal window).

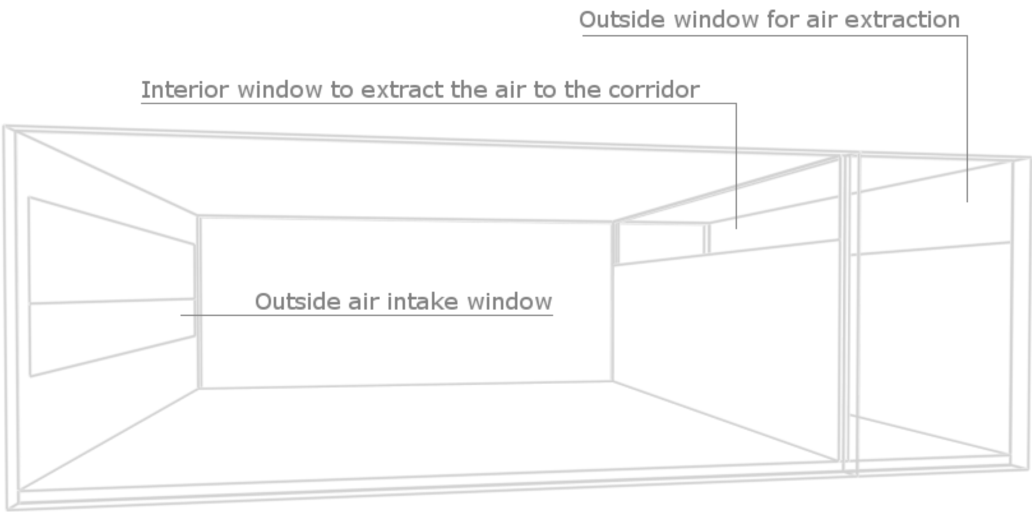
- <u>2 windows (one practicable + 1 fixed)</u>: The air intake is performed by opening the lower window (longitudinal window).

- <u>3 windows (2 practicable + 1 fixed)</u>: The ventilation of the area is done by opening the lower longitudinal windows (air intake) and upper (exhaust air)



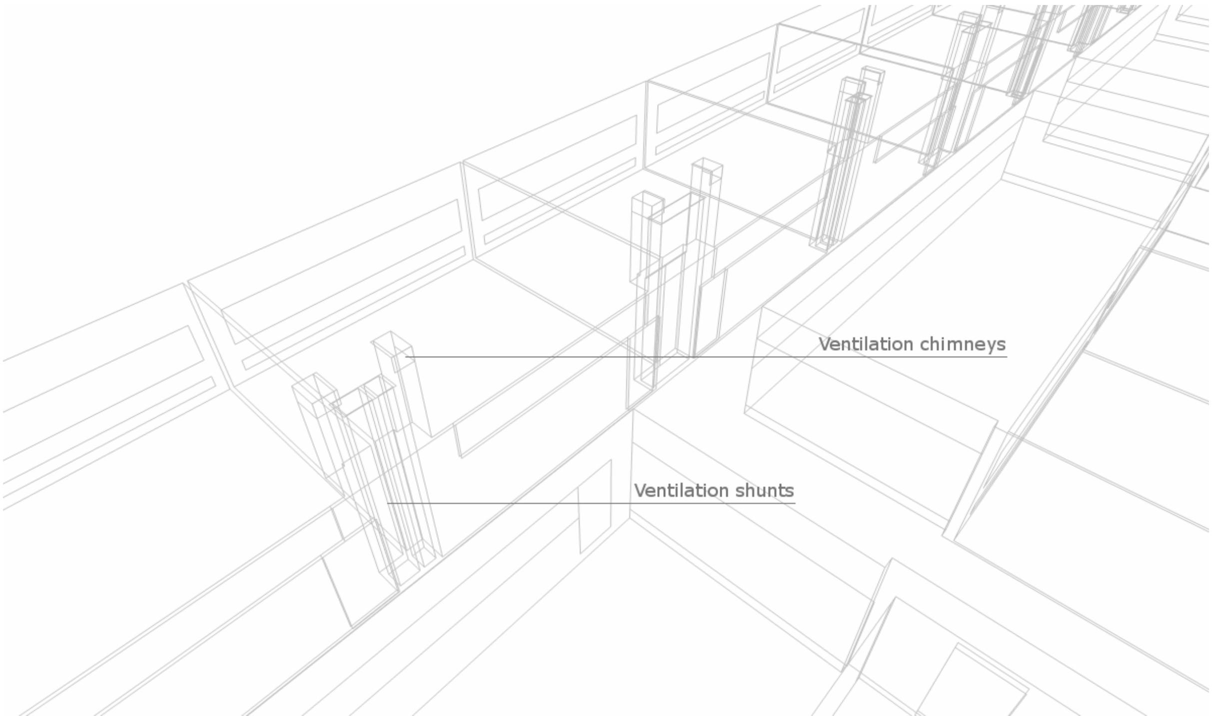
In addition to the different situations of the openings in the façade, two techniques for cross ventilation are evaluated. The first is achieved by arranging an opening in the top of the inner partition which separates each office with the corridor. And the second creating shunts and ventilation chimneys.

- <u>3 windows (1 practicable + 1 Fixed + interior to corridor)</u>: The ventilation of the area is performed by opening the lower longitudinal window (air intake) and the inner window (air discharge), which extracts the air into the corridor area which is ventilated with outside air.



- 2 windows (one practicable + 1 fixed + ventilation shunts): The ventilation of the area is performed by opening the lower longitudinal window (air intake) and opening the vents of the shunts (exhaust air), which extract the air from the zone to the outside area by the ventilation chimneys on deck. The admission of outside air can be introduced by a strip as specified in the following illustration, or through openings to the outside with motorized vents, the important to validate the solution is to measure the surface of opening to the outside according to the results obtained in the present study.

To allow cross ventilation in the zones in the floors below the top floor, ventilation shunts are designed which lead the exhaust air to ventilation chimneys on deck.

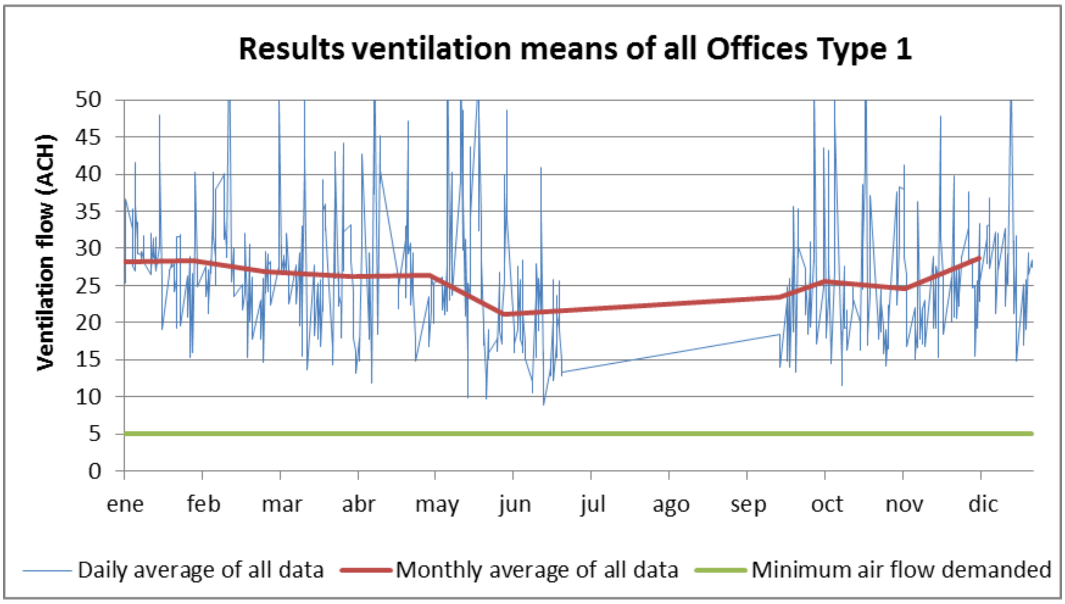


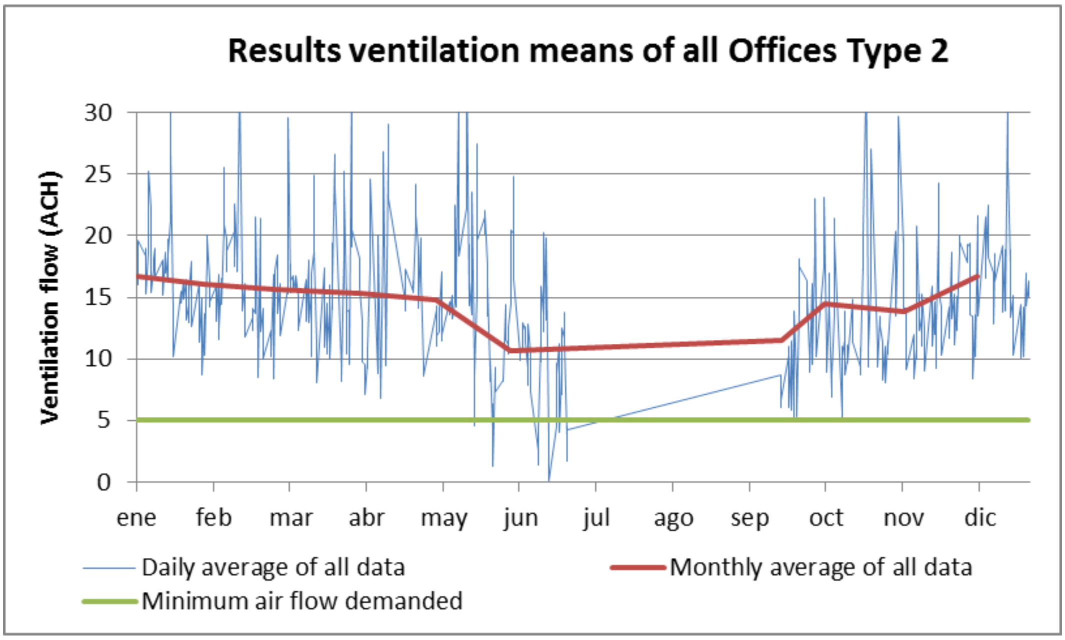
To perform the comparison between the 8 design options, different factors of natural ventilation that help us to make decisions in the most favourable option are analyzed, using design option of the natural ventilation system.

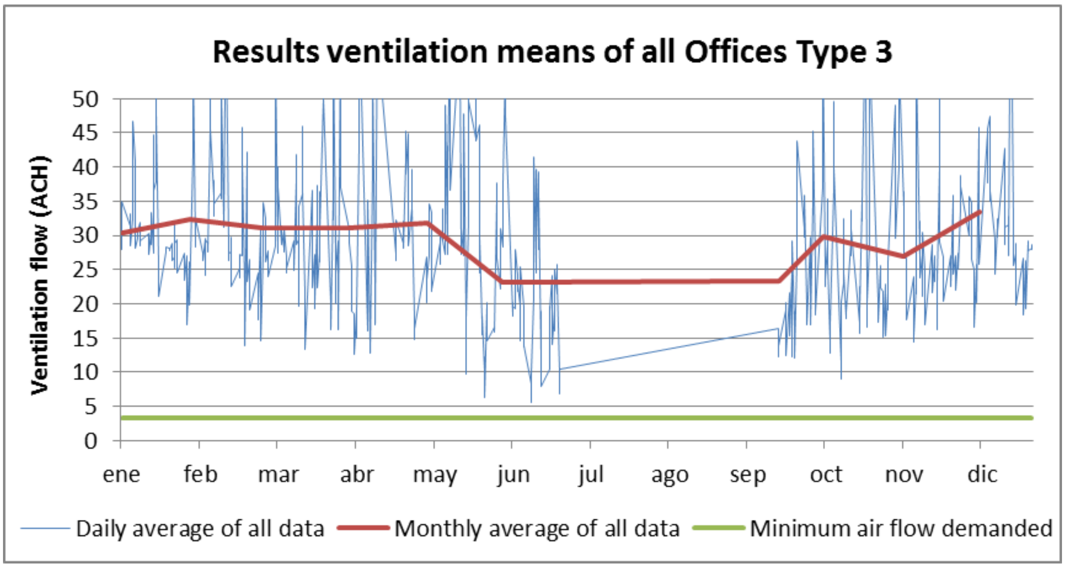
Finally, the design option that offers better results is based on the construction of different shunts in each area to be ventilated, with expulsion of air to deck through chimneys favoring the pressure difference between inside and outside, improving the stack effect of various shunts and ventilation in each area.

In the first place, outside air inlet opening surfaces through external façades for each office ventilated naturally are analyzed. It is remarked, that these surfaces are needed to provide the outside air flow in the most restrictive periods, summer times, since the effect of the thermal gradient is nonexistent. Thermal gradient is increasing and favoring the functioning of natural ventilation in winter times. The opening modulation of these ventilators should go from fully closed, when the rooms are empty or no ventilation is needed, to the desired and calculated opening surfaces.

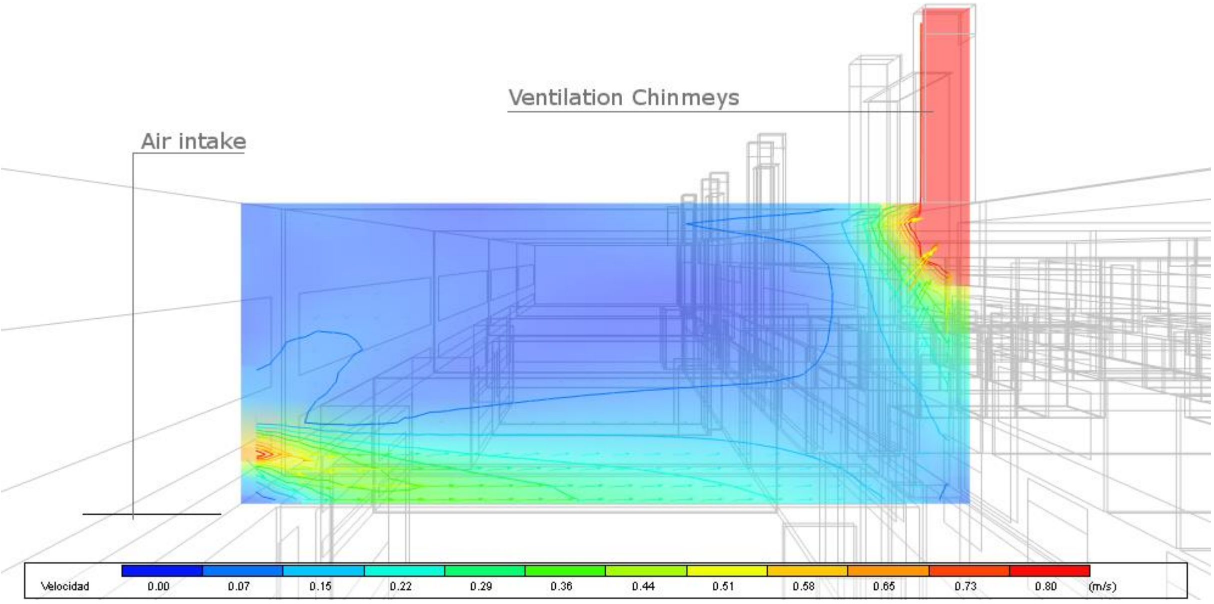
Air changes per hour and monthly averages results in the hours that the center is ventilated are presented:







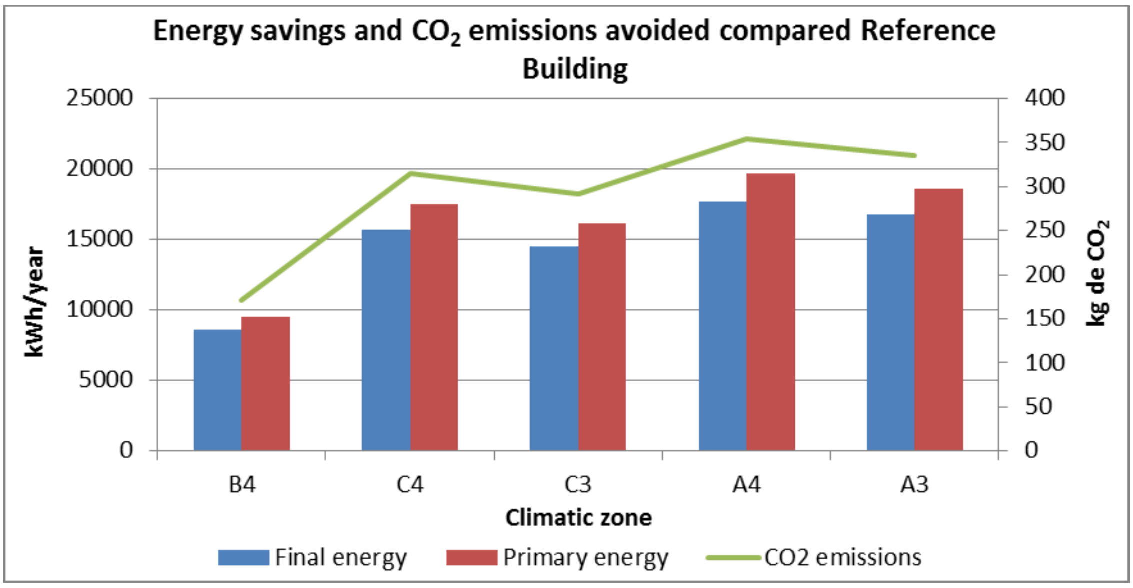
Air changes per hour are always above the minimum required, justifying the requirements of RITE. Thanks to the installation of motorized louvers in the openings to the outside, we can modulate the amount of air entering each zone, providing only the amount of the required air depending on the CO_2 concentration that exists in every area at every moment. The air flow pattern can be seen in the following image:



One of the objective of this study is to check the consumption of energy, besides the CO_2 emissions to the atmosphere, and these are lower in the Proposed Building using natural ventilation than in the Reference Building with mechanical ventilation and heat recovery, according to requirements of RITE.

This fact is justified by the consumption of the fans in the Reference building, which does not existent in a building with natural ventilation system, however there is the disadvantage that there is not the possibility to install an energy recovery system.

Energy simulations in both models were performed, in the Reference and Proposed Building in the warmest climatic zones in Spain, the climatic zones are A3, A4, B4, C3, C4 from the technical building code in Spain (CTE). The results of energy consumption obtained demonstrate that the energy consumption of the fans to condition the analyzed building is greater than the energy that can recover the energy recovery system. Thus, it has been demonstrated the energy savings needed by RITE in reference to ventilation systems.



4 CONCLUSIONS

The viability of the system has been demonstrated, even with the energy disadvantage due to the lack of heat recovery, since the energy savings produced by the absence of the energy consumption of the fans is bigger than the savings caused by the heat recovery in a building with the same characteristics using a mechanical ventilation system.

If we analyze the results of the air changes per hour of the outside air flow for the entire building, we checked that during all hours that the building is occupied, the outside air flow rates are suited to those required by the rules of obligatory fulfillment.

5 REFERENCES

(2007). Royal Decree 1027/2007: Regulation of thermal installations in buildings in Spain (RITE).

(2005). UNE EN 13779 Ventilation for non-residential buildings. Performance requirements for ventilation systems and conditioning of enclosures.

(2010). ANSI/ASHRAE Standard 62.1-2010 Ventilation for acceptable Indoor Air Quality.

(1996). A Guide to Energy Efficient Ventilation (AIVC).

(2005). AM10 Natural Ventilation in nondomestic buildings (CIBSE Applications Manual).

(1998). Natural Ventilation in Buildings: A Design Handbook (Earthscan Ltd).

PERFORMANCES OF A SIMPLE EXHAUST MECHANICAL VENTILATION COUPLED TO A MINI HEAT PUMP: MODELING AND EXPERIMENTAL INVESTIGATIONS

Frédéric Ransy*[1], Samuel Gendebien[1] and Vincent Lemort[1]

1 University of Liege, Thermodynamics Laboratory
Quartier Polytech 1
Allée de la Découverte, 17
4000 Liège, Belgium
**Corresponding author: Frederic.ransy@ulg.ac.be*

ABSTRACT

Nowadays, important efforts are deployed to reduce energy consumption in the field of residential buildings. Concerning new constructions, low-energy consumption buildings such as “passive” houses constitute a suitable solution to decrease the environmental impacts.
In this kind of building, air tightness is improved and heating needs are reduced compared to traditional constructions. In order to ensure a good indoor air quality, controlled mechanical ventilation is required. Coupling a simple exhaust mechanical ventilation with a mini-heat pump appears to be a good solution for heat recovery.
This latter system could provide an important (and even the whole) part of the heating demand related to domestic hot water. The system can also be coupled to the heating system of the building. Investigation on this system is carried out in the present paper.
The first part of the paper presents the system, its components and its control. A semi-empirical numerical model of the whole system is presented. This model associates the sub-models of the main components: the compressor, the condenser, the evaporator (in this case, the ventilation heat recovery heat exchanger), the domestic hot water heat exchanger and the domestic hot water storage tank.
The second part of the paper describes the experimental apparatus (and its control) designed to characterize the performance of the system in different operating conditions.
Experimental data are presented and analyzed, which includes a calibration of the parameters of the model and a comparison with simulation results.
Finally, the model is coupled to a building simulation model to determine the potential savings of such units.

KEYWORDS

Simple exhaust mechanical ventilation, heat recovery, heat pump, experimental tests, building simulation

1 INTRODUCTION

According to the European directive 2012/27/EU of October 2012 on energy efficiency, buildings represented 40 % of the EU's final energy consumption in 2011. The major part of this energy consumption is due to the residential sector for space heating and domestic hot water production. Moreover, buildings are crucial to achieve the EU objective of reducing greenhouse gas emissions by 80-95 % by 2050 compared to 1990.

In order to reduce these greenhouse gas emissions, retrofit measures regarding insulation and air-tightness have to be taken. However, such improvements of the building envelope lead to a relative increase in consumption related to ventilation. Indeed, according to Orme (2001), Roulet et al. (2001) and Fouih et al. (2012), the heating demand due to ventilation can reach more than 50 % of the total building heating demand for new and retrofitted buildings.

To reduce the energy consumption due to ventilation, exhaust air heat pumps (EAHPs) can be used instead of the traditional heat recovery with an air-to-air heat exchanger. EAHPs recover heat from the exhaust air of the ventilation system to produce domestic hot water and space heating. According to Fehrm et al. (2002), this technology is already widely used in the northern countries such as Germany and Sweden. In fact, according to Fracastoro et al. (2010), efficiencies of EAHPs are higher than those obtained with outside air or geothermal heat pumps in certain conditions, whatever the climate location. Berg et al. (2010) have monitored three houses in Sweden equiped with exhaust air heat pumps. The seasonal performance factor (SPF) values were all within the range 1.4-1.7. This factor takes into account the energy consumption of the heat pumps and the auxiliary heating systems. A 17 kW exhaust air heat pump has also been tested by Mikola et al. (2014). The measured SPF for the heat pump only (without taking into account the auxiliary heating system) was about 2.9-3.4 in winter and 3 in the summer.

Exhaust air heat pumps coupled with simple exhaust mechanical ventilation systems have many advantages compared to traditional balanced systems with heat recovery:

- Only one fan is necessary and the duct system is simpler. Consequently, EAHPs are suitable for retrofitted buildings.
- The heat pump can provide the whole part of the heating demand related to domestic hot water and 50 % of the heating demand related to space heating, according to Fracastoro et al. (2010).
- The heat pump can also provide active cooling by inversing the refrigerating cycle.
- The heat pump performance is high and remains constant with outdoor temperature changes since the temperature of the heat sink is constant (20°C). As a result, the system is cost-effective.
- The system is compact, quiet and requires little maintenance.

In the present paper, the energetic performances of an exhaust air heat pump are assessed through numerical and experimental studies. The thermal capacity of the machine is 1.5 kW when the inside air temperature is 20°C and the outside water temperature is 35°C. The heat pump is therefore ideally suited for new or retrofitted buildings. The system including a mechanical exhaust ventilation system and an exhaust air heat pump is first presented. Secondly, the heat pump model used afterwards to determine the heat pump seasonal performance factor is described. Thirdly, the model is calibrated to fit the measurement data. Finally, the heat pump model is coupled to a building model to determine the annual performance of the system. The system is compared to a traditional balanced ventilation system with heat recovery in terms of primary energy consumption, for different heating and DHW production systems (electric heater, heat pump, gas condensing boiler).

2 DESCRIPTION OF THE SYSTEM

The system is composed of a mechanical exhaust ventilation with natural air supply and an air-to-water heat pump for heat recovery purpose. The air side of the heat pump is connected to the exhaust duct of the ventilation system, while the water side is connected to a hot water storage tank. If necessary, it can be also connected to the heating circuit of the building. The schematic diagram of the system is shown in Figure 1.
The main components of the exhaust air heat pump investigated in this paper are shown in Figure 2 and described below.

- The compressor (CP in Figure 2) is an air cooled and hermetic rotary compressor working with R134a. Its control is a simple ON/OFF switch.
- The condenser (CD in Figure 2) is a water/refrigerant plate heat exchanger with a nominal water flow rate of 400 l/h. This exchanger has been chosen for its high efficiency, high compactness and low price.
- The evaporator (EV in Figure 2) is an air/refrigerant finned-tube heat exchanger with a nominal air flow rate of 215 m^3/h. This exchanger has been chosen for its good efficiency due to its high heat-exchange area and low price.
- The thermostatic expansion valve maintains the evaporator overheating constant (5K).
- The internal heat exchanger (IHE in Figure 2) is a simple refrigerant/refrigerant heat exchanger. The tubes are simply welded to realize a heat transfer by conduction. This welding provides a small overheating before the supply of the compressor to avoid the presence of liquid. The effect of this exchanger on the performance of the machine is negligible. For this reason, the modeling of the internal heat exchanger is not considered in this paper.
- The auxiliary equipments are the exhaust fan and the pump in the secondary water loop. The fan speed can be modified in order to provide the desired volume flow rate. On the other hand, the water pump rotation speed is constant. The pump is just turned off when heating capacity is not needed.

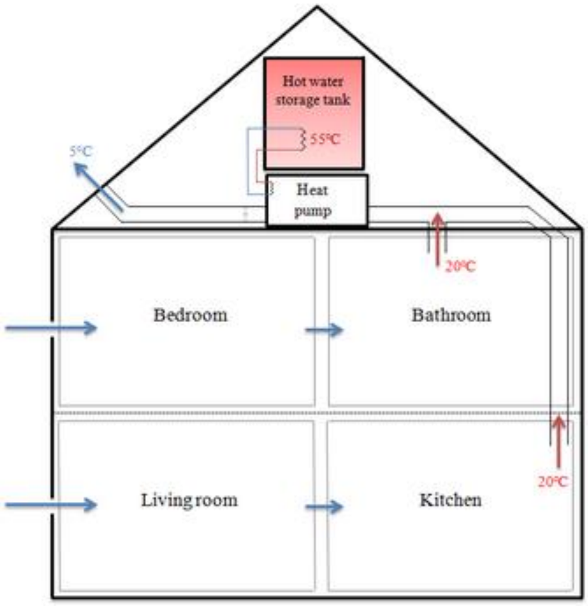


Figure 1: Schematic diagram of the system

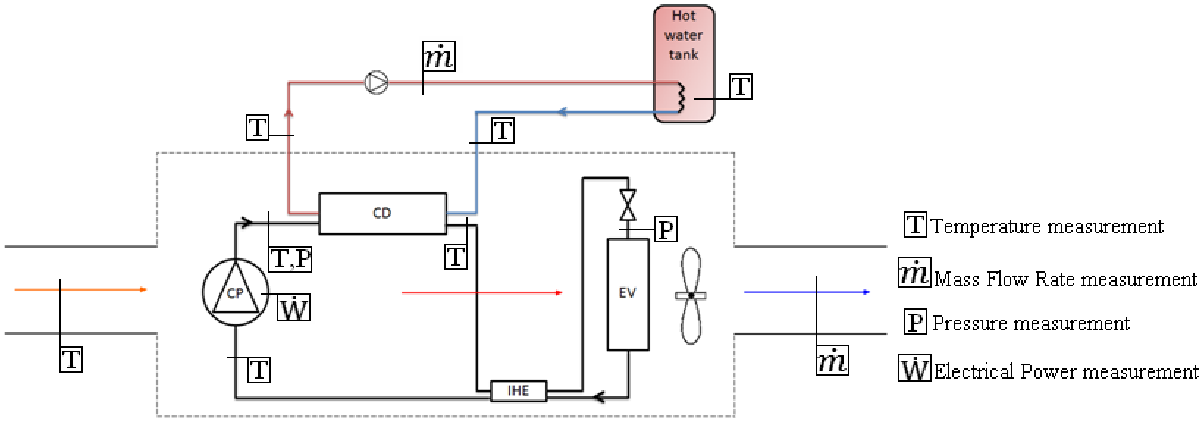


Figure 2: Schematic diagram of the heat pump

3 DESCRIPTION OF THE HEAT PUMP MODEL

A semi-empirical numerical model of the heat pump has been developed to determine its seasonal performance factor. Based on an identification of the components parameters based on experimental data, the model is able to predict the COP and the electrical consumption of the compressor. The objective is to predict, with a model as simple as possible, the performances with a maximum relative error of 5%. The heat pump is modeled in steady-state conditions whereas a dynamic model is used for the temperature evolution inside the water tank.

3.1 Compressor model

The compressor is described with three parameters: the swept volume V_s [m³], given in the compressor datasheet, the isentropic efficiency ε_{is} and the volumetric efficiency ε_v. The swept volume and the volumetric efficiency determine the refrigerant mass flow rate imposed by the compressor. The isentropic efficiency determines the compressor electrical consumption. The swept volume V_s is equal to 12.75 cm³.

The maximal refrigerant volume flow rate $\dot{V}_{s,cp}$ passing through the compressor depends on the rotational speed and the swept volume of the compressor, as given by equation (1):

$$\dot{V}_{s,cp} = \frac{\mathrm{N}}{60}\,\mathrm{V_s} \tag{1}$$

Due to the pressure drop at the compressor supply, the presence of the dead volume and internal leakage, the real volume flow rate $\dot{V}_{r,cp}$ is lower than the maximal refrigerant mass flow rate calculated by equation (1). It depends on the volumetric efficiency of the compressor and is calculated by equation (2):

$$\dot{V}_{r,cp} = \dot{\mathrm{m}}_{\mathrm{r,cp}} v_{r,su,cp} = \varepsilon_{\mathrm{v}}\,\dot{V}_{s,cp} \tag{2}$$

The specific volume $v_{r,su,cp}$ depends on the refrigerant pressure and temperature at the supply of the compressor.

The isentropic efficiency ε_{is} is used to take into account the thermal losses and the electrical motor efficiency. It is the ratio of the isentropic compressor consumption to the electrical compressor consumption, as given by equation (3):

$$\varepsilon_{is} = \frac{\dot{W}_{cp,is}}{\dot{W}_{cp,el}} = \frac{\dot{m}_{r,cp}(h_{r,ex,cp,is} - h_{r,su,cp})}{\dot{W}_{cp,el}} \quad (3)$$

with $h_{r,ex,cp,is}$ the isentropic specific enthalpy at the outlet of the compressor and $h_{r,su,cp}$ the specific enthalpy at the supply of the compressor.

As explained by Cuevas et al. (2010), the isentropic and volumetric efficiencies depend on the pressure ratio r_p which is the ratio of the condensing pressure to the evaporating pressure. In this paper, as proposed by Wenhua (2013) and Ouadha et al. (2008), the two efficiencies are estimated by polynomials forms given in equations (4) and (5):

$$\varepsilon_{is} = \alpha_0 + \alpha_1 r_p + \alpha_2 r_p^2 \quad (4)$$

$$\varepsilon_v = \beta_0 + \beta_1 r_p + \beta_2 r_p^2 \quad (5)$$

The parameters in equations (4) and (5) have been determined using measurement data. The calibration of these parameters is described in the next section.

3.2 Heat exchanger model

Each heat exchanger is described using one parameter: the global heat transfer coefficient AU at nominal conditions. This coefficient determines the heat exchanger efficiency. The exchangers are considered to be semi-isothermal and are modeled with the epsilon-NTU method. Table 1 gives the different fluids circulating in the three exchangers, for the isothermal and the secondary fluid sides.

Table 1: Isothermal and secondary fluids for the three exchangers

	Condenser	Evaporator	Domestic hot water heat exchanger
Fluid 1 (isothermal side)	Refrigerant	Refrigerant	Water inside the tank
Fluid 2 (secondary fluid)	Water (secondary water loop)	Exhaust air from ventilation	Water (secondary water loop)

The exchangers are described with the equations (6), (7) and (8):

$$\dot{Q} = \dot{m}_{iso}\,(h_{su,iso} - h_{ex,iso}) \quad (6)$$

$$\dot{Q} = \dot{m}_{sec}\,(h_{ex,sec} - h_{su,sec}) \quad (7)$$

$$\dot{Q} = \varepsilon\,\dot{m}_{sec}\,cp_{sec}\,(T_{iso} - T_{su,sec}) \quad (8)$$

with $\dot{Q}$ the heat flow exchanged between the two fluids, $\dot{m}_{iso}$ and $\dot{m}_{sec}$ respectively the mass flow rate of the isothermal fluid and the secondary fluid, $h_{su,iso}$ and $h_{ex,iso}$ the specific enthalpy of the isothermal fluid at the supply and the exhaust of the exchanger and $h_{ex,sec}$ and $h_{su,sec}$ the specific enthalpy of the secondary fluid at the exhaust and the supply of the exchanger. Equation (8) uses the definition of the exchanger efficiency ε. $\dot{m}_{sec}\,cp_{sec}$ is the thermal capacity of the secondary fluid. T_{iso} is the temperature of the isothermal fluid and $T_{su,sec}$ is the temperature of the secondary fluid at the supply of the exchanger.

The ε-NTU method is used to determine the exchanger efficiency. It depends on the global heat transfer coefficient and the secondary fluid heat capacity, as given by equation (9):

$$\varepsilon = 1 - \exp\left(\frac{-AU}{\dot{m}_{sec} cp_{sec}}\right) \tag{9}$$

3.3 Expansion valve model

The thermostatic expansion valve model supposes an isenthalpic expansion and it is also assumed that the expansion valve imposes a constant superheat of 5K at the outlet of the evaporator.

3.4 Domestic hot water tank model

The domestic hot water tank is a sensible heat storage with water as medium used to dissipate the heat produced by the air exhaust heat pump.
The tank is modeled by a one-node model with the water tank temperature T_{tank} as state variable. The energy balance of the tank is given by equation (10):

$$\dot{Q}_{tank} = \dot{Q}_{cd} = \dot{U}_{tank} = m_{w,tank} \cdot cp_{w} \cdot \frac{dT_{tank}}{dt} \tag{10}$$

with $\dot{Q}_{cd}$ the heat flux produced by the condenser of the heat pump, $\dot{U}_{tank}$ the internal energy variation of the water inside the tank, $m_{w,tank}cp_{w}$ and T_{tank} the heat capacity and the temperature of the water inside the tank. The inside tank temperature $T_{tank}(t)$ is calculated by integrating equation (10) over time.

4 DESCRIPTION OF THE HEAT PUMP TEST BENCH

The test bench developed to calibrate the heat pump model is composed of an air-to-water heat pump connected to a 300 liters hot water storage tank. The heat pump is connected to the tank through a secondary water loop in order to avoid any contact of the domestic hot water with the refrigerant.
Five input variables can influence the COP of the heat pump: the water mass flow rate at the condenser, the air mass flow rate at the evaporator, the water temperature inside the tank and the air temperature and humidity at the supply of the heat pump. The test set up is designed in order to have the possibility to change all these operating conditions. The water mass flow rate at the condenser can be adjusted with several valves of the hydraulic plant. The air flow rate can be modified by adjusting the rotational speed of the fan. The water temperature inside the tank can be kept constant. Moreover, it is also possible to heat the water inside the tank from 20°C to 60 °C only with the exhaust air heat pump. Lastly, the air temperature at the supply of the heat pump is kept constant for the duration of the test. The air humidity variation at the supply of the heat pump is not considered in this paper.
Different sensors have been placed on the test bench with the aim of determining the performances of the machine and calibrating the heat pump model. Figure 2 shows a schematic diagram of the test bench and the position of the different sensors.
Five types of sensors are used on the test bench: T-type thermocouples for the temperature measurements, absolute pressure sensors for the measurement of the condensing and the evaporating pressures, differential pressure sensors used for the airflow rate measurement, ultrasonic flowmeter for the water mass flow rate measurement at the condenser and electrical wall plug data loggers for electrical consumption measurements.
The measurement of the air volume flow rate is carried out with a nozzle connected to the exhaust duct on the air side of the heat pump. In fact, when the exhaust air flows through the

nozzle, a pressure drop proportional to the volume flow rate appears. This pressure drop is then measured by a differential pressure, which allows to measure the air volume flow rate. Table 2 gives the accuracy of the different sensors used on the test bench.

Table 2: Accuracy of the different sensors used on the test bench

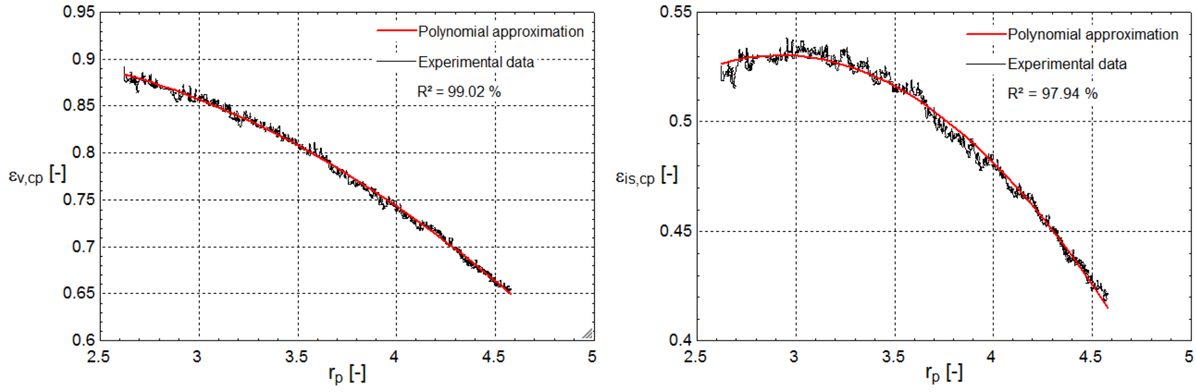
Thermocouples type T	Absolute pressure sensors	Differential pressure sensors	Ultrasonic flowmeter	Electrical wall plug data loggers	Air volume flow rate measurement
0.5K	0.15 % of the actual measurement	2.5 Pa	3 % of the actual measurement	1 % of the actual measurement	5 % of the actual measurement

5 EXPERIMENTAL RESULTS AND HEAT PUMP MODEL VALIDATION

Parameters of the heat pump model described in section 3 are calibrated using experimental data obtained with the test bench described in section 4. The parameters of the compressor and the heat exchangers are tuned to fit experimental data. One experimental test has been carried out. The tank has been heated with the exhaust air heat pump from 20°C to 60°C. During the test, the air flow rate at the evaporator, the water flow rate at the condenser, the air temperature and relative humidity were respectively equals to 215 m³/h, 400 l/h, 22°C and 50%.

5.1 Compressor model calibration

Figure 3 shows the experimental results for the volumetric and isentropic efficiencies of the compressor as a function of the pressure ratio (black lines). The refrigerant mass flow rate, used to determine the two efficiencies, is not directly measured because it is too small to get sufficient accuracy. The mass flow rate is determined using heating balances on the water and the refrigerant sides of the condenser.
The volumetric efficiency varies from 0.9 to 0.65 for a pressure ratio varying from 2.5 to 4.5. The isentropic efficiency varies from 0.53 to 0.42. The polynomial approximations (red lines) fit very well the experimental data with coefficients of determination (R^2) of 99.02 % and 97.94 % respectively. The parameters of the polynomial laws are given in the Figure 3.



$$\varepsilon_v = 0.824 + 0.105\, r_p - 0.031\, r_p^2 \qquad \varepsilon_{is} = 0.171 + 0.246\, r_p - 0.042\, r_p^2$$

Figure 3: Polynomial approximations of the compressor volumetric efficiency (left) and isentropic efficiency (right)

5.2 Heat exchanger models calibration

For the heat exchanger models calibration, one water tank load has been carried out. The water inside the tank has been heated from 20°C to 60 °C by using only the exhaust air heat pump. During the test, the water mass flow rate was 400 l/h, the air volume flow rate was 215 m³/h and the air temperature at the supply was 22°C. The global heat transfer coefficients AU_{ev}, AU_{cd} and AU_{tank} for the three exchangers have been calculated using the least squares method. The method consists in minimizing the sum S given by equation (11):

$$S = \sum_{i=1}^{n} \left[y_i - f(x_i, AU_{ev,n}, AU_{cd,n}, AU_{tank,n}) \right]^2 \qquad (11)$$

with n the number of experimental points, y_i the measured COP for point number i and $f(x_i, AU_{ev}, AU_{cd}, AU_{tank})$ the COP calculated by the model for the corresponding experimental inputs x_i. The parameters obtained for the three heat exchangers are given in Table 3.

Table 3: Parameters of the three heat exchangers

AU_{ev}^* [W/K]	AU_{cd}^* [W/K]	AU_{tank}^* [W/K]
73	684	362

*for the following nominal conditions: $\dot{m}_{w,n}$ = 400 l/h and $\dot{V}_{air,n}$ = 215 m³/h

5.3 Model validation

The aim of the heat pump model described in section 3 is to predict the COP of the machine with an acceptable accuracy. Figure 4 shows the compressor electrical consumption and the COP of the machine as a function of the temperature of the water inside the tank. The blue lines represent the experimental results with the corresponding 5 % vertical error bars. The black lines represent the model predictions for the same inputs as those for the experimental tests.

As shown in Figure 4, the model is accurate within 5 % error, which is considered acceptable. As expected, the water temperature inside the tank has a significant influence on the COP of the heat pump. As a result, the set-point temperatures for domestic hot water production and space heating have an important impact on the seasonal performance factor of the exhaust heat pump.

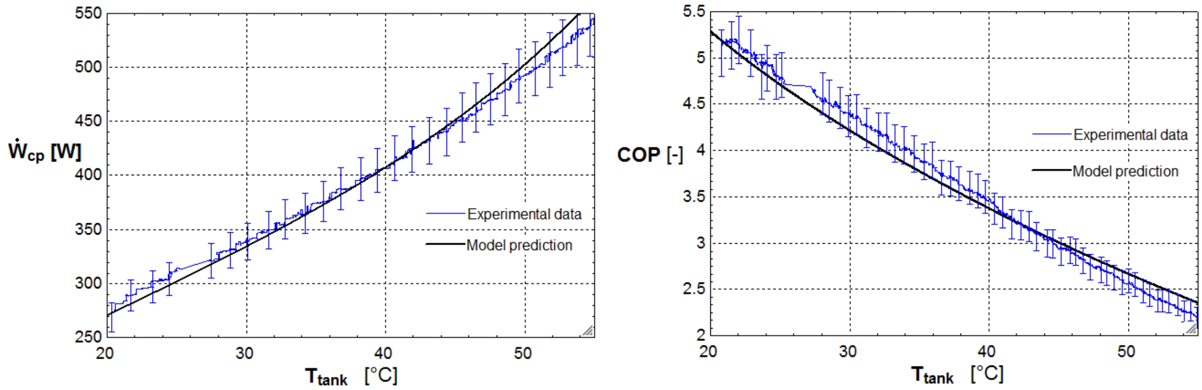


Figure 4: Compressor electrical consumption (left) and COP of the machine (right) (experimental data and model prediction)

6 DETERMINATION OF ANNUAL PERFORMANCES OF THE SYSTEM

In order to determine the annual performances of the system, the above heat pump model is combined with a dynamic building model. Then, the annual performances of the exhaust air heat pump coupled to the simple exhaust mechanical ventilation are compared to the performance of a balanced ventilation system with heat recovery.

6.1 Building model

The building model is the combination of a multizone thermal model and an airflow building model implemented in Modelica. The model is based on electrical analogy (Masy, 2008). For the thermal model, the temperatures and heat flows respectively correspond to electrical potential and current. The different walls of the building, the windows and the doors are modeled with thermal resistances and thermal heating capacities. Internal and solar gains are directly injected at the appropriate temperature nodes. For the airflow building model, the pressures and airflow rates respectively correspond to electrical potential and current. The pressures are imposed by the wind, the buoyancy effect and the fans (for mechanical ventilation). The components of the aeraulic circuit (doors, ducts, extract/supply grilles, etc…) are modeled with non-linear resistances (non-linear relation between pressure drop and mass flow rate). The fans are modeled with polynomial expressions fitted on manufacturer data.

The building model calculates the temperature and the pressure for each zone, and the mass flow rate between these zones. The mass flow rates entering and leaving the building due to the ventilation system and the infiltrations are also determined.

6.2 Case study

The case study, described by Laverge (2013), is a reference flat which represents an average dwelling in Belgium. The total floor area is 120.6 m² and the heated volume is 326 m³. The surface areas for the different rooms of the apartment are shown in Table 4.

Table 4: Surface areas of the different rooms of the flat building for one floor in m²

Bedroom1	Bedroom2	Bedroom3	Living room	Kitchen	Service room	Toilet	Bathroom	Hall
17.1	13.3	16.5	32.5	13.5	7.7	1.6	7.5	10.9

External walls are made of high density bricks (32 cm) insulated with 12 cm of polyurethane corresponding to a U-value of 0.22 W/m²K (K35). Internal walls are made of 20 cm of high density bricks and the floors are made of precast concrete. As a result, the dwelling has a high thermal inertia. Double-glazed windows with a U-value of 1.45 W/m²K and a g-factor of 0.6 are used. The infiltration rate is equal to 2.5 ACH at 50 Pa, corresponding to a typical value for renovated flat (Gendebien et al, 2014). The case study is therefore a typical retrofit apartment building. The nominal ventilation mass flow rates for each room of the building, imposed by the Belgian standard NBN D50-001, are given in the Table 5.

Table 5: Nominal ventilation mass flow rates for each room of the building

	Bedroom1	Bedroom2	Bedroom3	Living room	Kitchen+ Service room	Toilet+ Bathroom
Nominal exhaust airflow rate [m^3/h]	-	-	-	-	158	128
Nominal supply airflow rate [m^3/h]	62	48	59	117	-	-

The set-point temperature is supposed 18°C in all the rooms and 21 °C in the living from 16 PM to 22 PM.
The domestic hot water tank is designed for a typical family of four people. The tank volume is 200 liters and the set-point temperature is 55°C. Domestic hot water profiles from the Annex 42 of the International Energy Agency (Knight, 2007) are used to model the hot water consumption.
Four cases are considered in this study. For all this cases, the ventilation system is supposed to operate continuously and is designed to deliver the ventilation mass flow rates given in Table 5 in nominal conditions. However, the supply and exhaust air flow rates can be influenced by weather conditions (wind, temperature).

- The first case considers a simple exhaust ventilation system coupled with an exhaust air heat pump producing domestic hot water. The heat pump is activated when the water temperature inside the tank is lower than 35 °C and is turned off when the set-point temperature of 55°C is reached. The heat pump can also be used for space heating when domestic hot water production is not necessary. In this case, a fan-coil unit with a supply water temperature of 45 °C is used. If the heat pump heating capacity is not enough compared to the building demand, a 3 kW auxiliary electrical heater is activated. Finally, according to Laverge et al. (2010), the specific fan power (SFP) of the exhaust fan is supposed to be 0.21 W/(m^3-h).
- The second case considers a balanced ventilation system with an air-to-air heat exchanger for heat recovery. This exchanger is supposed to have a constant efficiency of 85 %. According to Laverge et al. (2010), the specific fan power (SFP) of each fan is supposed to be 0.35 W/(m^3-h). For DHW production and space heating, a 5 kW electrical heater with an efficiency of 100% is used.
- The third case considers the same ventilation system than the one in case two, but DHW production and space heating are provided by a 5 kW outside air-water heat pump. The COP of this heat pump depends on the outside temperature. A simplified empirical model based on the Conslim method (Bolher, 1999) using manufacturer data is implemented to calculate the COP. A fan-coil unit with a supply water temperature of 45 °C is also used.
- The fourth case considers the same ventilation system than the one in case two, but DHW production and space heating are provided by a 5 kW gas condensing boiler. A constant boiler efficiency (based on the higher heating value) of 93 % is supposed.

6.3 Numerical results

The four cases are compared in terms of annual primary energy consumption. To convert electricity consumption to primary energy consumption, a primary energy factor (PEF) of 2.5 is used, which corresponds to the power generation efficiency of Belgium. Climate conditions corresponding to Brussels are chosen for the different simulations. Figure 5 shows the annual primary energy consumption for the four cases investigated.

Figure 5: Annual primary energy consumption for the four cases investigated

The simple exhaust ventilation system coupled with an exhaust air heat pump is much more interesting than the balanced ventilation coupled with an electric heater. Indeed, in the case n°1, the seasonal performance factor for the domestic hot water production is 3.27, reducing significantly the primary energy consumption due to DHW production. Moreover, the primary energy consumption due to auxiliary equipments is lower for the simple exhaust ventilation system. In fact, the specific fan consumption is smaller for this case.
The major drawback for the simple exhaust ventilation is the high primary energy consumption due to space heating, compared to balanced ventilation systems. Indeed, the heating capacity of the exhaust air heat pump is limited and is not sufficient to satisfy the heating demand of the building. As a result, the auxiliary electrical heater is activated and the primary energy consumption increases. It would be interesting to use another auxiliary heating system (heat pump, gas boiler) but the investment cost would increase too much.
Compared to balanced ventilation with gas condensing boiler, the simple exhaust ventilation with EAHP has almost the same efficiency in terms of primary energy consumption. Moreover, if the primary energy factor (PEF) is lower than 2.26, the simple exhaust ventilation with EAHP is more efficient than the balanced ventilation with condensing gas boiler. Assuming that the PEF will decrease in the future (with the use of renewable energy sources), the simple exhaust ventilation system will become more interesting.
Finally, compared to balanced ventilation with outside air heat pump, the simple exhaust ventilation with EAHP is less effective (15 % of overconsumption). In fact, the energy consumption due to space heating is too high and is not counterbalance by the energy saving from domestic hot water production and auxiliary equipments. The system could be more effective in the case of a warmer climate corresponding to a lower building heating demand.

7 CONCLUSIONS

The present paper focuses on the energy analysis of a simple exhaust ventilation system coupled to an exhaust air heat pump. The system is compared to the traditional balanced ventilation system with heat recovery in terms of primary energy consumption.
In order to determine the annual performance of the system, a numerical model of the exhaust air heat pump based on experimental data is presented. Despite its simplicity, the model is able to predict the system performance with an error lower than 5 %.
The annual simulations are based on a case study which is a typical retrofitted flat representative of the Belgian building stock. In terms of primary energy consumption, the system is as effective as a balanced ventilation system coupled to a gas condensing boiler for

DHW production and space heating. However, compared to a balanced ventilation system coupled to an outside air heat pump, the system has a primary energy consumption 15 % higher.

8 ACKNOWLEDGEMENTS

The financial support of the Walloon Region of Belgium to this project is gratefully acknowledged.

9 REFERENCES

Bolher A., Casari R., Fleury E., Marchio D. and Millet J. 1999. *Méthode de calcul des consommations d'énergie des bâtiments climatisés "consoclim"*. Ecole des Mines (Paris).

Berg, J.M., Ruud, H.S, Lindberg, U., Stenlund, M. and Axell, M. 2010. *Exhaust Air Heat Pumps—Results from a Field Study in Sweden.*

Cuevas C., Lebrun J., Lemort V., Winandy E. 2010. *Characterization of a scroll compressor under extended operating conditions.* Applied Thermal Engineering, 30, pp 605–615.

El Fouih, Y., Stabat, P., Rivière, P., Hoang, P., Archambault, V. 2012) *Adequacy of air-to-air heat recovery ventilation system applied in low energy buildings*. Energy and Buildings, 54, pp. 29-39.

European Parliament, Council of the European Union (2012). *Directive 2012/27/EU on Energy Efficiency, Amending Directives 2009/125/EC and 2010/30/EU and Repealing Directives 2004/8/EC and 2006/32/EC.* http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2012:315:0001:0056:EN:PDF

Fehrm M., Reiners W., Ungemach M. 2002. *Exhaust air heat recovery in buildings.* International Journal of Refrigeration 25, pp 439–449.

Fracastoro.V, Serraino.M. 2010. *Energy analyses of buildings equipped with exhaust air heat pumps (EAHP).* Energy and Buildings, 42 (8), pp 1283–1289.

Gendebien S., Georges E., bertagnolio S., Lemort V. 2014. *Methodology to characterize a residential building stock using a bottom-up approach: a case study applied to Belgium.* International Journal of Sustainable Energy Planning and Management, 04, pp 71–88.

Knight I. 2007. *European and Canadian non-HVAC electric and DHW load profiles for use in simulating the performance of residential cogeneration systems.* Annex 42 of the International Energy Agency.

Laverge, J., Janssens, A. 2012. *Heat recovery ventilation operation traded off against natural and simple exhaust ventilation in Europe by primary energy factor, carbon dioxide emission, household consumer price and exergy.* Energy and Buildings 50, pp 315–323.

Laverge, J. 2013. *Optimization of design flow rates and component sizing for residential ventilation.* Building and Environment**,** 65, pp 81–89.

Masy G. 2008. *Definition and validation of a simplified multi-zone dynamic building model connected to heating system and HVAC unit*. Ph.D. thesis, University of Liege.

Mikola A., Kõiv T-A. 2014. *The Efficiency Analysis of the Exhaust Air Heat Pump System*.

Orme M. 2001. *Estimates of the energy impact of ventilation and financial expenditures.* Energy and Buildings, 33 (3), pp. 199–205

Ouadha A., En-nacer M., Imine I. 2008. *Thermodynamic modelling of a water-to-water heat pump using propane as refrigerant.* International Journal of Exergy, 5, No. 4.

Roulet C.A., Heidt F.D., Foradini F., Pibiri M.C. 2001. *Real heat recovery with air handling units*. Energy and Buildings, 33, pp. 495–502.

Wenhua L. 2013. *Simplified steady-state modeling for variable speed compressor.* Applied Thermal Engineering, 50, pp 318-326.

VENTILATION PERFORMANCE AND INDOOR AIR POLLUTANTS DIAGNOSIS IN 21 FRENCH LOW ENERGY HOMES

Gaëlle Guyot[1,*], Adeline Bailly[1], Anne-Marie Bernard[2], Gabrielle Perez[2], Claire-Sophie Coeudevez[3], Suzanne Déoux[3], Sandra Berlin[3], Enora Parent[1], Alexis Huet[1], Sylvain Berthault[1], Romuald Jobert[1], Damien Labaume[4], Guillaume Ferrier[5], Sandrine Justet[5]

[*] *Corresponding email:*

1 Cerema
25 avenue François Mitterrand CS 92803
69674 Bron Cedex, France
**Corresponding author: gaelle.guyot@cerema.fr*

2 Allie'Air
4 Clos Ballet, Immeuble La Fenière
01800 Meximieux, France

3 MEDIECO Conseil & Formation
355 Allée Jacques Monod
69800 Saint Priest, France

4 ALDES aéraulique
16 av Charles de Gaulle Bat 14
31130 Balma, France

5 SGS Multilab
10 allée Irène Joliot Curie
69800 St Priest, France

ABSTRACT

Ventilation's historical goal has been to ensure sufficient air change rates in buildings from a hygienic point of view. Regarding its potential impact on energy consumption, ventilation is being reconsidered today. An important challenge for low energy buildings lies in the need to master airflows through the building envelope. Data collected from controls in 1287 recent dwellings shows us that 68 % of the dwellings don't respect the French airing regulation. In this context, actors in the building's sector are reflecting on the risk of an unhealthy indoor air environment in this generation of high performance airtight dwellings.

"VIA-Qualité" project proposes to develop quality management (QM) approaches (ISO 9001) with the goal of increasing both on-site ventilation and indoor air quality in low energy houses.

The first step in this project consists in a campaign on 21 low-energy houses. First, we carried out an aeraulic measurement campaign on every dwelling including visual survey, airflow or pressure at the air-vents, ventilation duct airleakage, and acoustics measurements. Then, we selected 10 dwellings to carry out an indoor air quality winter campaign. This campaign included outdoor and indoor measurements of temperature, relative humidity, carbon dioxide, chemical pollutants (VOC, aldehydes). Inhabitant habits and building influence were studied through a complementary questionnaire survey.

With this campaign, we were able to identify ventilation dysfunctions and to compare them with the national statistics on dwelling stock. The second part of the measurement campaign allowed us also to identify links between aeraulic diagnoses and indoor pollutant measurements.

KEYWORDS:

Ventilation, indoor air quality, measurements, performance, low energy homes

1. INTRODUCTION

In order to insure a good indoor air quality, including a proper humidity level in buildings, adequate air change rates are necessary. On the other side, building energy performance requires to rethink of ventilation and air change rates, because of their impact on thermal losses: 1) New ventilation systems technologies, such as Demand-Controlled Ventilation (DCV) systems, aim at restricting airflows to the minimum level for healthy buildings; 2) Envelope airtightness treatment becomes essential, especially for low energy dwellings (Erhorn et al., 2008). Indeed, envelope air leakage entails thermal losses, but also modifies theoretical voluntary airflows circuits in building. (Boulanger et al., 2012) confirmed that envelope airtightness drives to a better indoor air quality in low-energy buildings, thanks to a better mastering of the theoretical airflows circuits in buildings.

In France, the recent thermal regulation (RT2012) generalizes low energy dwellings and imposes envelope airtightness requirement for any new dwellings. For a single-family dwelling, the requirement is Q_{4Pa_Surf}=0.6 $m^3.h^{-1}.m^{-2}$, that is around n_{50}=2.3 h^{-1}. This energy performance-regulation does not include any new requirement on ventilation rates. Dwellings airing is concerned by another 30 years old regulation (JO, 1982).

The “VIA-Qualité” project (2013-2016) focuses on low energy houses. As a first step, and in order to evaluate the ventilation system efficiency, we analysed available data from regulation compliance controls, related to several regulations, including energy performance (RT2005 & 2012) and dwelling airing (1982-1983). Dysfunctions analysis of a 1287 dwellings sample allows us to establish an accurate picture of on-site ventilation systems quality (Jobert and Guyot, 2014): 47% of the sample, do not comply with the airing regulation. The non-compliance rate is 68% for single-family dwellings, and 44% for multi-family dwellings. This analysis confirms others in Europe (Boersta, 2012; Caillou, 2012), which observe that in-site ventilation system mounting is often far from the hoped performance.

In this context, building’s sector wonders about the risk for a generation of performing airtight dwellings to contribute to an unhealthy indoor air.

This project proposes developing quality management (QM) approaches (ISO 9001) with the goal of increasing both on-site ventilation and indoor air quality (IAQ). Such QM approaches, when applied to the individual homebuilder sector, appear to be promising. In France, the individual homebuilders sector accounts for the majority of new single-family dwellings. The benefits would be to: 1- Improve ventilation system performance, especially thanks to rigorous monitoring from conception to installation; 2- Limit indoor internal pollution sources, monitoring materials selection (Wargocki and Hartmann, 2012); 3- Increase final users awareness and understanding. Individual homebuilders, with respect to an annex of the French EP-regulation, are already using such QM approaches in envelope airtightness field. Feedback from these experiences shows that such approaches are both successful and affordable for either small or large individual homebuilder (Carrié R. et al, 2007) (Charrier S. et al, 2014).

As data on classical low-energy dwellings were missing, we have also conducted an original campaign on ventilation performance and IAQ in 21 low-energy dwellings representative for French buildings stock.

First, the paper presents quickly the methodologies used in this campaign of the "VIA-Qualité" project. Then, we present results and analysis of observed dysfunctions, leading to some proposals for ventilation installation improvement. Then, we present briefly IAQ results and analysis. Finally, we conclude and present briefly the follow-up of the campaign, and of the project.

2. METHODS

We have selected 21 low-energy houses, representative for classical French new low-energy houses This sample includes 7 balanced ventilation with heat recovery systems and 14 humidity demand controlled ventilation (DCV) systems. On the whole sample, we conducted first ventilation performance measurements. Then, we selected 10 dwellings, with various kind and importance of non-compliance from a "ventilation point-of-view", to carry out an indoor air quality winter campaign. 8 of them have balanced ventilation with heat recovery, as there are only few feedbacks on this type of ventilation, and 2 have humidity demand controlled ventilation (DCV).

As a first step of this campaign, the ventilation performance evaluation consists in (Table 1): ventilation ducts leakage measurements, airflows (if balanced ventilation) or pressure differences (if DCV) at air outlets measurements, airflows at air supply (if balanced ventilation), noise level of the ventilation equipment measurements, and visual diagnostic of the installation regarding its compliance with French dwellings airing regulation (JO, 1982). We also perform envelope airleakage measurement, in the rare cases when such measurement reports are not available. The airing regulation requires general and continuous ventilation and describes the compulsory general layouts of ventilation installation. It also sets exhaust airflows in each humid room, depending on the total number of rooms in the dwelling. Total airflows drive to around 0.5 h^{-1} global air change rate in the dwelling. This regulation has been modified in 1983 in order to reduce these airflows in case of DCV system, for instance based on humidity, which are very often used in France. In this case, controls include also additional specific technical guidelines.

Tolerance margins for airflows and pressure measurements at out- and inlets are: ± 3 m^3/h for target airflows under 15 m^3/h; ± 5 m^3/h for target airflows over 15 m^3/h; measured pressure must be included in the [P_{min}-5Pa; P_{max}+5Pa] interval. Where P_{min} and P_{max} are respectively the minimum and the maximum working pressures of the outlet given by the manufacturer.

Total airflow in a dwelling is calculated summing measured airflows at outlets. In case of DCV system, we use the following formula to estimate the relative error (□Q) to the minimum regulatory total airflow (Q_{reg}), starting from measured pressures at the N outlets (Pi).

$$\delta Q = \frac{\Delta Q}{Q_{reg}} = \left(\frac{\sum_{i=1}^{N} P_i}{N * P_{min}}\right)^{0.5} - 1 \qquad (1)$$

As a result, we define 3 categories of houses: 1) "Houses with regulatory airflow" have total measured airflow between Q_{reg} and 1.3*Q_{reg} ; 2) "Houses with over-ventilation" have total measured airflow over 1.3*Q_{reg}; 3) "Houses with under-ventilation" have total measured airflow under Q_{reg}.

As a second step of this campaign, the IAQ evaluation consists in: indoor and outdoor pollutants measurements, hygrothermal and air stuffiness parameters measurements, and questionnaire surveys. The IAQ evaluation takes place during the winter 2014. The selected pollutants for the measurements (Table 2) are considered as priority for IAQ according to the French Indoor Air Quality Observatory (OQAI, 2013). During 7 days, 16 VOC, 8 aldehydes, NO_2, CO (if there is a source of combustion in the dwelling), PM 2.5, CO_2, temperature and relative humidity are measured in the living room and in a bedroom. Radon is measured during two months. VOC and NO_2 are also measured outdoor.

The questionnaire surveys (OQAI, 2013) aims at describing the houses, inhabitants habits, and their feeling about their inside environment. The influence of these factors on IAQ is taken into account in the quality management approached developed in the "VIA-Qualité" project. The data collected in the questionnaire describing the houses are about the outside environment, the construction materials, the heating and ventilation systems, the decoration materials, furniture, etc. The habits of the inhabitants regarding heating, ventilation, cleaning, smoking, etc. that influence IAQ are collected. There is also a perceptive questionnaire about acoustic, visual, olfactory and thermal comfort of the inhabitants. A feedback on the difficulties to use the heating and ventilation systems is also asked.

Table 1. Methods and equipment used in our ventilation performance diagnostics

Parameters	Equipment	Reference document
Airflow at air terminal device	Hood with hot wire or propeller manometer	prEN16211
Pressure difference at air outlet	Manometer	Guide CETIAT (Caré, 2012)
Standardized acoustic pressure level	Sound meter	ISO NF EN 10052
Duct air leakage	Duct leakage tester	FD E51-767, EN12237…
Envelope air leakage	Fan pressurization method (Blowerdoor, Permeascope)	NF EN 13829, GA P 50-784
General layouts of installation	Visual inspection	Airing regulation (JO, 1982) Label Protocole (Effinergie, 2013)

Table 2. Measurement and analysis methods and equipment for IAQ evaluation

Pollutants and parameters measured	Equipment for measurement and lab analysis	Standard used
Air temperature	Thermometer	ISO 7726
Relative Humidity	Hygrometer	
CO_2	Logger using non-dispersive infrared sensor	PR NF EN ISO 16000-26
VOC: benzene, toluene, ethylbenzene, (m+p) xylene, styrene, tetrachloroethylene, 1-methoxy-2-propanol, limonene, n-hexane, trichloroethylene, 1,4-dichlorobenzene, n-undecane, 2-butoxyethanol, alpha-pinene, o-xylène , 1,2,4-trimethylbenzene.	Radiello passive (diffusive) samplers by thermal adsorption Gas chromatography and mass spectrometry	ISO 16017-2
Aldehydes: formaldehyde, acetaldehyde, hexaldehyde, benzaldehyde, butyraldehyde, valeraldehyde, propionaldehyde, acroleine.	Radiello passive (diffusive) samplers by reaction with 2-4 DNPH and liquid chromatography and UV detection	ISO 16000-4
NO_2	Passam AG passive diffusive sampler and spectrophotometry	NF X43-009
CO	Electrochemical sensor and data logger with an alert if passing threshold (Drager PAC III or PAC 7000 or equivalent)	NF EN 14626
PM 2,5	Active air sampling on a filter with a pump	NF EN 14907
Radon	Passive sampling with Alpha track detection	ISO 11665-4

3. RESULTS AND DISCUSSION

With the first campaign, we are able to identify ventilation dysfunctions and to compare them with the national statistics. No house is found to comply fully with the ventilation regulation, 2 of them with minor non-compliances. The main non-compliance points are given in Figure 1a classified by type of the installation concerned. We can note than there are generally several non-compliance on the same installation (average 3 per inspected house). As a result, we can note that 81% of the houses don’t reach the minimum required total airflows (standard running mode), including 82% under-ventilation rates and 18% over- ventilation rates. Figure 1b shows the relative gap of the measured total airflows to the minimum required airflows, by compliance category. For over-ventilated houses (3 cases), the relative gap median value is +39%; for regulatory ventilated houses (4 cases) it is +22%, for under-ventilated houses (14 cases) it is -45%, with a maximum value of -73%.

100% of tested houses (7 cases) do not comply with the boost mode required in kitchen (when cooking), even those over-ventilating have difficulties achieving the maximum flow. Figure 2 shows that around 50% of the ducts, including exhaust- and supply duct, have an airleakage class greater than or equal to 3 times the class A, which is the worst class according to EN 14239 (CEN, 2004).

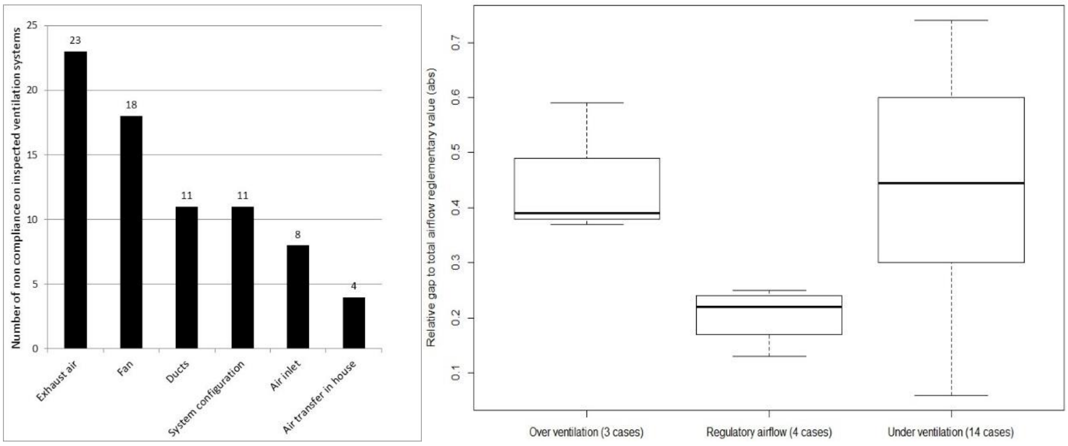


Figure 1. Non compliances on inspected ventilation systems. a) Number by type, b) Measured total airflows: relative gap to regulatory values according to the compliance category

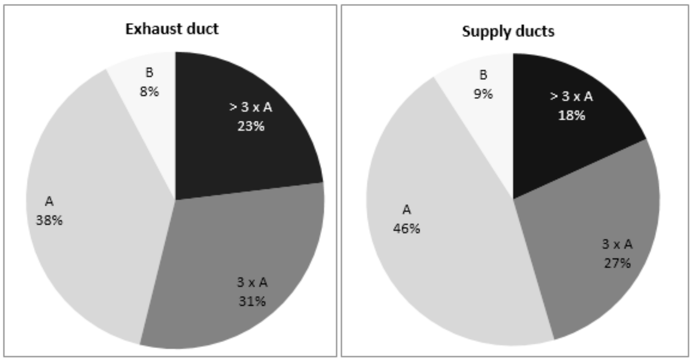


Figure 2. Duct air-tightness on: a) exhaust, b) supply ducts

It is therefore essential to check why these airflows are not obtained. We have classified, for each installation, the main non-compliance leading on these installations to a poor compliance of airflows (Figure 3). Firstly, the lack of flow control devices is recurrent and doesn't allow achieving a correct airflow in kitchen even when the fan is boosted, flow controls are mandatory in the French thermal regulation as well as in airing one. Secondly, ducts are the main cause of problem mainly through: poor air tightness loosing flow, too high pressure drop, causing lack of airflows at some air terminals, air exhaust ducts too small, sometimes not even connected or connected to roof outlet too small creating too much pressure drop. Thirdly, poor fan connections create too much pressure drop and a lack of access. And finally, in one case, the large outdoor pollution fills in the filters (no prefilter, F7 alone) and reduces the airflow to zero after 2 months, in widely polluted area, a specific care to filtration is needed.

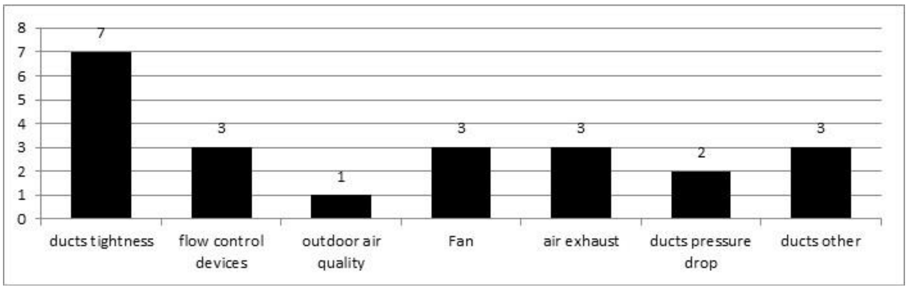


Figure 3. Main causes of non-compliance on airflows

These defaults are very recurrent and can be solved easily. Pressure drop is not calculated in most cases in individual houses because it used to be generally simple in single exhaust systems and to save time. The QM approach will have to recommend a pressure drop calculations on systems installed, mainly for supply and exhaust systems, but also for single exhaust if used components sizing are not strictly checked. Most other defaults are linked with incorrect practice in mounting and assembling the ventilation system. Training is a large part of the QM approach and will be addressed through specific guidelines in the next steps of "VIA-Qualité" project. Solving this kind of defaults should not increase the cost of the installation and need to be done because we can guess there will be a large impact on IAQ results and thermal performance (ie : 14% of dwellings over-ventilating, sometimes up to 3 times the mandatory airflow is a waste of energy to be considered).

The second measurement campaign allows us also to analyze links between ventilation diagnosis and indoor pollutant measurements. Firstly, we observe that the formaldehyde median value of the 20 measurement points is 17.4 µg/m^3, lightly under 19.6 µg/m^3, the median value measured during the national campaign on dwellings (OQAI, 2003-2005). In addition, only bedrooms were measured during this national campaign. If we would compare more precisely, we should also use the median value of the 10 measurement points in the bedrooms, which is even lower: 17.25 µg/m^3.

An interesting result concerns links between ventilation level and VOCs and aldehyde concentrations (Figure 4): The three over ventilated houses have lower concentrations while in the 2 regulatory houses and in the 5 under ventilated houses results are diverse. This observation is balanced by the fact that IAQ is connected to ventilation quality, but also to building materials, decorations, and furnishings, to occupant behaviors, to outdoor environment.

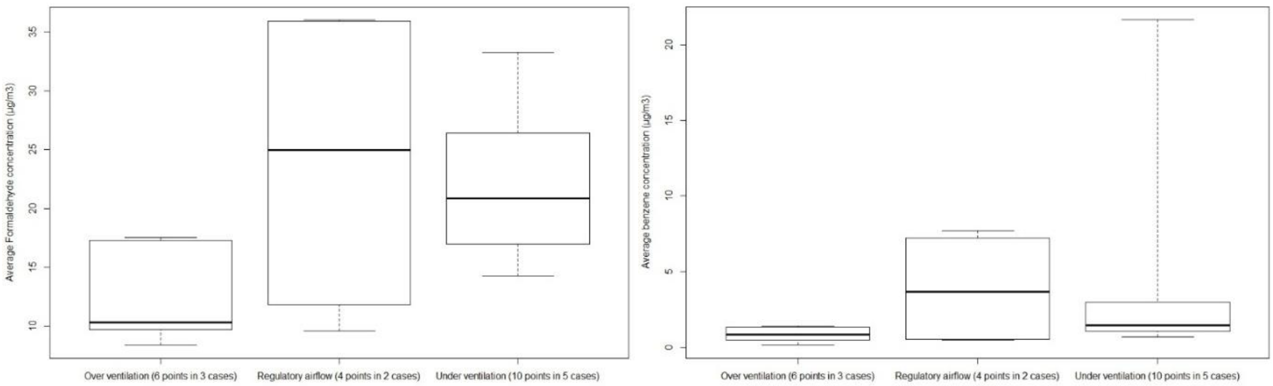


Figure 4. Average concentration distributions, according to the compliance category for: a) Formaldehyde, b) benzene.

Then, it is clear with CO_2 concentrations analysis (Figure 5) that problems in bedrooms occurs when the ventilation operates improperly (insufficient rates, or doors not correctly undercut). Case n°2 shows that even the global airflow is correct, bad doors undercuts can explain high CO_2 concentrations especially in main bedroom. We consider the limit of 1000 ppm (JO, 1978), the number of hours in a week with concentration higher than 1000 ppm, and the average of the 60 highest values (OQAI, 2006) as a performance indicator.

Then, we observe that the presence of an attached garage with access to the house can explain a part of high BTEX concentrations, but it is not a general explanation (Figure 6).

In no house, relative humidity median value is higher than 60%, and no fungal contamination is observed.

Finally, a very critical case is underlined with high CO concentration in the presence of ventilation- and wood stove-dysfunctions.

We observe no specific difference between balanced and single exhaust systems, even if our sample is very little.

House n°	CO_2 (ppm) Average value of the 60 highest values		Number of hours per week > 1 000 ppm in bedroom	Ventilation Performance			
	Living room	Bedroom		Base airflow % of regulatory airflow	Doors with no undercut	Duct air leakage (NF EN 12237)	Balancing between exhaust and supply
1	1 442	2 210	12,5 h	- 35 %	Compliant	3A	0,40 %
2	1 130	2 177	85 h	Compliant	No compliant	not measured	N/A
3	717	1 062	15,5 h	- 50 %	Compliant	3A	13 %
4	1 111	2 351	47 h	- 40 %	Compliant	not measured	N/A
5	642			+ 37 %	No compliant	3A	23,10 %
6	734	728	0 h	+ 39 %	Compliant	3A	not measured
7	875	1 503	22 h	- 74 %	Compliant	3A	48 %
8	979	1 666	57 h	- 13 %	Compliant	A	26 %
9	661	993	1 h	+ 59 %	Compliant	A	5 %
10	563	1 044	5 h	Compliant	Compliant	3A	0,30 %

Figure 5. Analysis of carbon dioxide concentration regarding ventilation dysfunctions

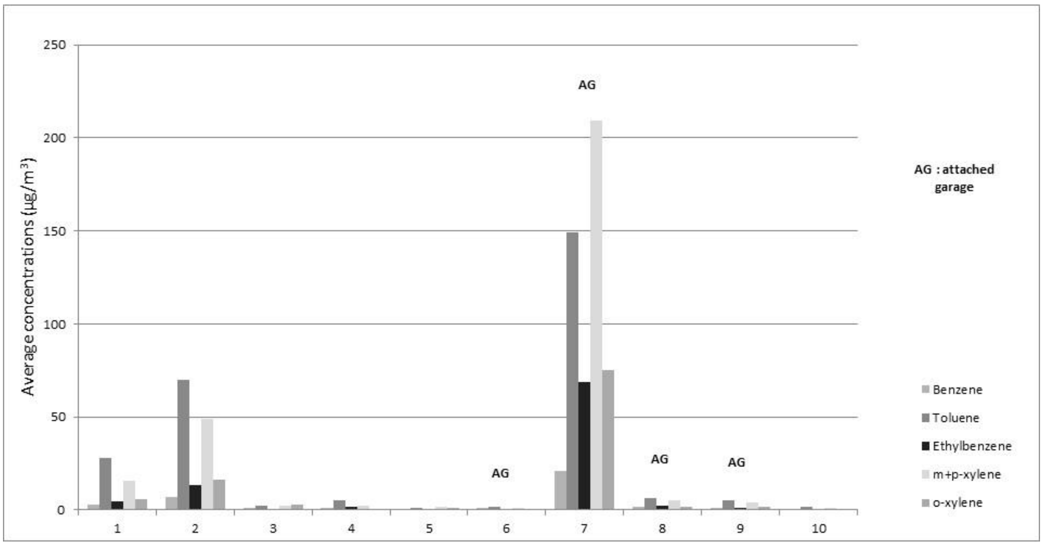


Figure 6. BTEX concentrations in living rooms of the 10 houses, regarding the presence of an attached garage

4. CONCLUSIONS

With this campaign on 21 low-energy single-family dwellings, we would like to produce precise crossed data on: 1-quality of in-site ventilation and 2-pollutants in indoor air of low energy single-family dwellings, representative for French new low-energy houses.

With the first campaign, we were able to identify ventilation dysfunctions and to compare them with the national statistics. No house was found to comply fully with the ventilation regulation, 2 of them with minor non-compliances. Concerning the ventilation rates, 81% of the dwellings do not respect requirements, including 82% over-ventilation rates and 18% under ventilation rates.

The second measurement campaign allowed us also to analyze links between ventilation diagnosis and indoor pollutant measurements. First results are: Pollutant concentrations are slightly under the median values measured during the national campaign on dwellings (OQAI, 2003-2005) for formaldehyde even if they stay under the French guide value of 30 µg/m^3; Over ventilated house have lower VOCs and aldehydes concentrations; In regulatory and under ventilated houses results are diverse; No specific difference is noted between balanced and single exhaust systems. A critical case was underlined with high CO concentration in the presence of ventilation- and wood stove-dysfunctions.

This analysis confirms that, even if adapted industrial solutions are available, ventilation system dysfunctions are very frequently observed in low-energy dwellings like in every dwelling (Jobert and Guyot 2013), which entails the reliability of these installations.

The main stake now consists in developing tools leading to better practices at every stage of the construction. In this project, we want to show that many dysfunctions could be avoided through the implementation of quality management tools. With such tools, one could pretty easily, not costly, but efficiently, control ventilation system at each stage of the building construction: from design to installation, even including maintenance and final use. We could also manage and limit indoor air pollutants.

In the VIA-Qualité project, we are now testing these tools with the construction of 8 pilot projects of low energy houses, thanks to two builder’s partners. We are going to measure ventilation and IAQ performances at commissioning and during the 6 first months for IAQ on 2 houses.

5. ACKNOWLEDGEMENT

The “VIA-Qualité” project has received funding from the French Environment and Energy Management Agency under the contract ADEME/1304CO014 and from the Rhône-Alpes region. The contribution of CEREMA is funded by the French ministry for ecology, sustainable development, and energy (MEDDE).
The sole responsibility for the content of this publication lies with the authors.

6. REFERENCES

Boersta A (2012) *Residential ventilation system performance: outcomes of a field study in the Netherlands*. Conference paper, AIVC, Tightvent, Copenhague, Denmark, 7 p.

Boulanger X et al (2012*) Lessons learned on ventilation systems from the IAQ calculations on tight energy performant buildings*. 33th AIVC Conference, Copenhagen, Denmark, 7 p.

Caré I, Chaffois P, Henry P (2012) *Guide de bonnes pratiques des mesures de débit d’air sur site pour les installateurs de ventilation*. CETIAT, www.cetiat.fr, France, 49 p.

CEN (2004) EN 14239:2004. *Ventilation for buildings. Ductwork. Measurement of ductwork surface area*

Charrier, S., Ponthieux, J., Huet, A. (2014). The Airtightness Quality Management Approach in France – Assessment after more than Five Years of Operation. *International Journal of Ventilation*. ISSN 1473-3315. Vol. 13(2), pp. 125-140.

Caillou S (2012) *Performances of ventilation systems: on site measurements related to energy efficiency, comfort and health*. 33th AIVC Conference, Copenhagen, Denmark, 7 p.

Carrié R et al. (2007) *Performance de la ventilation et du bâti. Rapport final phase 1 et 2*, Association air.h, France, 109 p.

Effinergie (2013) *Protocole de contrôle des systèmes de ventilation des bâtiments demandant le label Effinergie +*, France, www.effinergie.org, 12p.

Erhorn H, Erhorn-Kluttig H, Carrié F.R (2008) *Airtightness requirements for high-performance buildings*. Conference paper, AIVC, Kyoto, Japan, 6 p.

JO (1978), *Circulaire du 9 août 1978 relative à la révision du règlement sanitaire départemental* (RSDT)

JO (1982), *Arrêté du 24 mars 1982 consolidée relatif à l'aération des logements : aération générale ou permanente*. JO du 15 novembre 1983.

Jobert R, Guyot G (2013) *Detailed analysis of regulatory compliance controls of 1287 dwellings ventilation systems*, 34th AIVC Conference, Athens, Greece, 2013, 10 p.

OQAI (2006), *Campagne nationale logements : état de la qualité de l'air dans les logements français rapport final*. Novembre 2006, mis à jour mai 2007. DDD/SB 2006-57. 183p.

OQAI (2013), *Base de référence sur la qualité de l'air intérieur, le confort et les consommations énergétiques des bâtiments performants en énergie – stratégie générale de collecte des données*, Protocole n°OQAI-CSTB-2012-106, février 2013, 31p.

Wargocki P, Hartmann T (2012) *Health-Based ventilation Guidelines for Europe. HealthVent project*, 33th AIVC Conference, Copenhagen, Denmark.

REVIEWING LEGAL FRAMEWORK AND PERFORMANCE ASSESSMENT TOOLS FOR RESIDENTIAL VENTILATION SYSTEMS

Rob C.A. van Holsteijn[1], Bas Knoll[*2], Harm J.J. Valk[3], and Marco C. Hofman[4]

1 VHK BV
Project manager MONICAIR
Elektronicaweg 14
2628 XG Delft, Netherlands
e-mail address: r.van.holsteijn@vhk.nl

2 TNO Delft
Researcher Energy, Comfort & IAQ
van Mourik Broekmanweg 6,
2628 XE Delft, Netherlands

3 Nieman Raadgevende Ingenieurs
Senior Consultant
Atoomweg 400
3542 AB Utrecht, Netherlands

4 ISSO, Dutch Buildings Services Institute
Chair National Standard Committee on Ventilation
Kruisplein 25
3014 DB Rotterdam, Netherlands

ABSTRACT

The field research project MONICAIR indicates that ventilation systems that fully comply with Dutch building codes show large differences in their IAQ-performance in habitable rooms during heating season and do not always achieve acceptable IAQ-levels [lit.1]. The results indicate that there are considerable differences in the actually achieved air-exchange rates per person during presence in habitable rooms. System averages on CO2-excess doses per heating season (an indicator for the duration and the amount of the excess above 1200 ppm CO2) vary from 68 to 349 kppmh per person. This roughly corresponds to situations in which either 7% or 35% of the time spent at home ventilation rates per person are insufficient. Variations between individual dwellings are even bigger with values ranging from 0 to 853 kppmh per person per heating season. These differences in habitable rooms occur despite the fact that the overall ventilation rates for all dwellings investigated are well above 10 l/s per person.

Concerning the RH-levels, only limited exceedance of threshold values were monitored. Generally these levels are well below 70% in all rooms, and only in bathrooms RH-values may rise above 70% for a period off - on average - less than two hours per day. Periods with RH-values less than 30% occur in all rooms of all dwellings and last on average 4 to 6 hours per day.

Other recently concluded monitoring studies further substantiate the results from the MONICAIR field research [lit.2,3]. These results point to the conclusion that the assumption that all code compliant ventilation systems perform comparable on IAQ, is clearly not justified. In consequence, the declaration of the Energy performance of ventilation systems has limited meaning as long as a direct link to a properly assessed IAQ-performance is lacking.

Current building codes, test standards and determination methods apparently are not sufficient to ensure the required air exchange rates in habitable rooms and consequently reduce the exposure to human odours (hygienic threshold values) and all other indoor pollutants.
In respect to these findings the MONICAIR consortium reviewed the existing Dutch building codes and related test- and determination method to determine if and where the legal framework and related guidelines can be further refined to ensure a minimal IAQ-performance of ventilation systems.
On several topics the legal framework and related assessment and test tools can be further expanded and refined. But the key item here is the fact that an actual performance requirement is missing, as well as a proper test protocol for assessing the performance of a ventilation system on its primary function "diluting concentrations of indoor pollutants by exchanging air and thus reducing exposure".

KEYWORDS

IAQ-aspects in ventilation regulation, controls and user interaction

1 INTRODUCTION

In the developed countries people spend around 80 to 90% of their time indoors, of which approximately 60% in their own houses [lit.4]. Although some pollutants might originate outdoors, most pollutants that affect Indoor Air Quality (IAQ) come from sources inside the dwelling. Building materials, interior- and decorative products as well as humans and their activities cause emissions, resulting in concentrations of pollutants that are generally considerably higher than the concentrations outside [lit.5]. Exposure to these higher concentrations of indoor pollutants has an effect on people's health. The WHO Guidelines for Indoor Air Quality [lit.6] present guidelines for the protection of public health from risks due to a number of chemicals commonly present in indoor air.

The first and most important step towards the improvement of the IAQ is the reduction of source emissions from building materials, interior products and people's activities. Various guidelines and regulations related to source emissions of building materials and interior products are already in place and regularly revised. The second important mechanism for reducing concentrations of indoor pollutants is *ventilation*, or, the removal and/or dilution of indoor pollutants by replacing polluted indoor air (preferably where is it highly concentrated) with cleaner outdoor air. This not only implies that all in home combustion systems – as far as they don't operate independently from the indoor atmosphere – are effectively vented. It also means that kitchen hoods must comply to minimum pollutants capture efficiencies and bathrooms and utility rooms need to be properly vented to remove moisture. And finally it implies that built-up concentrations of indoor pollutants in habitable rooms are effectively diluted, particularly during presence when exposure occurs.

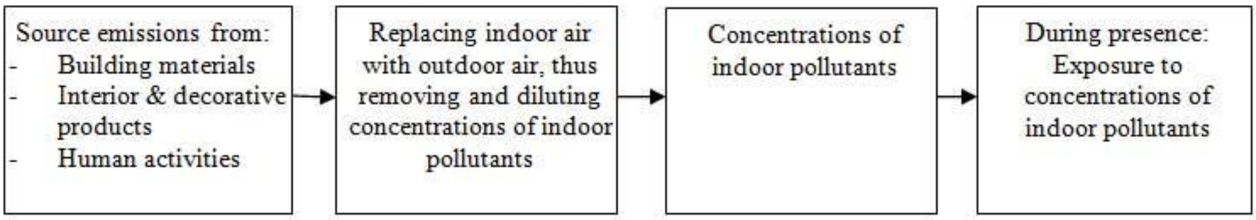


Figure 1. Factors that determine the concentration of and the exposure to indoor pollutants

If we want to determine the IAQ-performance of a ventilation system, we need to assess the performance on its primary function: 'the ability to remove and/or dilute concentrations of indoor pollutants to an acceptable level, by replacing indoor air with cleaner outdoor air, not only in wet rooms, but also in habitable rooms where people spent most of the time en where there is long-term exposure'.
Only by fulfilling this primary function, exposure to concentrations of indoor pollutants can be reduced. Depending on the type of indoor pollutant and the rate of indoor source emissions, air exchange rates of ventilation systems can be further increased to achieve acceptable concentration levels. The prEN 16798-1:2015 [lit.7] lists the following air exchange rates for ventilation systems in residential buildings.

Table 1. Air-exchange rates ventilation systems in residential dwellings

Category	**Air-exchange rates**		
	Ventilation per person	**Air change rate dwelling**	
	l/s/pp	l/s/m2	ach
I	10	0.49	0,7
II	7	0.42	0,6
III	4	0.35	0,5
IV		0.23	0,4

To assess the performance of a ventilation system on its primary function, the MONICAIR consortium – a broad consortium of Dutch ventilation unit manufacturers and research institutes supported by the Dutch government – monitored the real life performance of ten code-compliant ventilation systems in 62 dwellings for over a year. The project not only monitored the relative humidity (RH-) levels in all rooms, but also the CO_2-levels in all habitable rooms. Relative humidity is a valid indicator for the ventilation performance in wet rooms. CO_2-concentrations are generally considered a good indicator for the ventilation performance in habitable rooms. It does not only point out whether hygienic limit values are exceeded or not, it also indicates the actual occurring air exchange rates in relation to the number of people present [lit.8,9].

2 RESULTS MONICAIR FIELD RESEARCH PROJECT

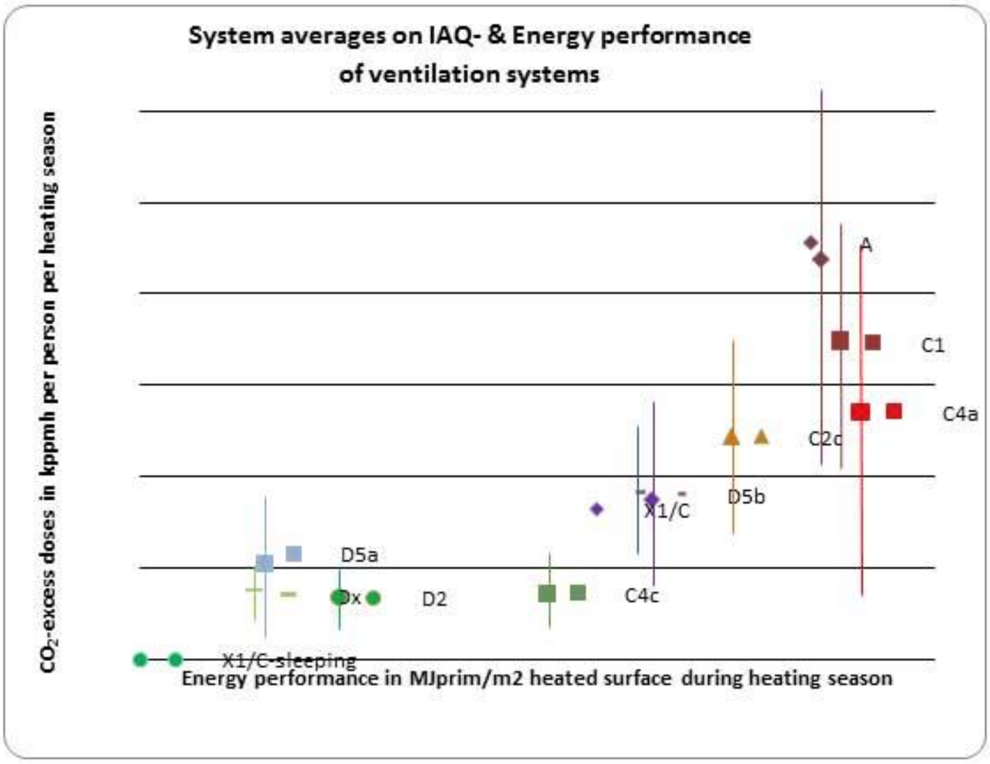
Because the results of the MONCAIR field research were already discussed in lit.1 and lit.10, only a summary is presented here. The code compliant ventilations systems investigated are C-type (or MEV-) systems that apply mechanical extraction in wet rooms and natural supply in habitable rooms, except for system type C.4c which applies mechanical extraction in all rooms. The D-type (or MVHR-) systems are ventilation systems with heat recovery that use both mechanical extraction from wet rooms en mechanical supply for the habitable rooms. X1-type systems use local MVHR in the living section and MEV or fully natural systems in the sleeping section (see also lit.1 for further explanation of ventilation system types). The results on RH-levels are presented in table 2. With the exception of a few houses were the natural exhaust provision in the bathroom is inadequate, all systems in all dwellings show fairly good results on achieved RH-levels, indicating a sufficient ventilation performance in the wet rooms (please note that most of the kitchens in the monitored dwellings have dedicated cooker hoods). Occupant behaviour like leaving the bathroom door open or opening a window after showering may have contributed to these results.

Table 2. System averages on hours per day RH-limits are exceeded during heating season

ventilation system type		average No. of hours per day RH> 70%				average No. of hours per day RH< 30%			
		Total of all rooms	bathroom	kitchen	per indiv. hab.room	Total of all rooms	bathroom	kitchen	per indiv. hab.room
		[h/day]	[h/day]	[h/day]	[h/day]	[h/day]	[h/day]	[h/day]	[h/day]
	A	1.47	1.46	0.01	0.00	27.26	5.02	5.44	4.42
MEV-systems	C1	1.31	1.31	0.00	0.00	49.53	10.10	7.77	8.56
	C.2c	1.09	0.60	0.00	0.11	20.03	10.37	n.a.	2.25
	C.4a	0.29	0.29	0.00	0.00	29.70	10.48	2.73	4.34
	C.4c	1.32	1.32	0.00	0.00	17.13	2.80	n.a.	4.09
MVHR systems	D.2	1.52	0.88	0.00	0.16	23.02	5.18	n.a.	4.46
	D.5a	0.54	0.54	0.00	0.00	47.69	10.09	n.a.	8.55
	D.x	2.78	2.78	0.00	0.00	18.33	5.33	n.a.	3.25
	D.5b	0.35	0.35	0.00	0.00	33.15	7.68	9.23	5.41
Comb.	X1/C	1.53	1.52	0.01	0.00	25.80	3.20	5.00	4.40
	X1/A	7.35	7.35	0.00	0.00	26.60	5.90	2.70	6.00

The results on CO_2-concentrations in habitable rooms and on energy consumption are presented in figure 2. In the graph the diamond markers represent system averages on CO_2-excess dose per person (product of duration and amount of excess above 1200 ppm during heating season). The vertical lines represent the standard deviation of a group of ventilation systems (both refer to the vertical axis of the graph). On the horizontal axis the system averages for energy consumption of ventilation systems in total primary energy per m^2 heated surface during heating season is indicated. The standard deviation on energy consumption is limited and therefore not displayed in the graph (see lit.1 for further explanation of ventilation system types).

Figure 2. System averages and standard deviation on CO_2 excess dose and Energy performance



Apart from ventilation system A (which was added at the request of housing associations because they represent a significant share of the housing stock) all other ventilation systems are code compliant. The large differences both in CO_2 excess dose per person and in the standard deviation illustrates that there are considerable variations in the IAQ-performance of the code compliant ventilations systems in habitable rooms. The data illustrates whether the ventilation systems were capable of achieving the requested air exchange rates per person in the various habitable rooms and consequently dilute all concentrations of indoor pollutants. Although periods of insufficient air exchange were observed in all habitable rooms, main problems occur in the bedrooms. The large extent of these differences in achieved air exchange rates were not fully anticipated, especially given the fact that all ventilation systems achieved overall air exchange rates well above 10 l/s per person, which complies with the highest IAQ category in table 1.
Ventilation systems type D (MVHR-systems) vary in their IAQ-performance, mainly due to the fact that some units or individual fans are temporary switched off because of noise or draught. Type C (or MEV) systems show large variations due to the fact that some of these systems have insufficient control over the air exchange in the habitable rooms. Some of these MEV systems require frequent interventions from active occupants and the monitoring results show that this type of behaviour was not exhibited. Unlike with temperature, light and noise, people neither possess the requested sensory capacities to perceive the IAQ, nor the expertise to properly control these ventilation system with their complex interactions between window- and door openings, fluctuating natural buoyancy and pressure differences over facades due to varying weather conditions.

Apart from the fact that various indoor pollutants are odourless (radon, CO, various VOC’s), people also adapt to their IAQ. Although some large and sudden changes in concentrations of pollutants might be notices, occupants at least need to be awake to observe this which obviously is not always the case. Therefore, the assumption that people are able to judge and manually control the IAQ by correctly operating their ventilation system does not seem appropriate, even if the required operations are easy.
In respect to the energy performance of the investigated ventilation systems, it may be concluded that the real life energy consumption is not in line with the EPBD assessment methods. This is partly caused by differences in the assumed and real life system operation. Apart from that, there are certain types of ventilation systems with CO_2- control, that actually perform considerably worse in real life than according the assessment methods. Overall there are also differences in their relative ranking. And above all the question arises as to what the meaning is of an assessment of the energy performance without a direct link to its IAQ-performance.
Clearly the (implicit) assumption that all code compliant ventilation systems perform comparable cannot be substantiated by these findings. Further refinement of the legal framework and the performance assessment tools is necessary.

3 EXISTING LEGAL FRAMEWORK , ASSESSMENT AND TEST TOOLS

In Work Package WP3a, the MONICAIR consortium reviewed the existing legal framework and performance assessment test tools in the Netherlands, which resulted in the following list of items than can be further improved to ensure a minimal IAQ performance of ventilation systems.

3.1 Actual performance requirement is missing

The leading article in the Dutch building codes related to ventilation systems (article 3.28) states that *"a dwelling must have provisions for air exchange which prevent adverse health impacts"*.
Section 3.6 of the Dutch building codes further describes that all individual rooms need to have such a provision and indicates what air exchange rates these provisions must be able to achieve (values are based on the advice from the Dutch Health Council, demanding that at least 7 l/s per person can be achieved). For the determination method the building codes refer to NEN1087. This Dutch standard describes the conditions for properly dimensioning and implementing ventilation systems and fulfilling requirements.
Although this legal framework apparently is sufficient to ensure that requested ventilation rates are achieved at total dwelling level and (to a considerable extend) in the wet rooms, it does not suffice to ensure the requested air exchange rates in the distinctive habitable rooms. What is missing, is a clear performance requirement describing where and when these air exchange provisions must (at least) operate properly, not only in wet rooms but also in the habitable rooms, where exposure time is the longest.

3.2 Ventilation requirements based on m^2

To enable different lay-outs within a dwelling, Dutch building codes now specify air exchange rates based on the room surface area. For bedrooms this may not always lead to a correct sizing of the air exchange provisions. Bedrooms for two people are generally (at least in the housing stock of both housing associations and a vast majority of privately owned dwellings in Netherlands) smaller than 16 m2. With the ventilation requirements of 0,7 $l/s/m^2$ for habitable rooms, this will lead to insufficient minimum ventilation, even if the air exchange rates are actually achieved. For ventilation systems that have limited control over air exchange rates in habitable rooms, the IAQ will be even worse.

3.3 Requirements outdoor air quality

Ventilation systems can only succeed in their final goal if the outdoor air is sufficiently cleaner than the indoor air. Bearing in mind the fact that the limit values on ambient air quality (Directive 2008/50/EC) are not always and everywhere achieved in the Netherlands, it seems only right to add certain requirements related to the cleaning or filtering of the supplied air. Exposure to particulate matter (PM10 and PM2,5) from traffic and industry for instance, has a large impact on people's health. Moreover, there are large groups of the population that have a predisposition for lung diseases and allergies that will largely benefit from targeted cleaning of the outdoor air before is it supplied into the dwelling.

3.4 Ventilation system control based on human intervention

The legal framework and performance assessment procedures are partly based on the assumption that people are capable of assessing the IAQ in their dwellings and act upon it by correctly operating the ventilation provisions. This assumption is questionable to say the least. Various studies have been performed to investigate how people judge their IAQ and their ventilation system. Apart from monitoring the actual IAQ in 62 dwellings, the MONICAIR project also did a survey amongst its inhabitants. And although CO_2-levels in bedrooms and living rooms frequently exceeded the 1200 ppm limit values (sometimes even exceeding the 3500 ppm CO_2), 95% of the interviewees are content or fairly content with their ventilation systems and rate the freshness of their indoor air with a 7 or an 8 out of 10. Combined with the facts that various indoor pollutants are odourless (radon, CO, various VOC's), that people adapt to their IAQ, and that people do not possess the required sensory capacities to perceive the IAQ, this leads to the conclusion that people cannot be held responsible for the correct operation of their ventilation system and the resulting IAQ. Consequently the design paradigm that performance of ventilation systems can be based on human intervention should be abandoned.

3.5 No test procedure for determining the performance of ventilation systems

Apart from the fact that there is no clear legal performance requirement for ventilation systems (see 3.1), a proper test procedure for determining the performance of a ventilation system on its primary function, is also missing. There are several national and European test standards on the individual components of a ventilation system, but so far a validated test procedure for the overall ventilation system is lacking. Simulation programs are now filling this gap, but without ample validation from real life field research, their link with reality is questionable. As illustration: for all ventilation systems that were investigated in the MONICIAR field research project, simulation were done using the software of VLA-method [lit.10] . The result was that all systems complied with the Air Quality Index limit value of 30 kppmh per person per heating season. The MONICAIR field research project supplied data showing that real life values for this Air Quality Index vary from 0 tot 853 kppmh per person per heating season, with system averages way above 30 kppmh/pp per heating season.

As a consequence of the situation described above, the ventilation market and the IAQ-topic is facing difficulties, because:

- Fact and fiction on the performance of ventilation systems cannot be distinguished (claims cannot be verified)
- Product development and design of ventilation systems is focussed on compliance with building codes and not on performance requirements
- Procurement procedures cannot dispose of adequate IAQ and energy performance data
- IAQ is not an issue when selecting ventilation systems

4. CONLUSIONS : PROPOSALS FOR IMPROVEMENT

Over the years the performance of ventilation systems has certainly improved, as well as the legal framework and performance assessment tools for these systems. Recent insights and knowledge however give cause for a next step in refining this framework and related assessment methods, especially with a building stock that is gradually improving in terms of thermal insulation and air tightness.

With the right focus both MEV and MVHR systems can be further improved to ensure that the required air exchange rates can also be achieved in the distinctive habitable rooms, particularly during presence.

Key requisites for achieving these improvements are:

4.1 Drawing up of minimal requirements

Give a stricter and verifiable definition of the primary function of residential ventilation systems and add minimal performance requirements as to when and to what extent air exchange rates must be achieved in both wet rooms and habitable rooms.

4.2 A validated and (internationally) accepted test protocol for complete systems

Develop a test protocol that facilitates the measurement of the real life performance of ventilation systems. This protocol can for instance be based on the methods and indicators used in the MONICAIR field-research project, for which the Air Quality Index of the VLA-method [lit.11] was the underlying principle. Thus acquired results can be used for fine tuning of existing simulation software. The acquired knowledge can also be used for the further characterization of ventilation systems and their relation with its IAQ-performance and the energy performance.

4.3 Design a system for indicating the IAQ-performance of ventilation systems

Obviously there will be ventilation systems that can do more and perform better than the minimal requirements. Examples are ventilation systems that:

- are able to (silently) increase the air exchange rates per person and reduce CO_2-concentrations to long-lasting levels below 1000 or 800 ppm,
- automatically adjust air exchange rates per room to the number of people present or to their (polluting) activities
- have (standards or improved) filter capabilities,

Such differences must become visible to the market. Large quantities of real life performance data (see 4.2) will also facilitate the categorization or labelling of ventilation systems on their IAQ-performance.

4.4 Link the IAQ-performance with the energy performance

Finally a correct link between the IAQ- and the energy performance must be made and communicated. Without such a link, an assessment of the energy performance is without meaning.

The MONICAIR consortium is discussing with the Dutch standardization body NEN, the Ministry of Interior Affaires and research institutes how and to what extend these modification can be implemented. The Consortium is also initiating the process of developing a test protocol with a corresponding ventilation test kit and is looking for international backing and support.

5. ACKNOWLEDGEMENTS

This paper was prepared within the MONICAIR project, part WP3a. The project is financed by the consortium members and by the Dutch Ministry of Economic Affairs within the framework of TKI (Top consortia for Knowledge & Innovation). Consortium members and their particular representatives involved in this part of the MONICAIR study are: Jelmer de Jong, Peter Schabos, Rutger Vasters, Rolf-Jan Hans, Hans van Klooster, Roelof Ziengs, Leon van Bohemen, Jan Rijnbeek, Harm Valk, Wim Kornaat, Bart Cremers, Rick Bruins, William Li, Bas Knoll and Rob van Holsteijn. The MONICAIR consortium owes thanks to Peter Cool for his support on the data monitoring of the boilers, to Robèrt de Haas for his patience and support in developing the MONICAIR database and queries, to Wilbert van den Eijnde for the face-to-face interviews with all inhabitants and finally to the housing corporations for their support in the selection of dwelling for this part WP1a of the MONICAIR project: Barry Top, Gerben Heering and Winfried van Emmerik.

6. REFERENCES

[1] Van Holsteijn, R.C.A., Li, W.L., Valk, H.J.J., Kornaat, W. (2015), Improving the Energy- & IAQ Performance of Ventilation Systems in Dutch Residential Dwellings, Conference Proceedings Healthy Buildings Europe May 2015, The Netherlands, Paper ID504.

[2] McGill, G.M., Oyedele, L.O., Keeffe, G.K., McAllister, K.M., Sharpe, T. (2015), Bedroom Environmental Conditions in Airtight Mechanically Ventilated Dwellings, Conference Proceedings Healthy Buildings Europe May 2015, The Netherlands, Paper ID548.

[3] Tappler, P., Hutter, H.P., Hengsberger, H., Ringer, W. (2014), Lüftung 3.0 – Bewohnergesundheit und Raumluftqualtität in neu errichteten energieefficienten Wohnhäusern, Österreichisches Institut für Baubiologie und Bauökologie (IBO), Wien, Austria

[4] Klepeis, N.E., Nelson, W.C., Ott, W.R., Robinson, J.P., Tsang, A.M., Switzer, P. (2001), The National Human Acitivity Pattern Survey (NHAPS): A Resource for Assessing Exposure to Environmental Pollutants, Lawrence Berkeley National Laboratory, Berkeley, USA.

[5] Logue, J.M., Sherman, M.H., Price, P.N., Singer, B.C. (2011), Why We Ventilate, Environmental Energy Technologies Division, Lawrence Berkeley National Laboratory, Report Number 5093-E, Berkeley, USA.

[6] Directive 2008/50/EC of the European Parliament and of the Council of 21 May 2008 on ambient air quality and cleaner air for Europe, (2008), Official Journal of the European Union.

[7] Energy performance of buildings – Part 1: Indoor environmental input parameters for design and assessment of energy performance of buildings ddressing indoor air quality, thermal environment, lighting and acoustics – Module M1-6, prEN 16798-1:2015

[8] De Gids, W.F., Wouters, P., (2010). CO_2 as indicator for the indoor air quality, General principles, AIVC Paper N° 33.

[9] Schell, M., Inthout, D. (2001), Demand Control Ventilation Using CO_2, Ashrae Journal, February 2001.

[10] Van Holsteijn, R.C.A., Li, W.L. (2014), Monitoring the energy and IAQ performance of ventilation systems in Dutch residential dwellings, Proceedings of the 35th AIVC conference, Poznan, Poland.

[11] Vereniging Leveranciers Luchttechnische Apparaten, VLA, (2012). VLA methodiek gelijkwaardigheid voor energiebesparende ventilatieoplossingen in woningen.

EFFECTS OF CARBON DIOXIDE WITH AND WITHOUT BIOEFFLUENTS ON HUMANS

Xiaojing Zhang[1], Pawel Wargocki[2,*], and Zhiwei Lian[1]

[1] *School of Naval Architecture, Ocean & Civil Engineering, Shanghai Jiao Tong University, Shanghai, China*

[2] *International Centre for Indoor Environment and Energy, Technical University of Denmark, Copenhagen, Denmark*
**Corresponding author: paw@byg.dtu.dk*

ABSTRACT

Carbon dioxide (CO_2) has traditionally been assumed innocuous at the typical levels indoors, and merely an indicator of metabolic emissions from humans (bioeffluents). Recent studies suggest that exposure to pure CO_2 at concentrations of 2,500 to 4,000 ppm, the levels that occur periodically indoors, can have negative effects on mental performance in form of reduced ability for making decisions, typing and proofreading. Present study aimed to examine further these effects. Twenty-five human subjects were exposed to elevated CO_2 with and without bioeffluents in a chamber. The exposure levels were as follows: background exposure with CO_2 at 500 ppm, exposure to pure CO_2 (without bioeffluents) at 1,000 and 3,000 ppm, exposure to metabolically generated CO_2 (with bioeffluents) at 1,000 and 3,000 ppm. Each exposure lasted 4.25 hours and their order was balanced. Subjects rated perceived air quality and acute health symptoms. They performed several cognitive tasks. Their physiological responses were monitored. No effects were seen during exposures to elevated levels of pure CO_2 (without bioeffluents). Exposures to elevated levels of bioeffluents (with CO_2) reduced significantly perceived air quality assessed upon entering the chamber, increased significantly intensity of neurobehavioral symptoms, and reduced mental performance of addition in terms of lowering the speed and increasing error rates the latter two effects particularly so when metabolic CO_2 was at 3,000 ppm. Physiological measurements and the performance of Tsai-Partington test suggested that exposures to bioeffluents increased arousal/stress and this result together with the stronger intensity of neurologic symptoms and complaints could be the underlying mechanisms of reduced performance. Present results confirm the expected negative effects of exposure to elevated levels of bioeffluents thereby providing basis for ventilation requirements. The effects of exposures to pure CO_2 up to 3,000 ppm seem to be imperceptible and undetectable by the used cognitive tests, but the role of CO_2 in the mixture with bioeffluents shall not be underestimated and further examined.

KEYWORDS

Carbon dioxide; Human bioeffluents; Perceived air quality; Acute health symptoms; Cognitive performance; Physiological responses

1 INTRODUCTION

Since the 19th Century, the carbon dioxide (CO_2) concentrations indoors have been used as an indicator of air quality in buildings (Pettenkofer, 1858). CO_2 is a product of metabolic processes in human body and exhaled in abundance allowing simple measurements and tracking, but only in the presence of humans. In numerous studies, CO_2 has therefore been measured and associated significantly with subjectively assessed acute health symptoms such as headaches, fatigue, eye, nose and respiratory tract symptoms. In these studies, CO_2 is merely considered as an indicator of exposures to other pollutants and insufficient ventilation with outdoor air and not as a causative agent causing the observed effects (Seppänen et al., 1999). The concentration of CO_2 at which these effects were observed in the buildings were <5,000 ppm and usually <3,000 ppm.

Recent studies suggest that exposure to pure CO_2 (without bioeffluents) at the levels relevant for indoor non-industrial environments can cause significant negative effects on mental performance. One study by Kajtár and Herczeg (2012) showed that performance of proofreading test was negatively affected when taken by ten subjects during exposures to CO_2 without bioeffluents at 3,000 ppm for 2 to 3 hours. In parallel to the effects on performance, Kajtár and Herzceg (2012) observed that diastolic blood pressure increased and mid-frequency components of heart rate variability decreased. The effects were attributed to increased mental effort during these exposures. Another study by Satish et al. (2012) showed systematic reduction in the decision-making performance with increasing pure CO_2 (without bioeffluents) from 600 to 1,000 and 2,500 ppm during a 2,5-hour exposure; this effect reached statistical significance at 2,500 ppm. No physiological measurements were however made in this study to examine the potential underlying mechanism of the observed effects. That these low CO_2 levels could be of importance have been additionally showed by Maddalena et al. (2014). They exposed subjects for 4 hours this time to 1,800 ppm metabolically generated CO_2 (with bioeffluents) and found that the decision-making performance was also significantly reduced compared with 900 ppm. The magnitude of effects was however much lower than in the study of Satish et al. (2012). Maddalena et al. (2014) did not measure physiological responses either. They asked subjects to rate their acute health symptoms and perceptions of environment; no differences were however observed in these ratings between different exposures examined.

The results of Kajtár and Herzeg (2012) and Satish et al. (2012) suggest that CO_2 at the levels measured normally indoors may have negative effects on human responses. They suggest thus that CO_2 should no longer be considered as only a surrogate for indoor air quality but also as a pollutant. Consequently, present study was performed to examine further the validity of this postulate.

2 METHODS

The experiment was carried out in a 3.6 × 2.5× 2.5 m stainless steel chamber (30 m^3 volume with recirculation ducts) (Figure 1), which was described in detail by Albrechtsen (1988). The construction minimizes the emissions and sorption of pollutants and ensures that the chamber volume is tightly sealed. The chamber has its own system for supplying and conditioning outdoor air. There were six workstations in the chamber for the subjects and the experimenter, each consisting of a table, a chair, a laptop PC and a desk lamp.

Twenty-five subjects were recruited through advertisements placed on the university campus. They were all students. Ten males and fifteen females were included with a mean age of 23±2 (mean ± SD) years, mean height of 173±11 cm and mean weight of 74.9±21.8 kg. During experiments, the subjects wore the same type of self-selected garment (with mean thermal insulation estimated to 0.75 clo) in order to remain thermally neutral during each exposure.

In three of the five exposures examined in the present experiments, the outdoor air supply rate was high enough to remove bioeffluents, creating a reference condition with CO_2 at 500 ppm (referred to as B500), while chemically pure CO_2 was added to the supply air to create exposure conditions of 1,000 ppm or 3,000 ppm (referred to as P1000 and P3000). In two other conditions, the outdoor air supply rate was restricted (reduced) to allow the metabolically produced CO_2 to reach 1,000 ppm or 3,000 ppm (referred to as M1000 and M3000), thereby ensuring that other bioeffluents reached concentrations corresponding to those in the occupied rooms with CO_2 at these levels. Temperature and noise level were kept constant during the exposures, however, due to the lack of a dehumidifier, the relative humidity (RH) increased by a few percentage at M3000.

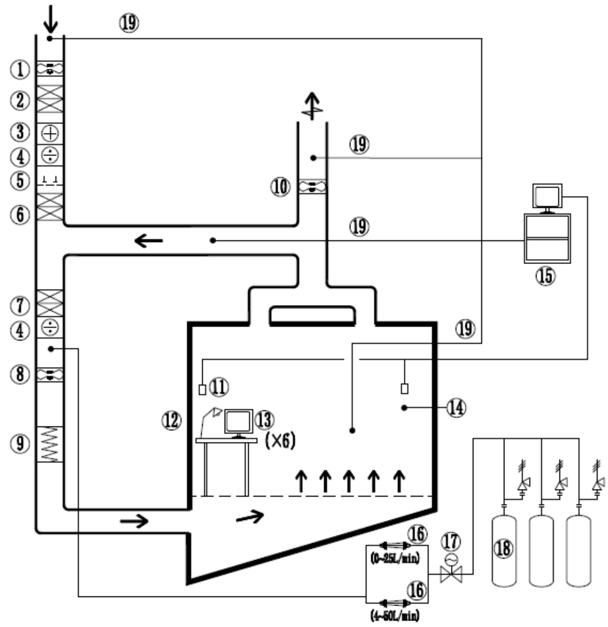


Figure 1 Schematic figure of the chamber, where the experiments were carried out (left) and a view of the inside of the chamber (right):① supply fan, ② two stage filter G3/F7, ③ heating coil, ④ cooling coil, ⑤ dampers, ⑥ filter box for charcoal filters (empty), ⑦ filter box (empty), ⑧ recirculating fan, ⑨ electric heating coil, ⑩ exhaust fan, ⑪ HOBO logger (temperature & relative humidity sensor) with CO_2 sensor, ⑫ desk lamp, ⑬ laptop, ⑭ temperature and humidity sensor of the chamber control system, ⑮ multi-gas analyser, ⑯ flowmeters, ⑰ pressure regulator, ⑱ CO_2 gas cylinders (30L), ⑲ sampling point

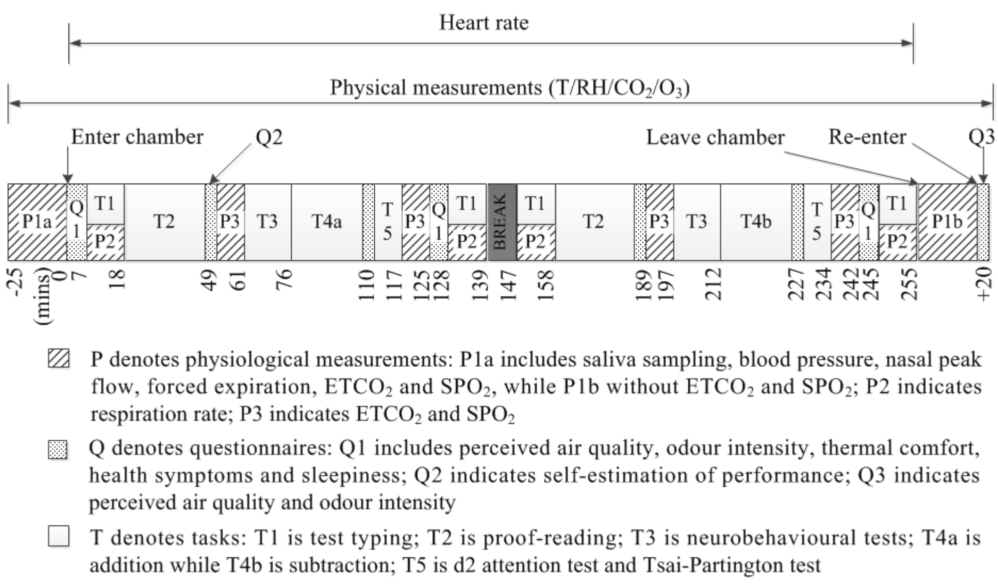


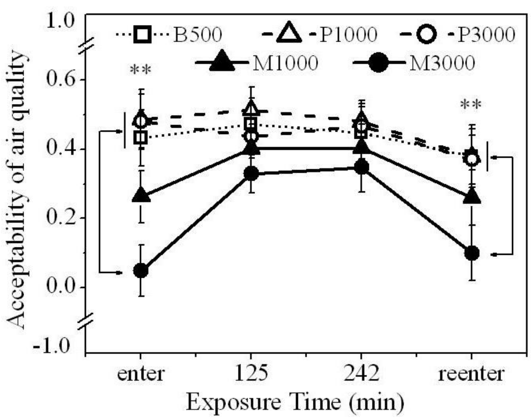
Figure 2 Experimental procedure

Twenty-five subjects were exposed in the climate chamber for 255 minutes in groups of five persons at a time in a Latin-square design; each group participated in the experiment in one week from Monday to Friday. Mental performance was examined by the test battery (TB) including multiple tasks somewhat resembling office work (proofreading test, addition, subtraction and text typing), as well as by neurobehavioral tests (redirection, digit span memory, grammatical reasoning, Stroop and Stroop with feedback), Tsai-Partington test and d2 attention test. Subjective ratings of comfort and acute health symptoms were rated by the subjects during exposures, as well as physiological responses (PM) were monitored and saliva samples were collected to analyse levels of stress biomarkers: α-amylase and cortisol. Figure 2 shows in details the experimental procedure during each exposure.

The effects of exposures on different outcomes were analysed using a mixed ANOVA model; the significance level was set to 0.1 for random effects and to 0.05 for fixed effects. Experimental conditions (C), time at which different assessments were made during the day (T), condition×time interaction (C×T), order of exposure of conditions (O) and gender (G) were included as fixed factors. Subjects (S), groups (Gr), subject×condition interaction (S×C) and subject×time interaction (S×T) were included as random factors in the model. In addition to mixed ANOVA model, Page test for trend was used for these outcomes that changed monotonically with CO_2 levels with significance level set to 0.05 (1-tail).

3 RESULTS AND DISCUSSION

Figure 3 (left) shows that the acceptability of the air quality upon entering the chamber polluted by human bioeffluents prior to and after exposure was assessed to be lower than in the other three exposure conditions; ratings at M3000 were statistically significantly different from the other assessments except these at M1000. There were no significant differences in the acceptability of the air quality between background exposure (B500) and exposures to artificially raised CO_2 concentrations (P1000 and P3000). Assessments of odour intensity confirmed that air quality was worse only during exposures with elevated levels of bioeffluents (M1000 and M3000). Using assessments of acceptability of air quality the % dissatisfied with air quality was estimated and compared with the relationship developed by Fanger (1988); the results from the present study matched well the previously established relationship except for the low levels of CO_2 (Figure 3, right).



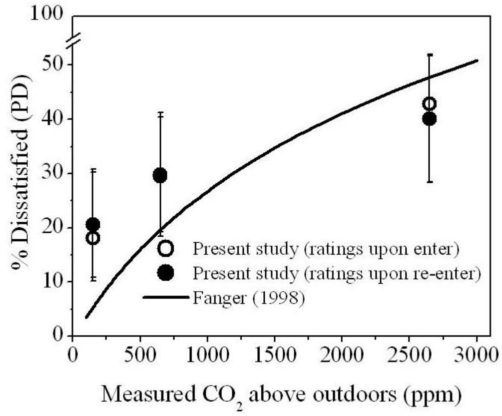


Figure 3 Acceptability of air quality as a function of exposure time in the present experiments (left) and the % dissatisfied as a function of CO_2 concentration based on the present data and as reported by Fanger (1988) (right); ** indicates the specific periods during exposure, at which the differences between some conditions reached statistical significance; for the assessment of acceptability, -1=clearly unacceptable, +1=clearly acceptable and 0= just not acceptable/just acceptable

Figure 4 shows that difficulty in thinking clearly, headache, fatigue and sleepiness were significantly higher at M3000, while there was a very small difference in the intensity of these symptoms between other exposure conditions. Only difficulty in thinking clearly was marginally worse in M1000 than in the other exposure conditions but the difference did not reach formal statistical significance.

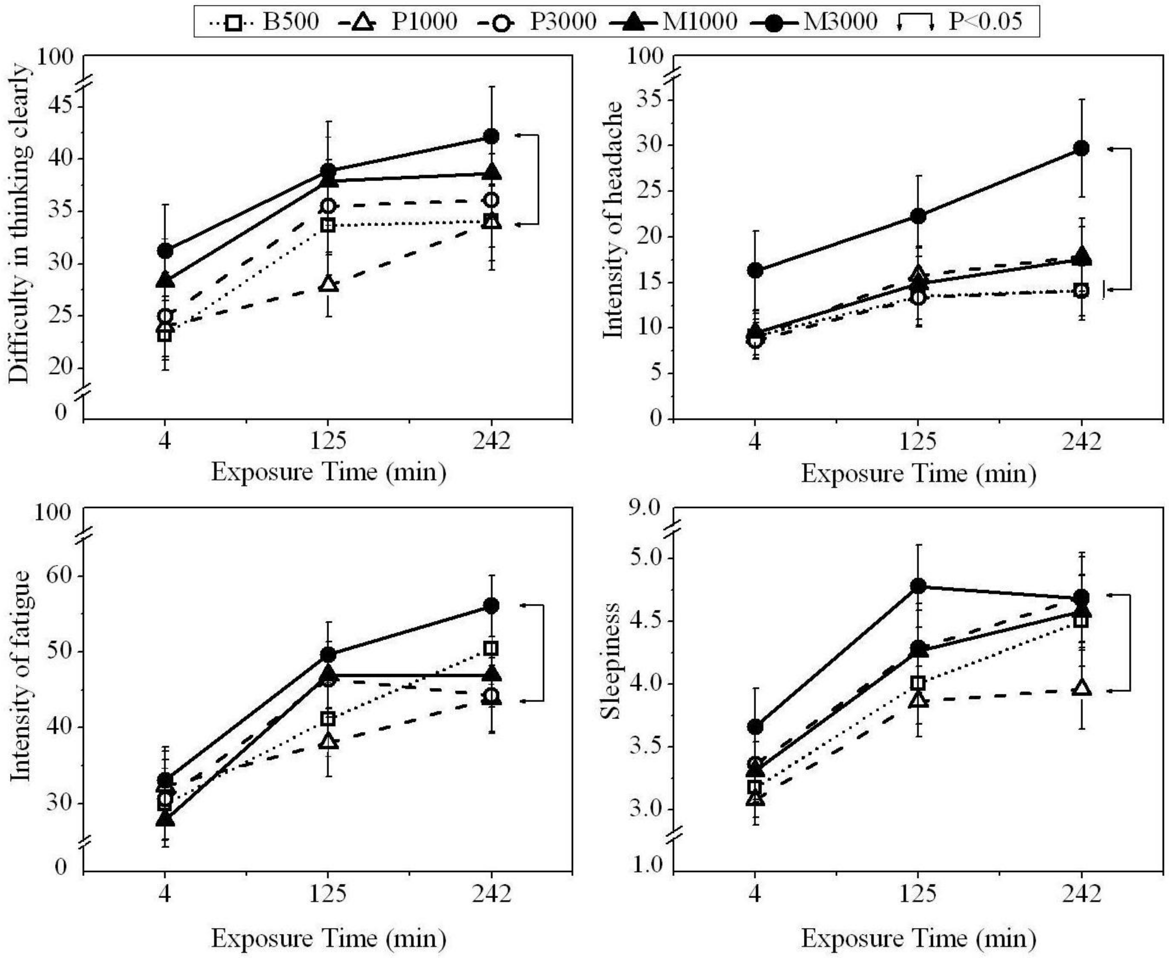


Figure 4 Intensity of acute health symptoms, which in the statistical models were seen to be statistically significantly different between exposures; the arrows indicate significant difference between exposure conditions; for the assessment of difficulty in thinking clearly, 0=easy, 100=hard; for the assessment of headache, 0=no headache, 5=severe headache; for the assessment of fatigue, 0=rested, 100=tired; for the assessment of sleepiness, 1=very alert, 9=very sleepy

Among the many tests examining mental performance, there were only few, for which the performance differed significantly between conditions (Table 1). Speed at which subjects added units was lower (P=0.023) for different conditions compared with B500, the difference reaching statistical significance at M3000. Analysis of variance showed that % of errors made by the subjects varied also between different conditions (P=0.049). It was however not possible to determine, at which conditions the difference was statistically significant through the post-hoc tests. Raw data showed the highest % of errors at B500 and M3000. Analysis of variance showed also that the difference in performance of proof reading performed in various exposure conditions approached statistical significance (P=0.062): Speed, at which subjects proof-read the text was varying across different conditions but was the lowest at M3000 while highest at P3000; there were no effects on errors or false positives. Page test showed that speed increased systematically between B500, P1000 and P3000 (P<0.05); systematic effect was also seen in reduction of speed between B500, M1000 and M3000 but this trend did not reach formal statistical significance.

The results of Tsai-Partington test showed that the number of correct links made by the subjects was lower for different conditions compared with B500, the difference reaching statistical significance at M1000 (P<0.001) (Figure 5). The performance of Tsai-Partington test depends on the level of stress and is improved (more links are made) at lower stress and large attention field (Eysenck and Willett, 1962). Thus this result suggests that the arousal of subjects was higher at elevated CO_2 levels, particularly so when other bioeffluents were present. This is to some extent confirmed by the results of redirection test, subjects responding significantly quicker at M3000 compared with P1000 (P=0.015) as would be expected at higher arousal (Duffy, 1957).

Table 1 Performance of mental tasks that differed between conditions at P<0.1 (LS Mean± SE)

Outcomes	Condition					P
	B500	P1000	P3000	M1000	M3000	
Units completed per minute in addition	3.6±0.2	3.3±0.2	3.2±0.2	3.2±0.2	3.2±0.2	0.023
% Errors made in addition	9.4±1.3	7.1±1.3	8.3±1.3	7.0±1.3	9.1±1.3	0.049
Number of correct links in Tsai Partington test	14.6±0.6	14.2±0.6	13.5±0.6	12.6±0.6	13.2±0.6	< 0.001
Lines proof-read per minute in proof-reading test	10.5±0.5	11.0±0.5	11.0±0.5	10.5±0.5	10.3±0.5	0.063
Response time in redirection test (s)	165.4±6.3	177.0±6.3	164.8±6.3	165.7±6.3	158.7±6.3	0.015

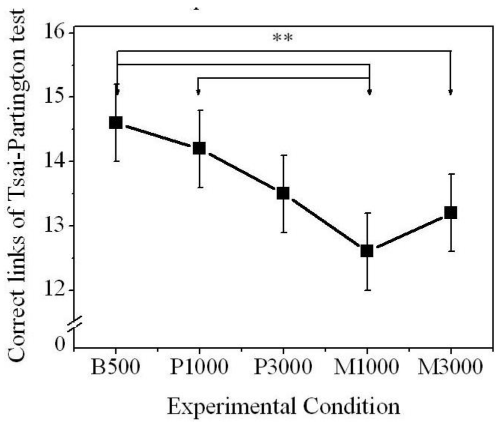


Figure 5 Average performance of Tsai-Partington at different exposure conditions; ** shows the differences that reached statistical significance; bars show standard error

Heart rate was higher at the beginning of each exposure when compared with the subsequent sessions. In particular, heart rate was significantly higher at M3000 and P3000 compared with B500 at 128 min and at M3000 vs. B500 at 147 min (Figure 6).

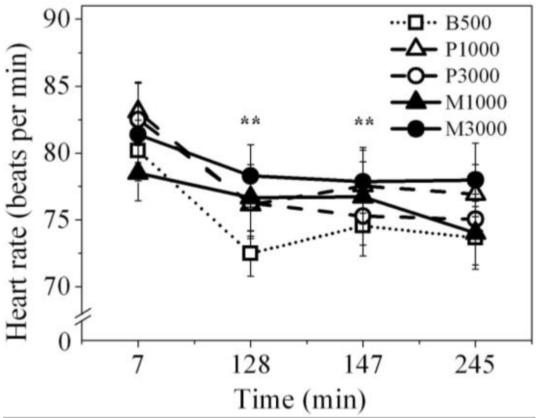


Figure 6 Change of heart rate along the course of exposure; the figure shows average heart rate for four typing sessions; ** indicates where the differences between some conditions reached statistical significance (see text for details); bars show standard error

Diastolic blood pressure increased after the exposure compared with the pre-exposure level in all conditions; the increase after exposure to CO_2 with bioeffluents at 3,000 ppm (M3000) was statistically significant. Moreover, the magnitude of increase in diastolic blood pressure at M3000 was significantly higher than that in the other four exposures (Figure 7). Higher diastolic blood pressure can be caused by vasoconstriction as a reaction of the sympathetic nervous system to higher stress/arousal. This result is consistent with the findings of Kajtár and Herczeg (2012), who observed that diastolic blood pressure increased after exposure to CO_2 without bioeffluents at 5,000 ppm compared with 600 ppm. In addition, increased heart rate could also be a result of activation of sympathetic nerve and manifestation of higher physiological stress.

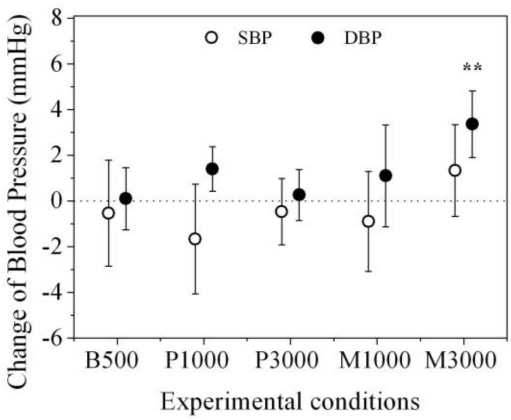


Figure 7 Difference in systolic and diastolic blood pressure between the levels before exposure and after exposure; ** shows the differences that reached statistical significance; bars show standard error

The results of analysis of biomarkers in saliva seem to confirm higher arousal/stress. Although salivary α-amylase increased significantly, while cortisol levels decreased significantly after 4.25 h exposure, independently of conditions, which is likely due to diurnal changes in these two biomarkers, the exposure to CO_2 with bioeffluents at 1,000 ppm and 3,000 ppm (M1000 and M3000) increased alpha-amylase more than would be expected due to diurnal rhythm as in other exposures (Figure 8).

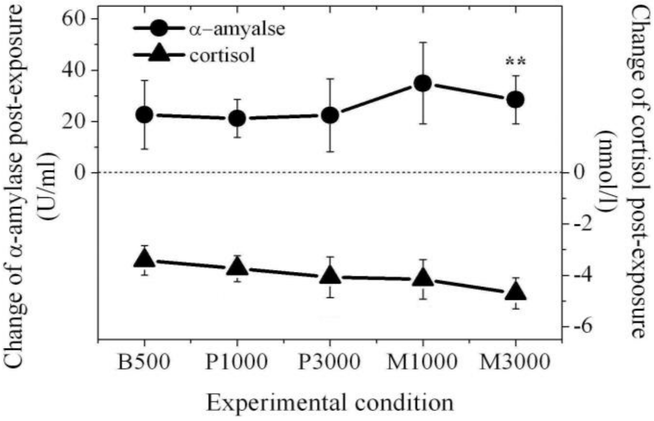


Figure 8 Difference in concentration of α-amylase and cortisol between the levels before exposure and after exposure; ** shows the differences that reached statistical significance; bars show standard error

4 CONCLUSIONS

Exposure to CO_2 without bioeffluents at or below 3,000 ppm did not produce any adverse subjective ratings concerning air quality and acute health symptoms, neither decreased the performance of mental tasks during 4.25-hour exposures.

Exposure to metabolically generated CO_2 with bioeffluents at 3,000 ppm reduced perceived air quality, increased the intensity of neurobehavioral acute health symptoms, and significantly affected performance of some mental tasks.

Increased arousal level and neurologic symptoms during exposures to bioeffluents with CO_2 at 3,000 ppm may be the reason for the observed effects on mental performance.

5 ACKNOWLEDGEMENTS

This work is supported by Bjarne Saxhof Foundation and Key Program of National Natural Science Foundation of China (51238005).

6 REFERENCES

Albrechtsen, O. (1988) Twin climatic chambers to study sick and healthy buildings. *In: Proceedings of Healthy Buildings '88*, Vol 3, pp. 25-30.

Duffy, E. (1957) The psychological significance of the concept of arousal or activation. *Psychological review*, 64(5), 265-275.

Eysenck, H. J. and Willett, R. A. (1962) Cue utilization as a function of drive: An experimental study. *Perceptual and motor skills*, 15(1), 229-230.

Fanger, P.O. (1988) Introduction of the olf and the decipol units to quantify air pollution perceived by human indoors and outdoors, *Energ. Buildings*, **12**, 1-6.

Kajtár, L. and Herczeg, L. (2012) Influence of carbon-dioxide concentration on human well-being and intensity of mental work, *Q. J. Hungari. Meteor. Serv.*, 116, 145-169.

Maddalena, R., Mendell, M.J., Eliseeva, K., et al. (2014) Effects of ventilation rate per person and per floor area on perceived air quality, sick building syndrome symptoms, and decision-making, *Indoor Air*.

Pettenkofer, M.V. (1858) *Über den Luftwechsel in Wohngebäuden*, München, Cotta'schen buchhandlung.

Satish, U., Mendell, M.J., Shekhar, K., et al. (2012) Is CO_2 an indoor pollutant? Direct effects of low-to-moderate CO_2 concentrations on human decision-making performance, *Environmental Health Perspectives*, **120**, 1671-1705.

Seppänen, O.A., Fisk, W.J. and Mendell, M.J. (1999) Association of ventilation rates and CO_2 concentrations with health and other responses in commercial and institutional buildings, *Indoor Air*, 9, 226-252.

CAPTURE EFFICIENCY OF AIR CURTAIN ASSISTED RESIDENTIAL RANGE HOODS

Bruno Claeys[1], Jelle Laverge[1], Ivan Pollet[2,3] and Giel Bruyneel[3]

1 Buidling Physics research group
Ghent University
St.-Pietersnieuwstraat 41 B 4
9000 Gent, Belgium
**Corresponding author: jelle.laverge@ugent.be*

2 Biosystems engineering
Ghent University
Coupure links 653
9000 Gent, Belgium

3 Renson Ventilation
Renson NV
Maalbeekstraat 10
8790 Waregem, Belgium

ABSTRACT

Air curtain assisted range hoods are very customary in large industrial kitchens. They allow to increase the capture efficiency of the range hood while lowering the net exhaust flow rate. For applications in residential settings, there is a lack of data on the performance of air curtain assisted range hoods, as well as a lack of information on the required settings and boundary conditions to come to the successful application of air curtain assisted range hoods. In this paper we present the results from an experimental test campaign in which we investigated the capture efficiency of a residential air curtain assisted range hood in comparison with a regular range hood, as well as the sensitivity of the capture efficiency to boundary conditions such as net exhaust flow rate, height above the range, enclosure etc. The results show that air curtain assisted range hoods are more efficient at lower flow rates, especially in non-enclosed settings, confirming the performance known from industrial kitchens, but are sensitive to higher mounting and on-going cooking activities.

KEYWORDS

Range Hood, Carbon Dioxide, Air Curtain, Efficiency

1 INTRODUCTION

The acoustic and airflow performance of range hoods has been captured by ISO standards 5167-1 (2003) and 10140-1 (2010), while the IEC 61791 standard (1997), discusses fat absorption and odor extraction performance. These aspects are also treated in the EN 13141-3 (2004) standard. There is, however, no mention of capture efficiency, the efficiency of the range hood to capture and exhaust the pollutants emitted by the cooking activities as is used for the testing of range hoods in commercial kitchens (CEN 2014). Test methods for this measure have been proposed and tested on residential range hoods available on the US market by Delp (2012) and Lunden (2014). On the other hand, air curtain assisted range hoods have been introduced and tested successfully in commercial kitchens where they reduce the exposure of the staff not directly working at the ranges. In this paper, we build on the methodological work done to define capture efficiency and compare the performance of air curtain assisted range hoods with that of normal direct extraction range hoods in residential kitchens in climate chamber experiments. In both cases, the range hood doubles as the kitchen ventilation vent hole.

2 METHODS

In this section, we will first discuss the experimental setup that was used for the experiments and then briefly introduce capture efficiency as the used metric to process the results.

2.1 Experimental setup

A test range was constructed within the hotbox of a hot/cold/hot box suite. The space measures 5 by 5 by 2,7 meters, dimensions that are fairly representative for a kitchen in Belgium. The height of the test range is adjustable, the height of the range hood is fixed. Two range hoods are mounted on either side of the setup, one standard residential range hood and one air curtain assisted range hood, so that they can be tested side by side. The enclosure of the range hood can be adapted from free hanging to enclosed in kitchen cupboards by adding or removing wooden paneling. The test setup is shown in figure 1 below.

Figure 1: The experimental setup with adjustable range height, electric range and air curtain assisted range hood in wall mounted non-enclosed mode.

2.2 Capture efficiency calculation

To process the measurements and calculate the capture efficiency, we use the 'indirect approach' put forward by Delp (2012) and Lunden (2014). Lab grade carbon dioxide is released at a constant rate above a boiling pot of water on the range. The carbon dioxide concentration is measured at 2 locations in the exhaust, at 2 locations in the room and in the fresh air intake as is shown in figure 2.

Capture efficiency is then defined as:

$$CE_{FP} = \frac{C_{hood} - C_{room}}{C_{hood} - C_{inlet}} \quad (1)$$

where CE_{FP} is the first pass capture efficiency, C_{hood} is the carbon dioxide concentration measured in the exhaust of the range hood, C_{room} is the carbon dioxide concentration measured in the test room and C_{inlet} is the carbon dioxide concentration measured in the inlet of the room. All measured concentrations are reported in parts per million (ppm).

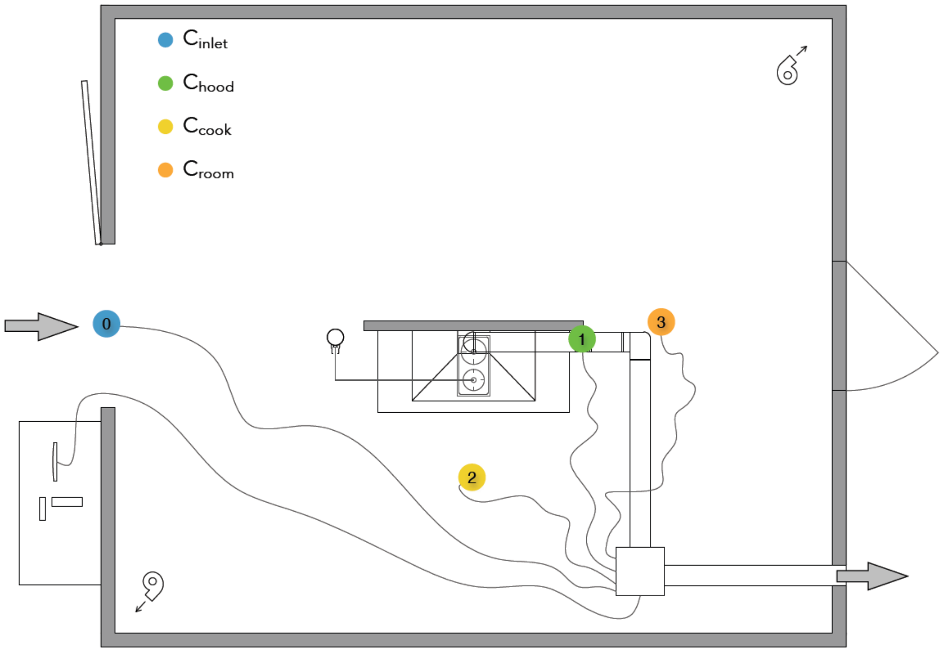


Figure 2: Scheme of the climate chamber with the location of the setup and the different sensors.

The concentrations measured during a 3 minute cooking event with the air curtain assisted range hood mounted at 0,75 meter above the range are shown in figure 3. The exhaust flow rate of the range hood during the test is 170 m3/h.

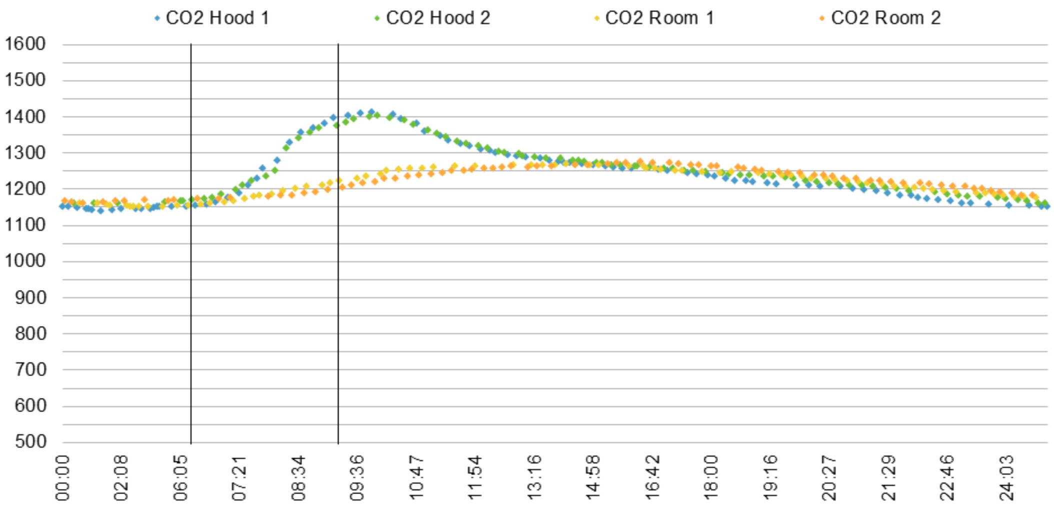


Figure 3: Measured carbon dioxide concentrations during a 3 minute cooking event with the air curtain assisted range hood and the cooking pot on the front burner.

Room and exhaust concentrations coincide in the pre-cooking phase of the test (minute 1-6), when no carbon dioxide is released. The concentration in the exhaust of the range hood rapidly climbs during the cooking event (minute 6-9, marked with the 2 vertical black lines in figure 3) while that in the rooms remains relatively low. This is, of course, the desired effect of the range hood and constitutes the effectiveness of the range hood. At the end of the cooking event, the momentary first pass capture efficiency is 77%.
23% of the emitted carbon dioxide, however, still escapes to the room. After the cooking event is stopped, the test is continued until all emitted carbon dioxide is exhausted and room as well as hood concentrations are back at inlet level (minute 9-25). The carbon dioxide concentration within the cooking zone decreases rapidly and becomes equal to the room concentration. The overall capture efficiency integrated over the total length of the test is only 30%. This metric, compared to the first pass efficiency, is not commonly used, but is more representative for the effectiveness of the range hood in reducing the total exposure of the occupants, especially since in Belgian kitchens, the kitchen tends to also be the dining area for family dinners, the dining room is usually only used for formal dinners.

3 RESULTS

The difference between the standard and air curtain assisted range hood become apparent when we compare the results of the test on both hoods under the same circumstances. For this test, we selected the same height between the range hood and the range as in the test shown in figure 3, 0.75 m, but instead of just having a pot on the front burner, 2 pots, one on the back and one on the front burner, were used. An summary of the test conditions and the measured capture efficiencies is given in table 1.

Table 1: standard vs. Air curtain assisted range hood performance
(experiment with pots on the front and back burner)

Range hood	Flow rate	Height	CE_{FP}	CE_{total}
Standard	170 m^3/h	0.75 m	0.85	0.52
Air curtain assisted	175 m^3/h	0.75 m	0.49	0.16
Air curtain assisted	250 m^3/h	0.75 m	0.50	0.16

At similar flow rates, the capture efficiency of the air curtain assisted range hood is better both during the cooking event (first pass) and over the complete period of exposure of the occupants.
The concentrations measured at each of the measuring points are shown in figure 4, for the test with standard and air curtain assisted range hoods both operating at approximately 175 m^3/h. Although they do both seem to function rather well during first few instances of the cooking event, the capture efficiency of the standard range hood plummets after about 2 minutes due to the turbulence induced by the pots of boiling water. While the capture efficiency of the air curtain assisted range hood also deteriorates at that point, it remains relatively high due to the containment of the pollutant above the range due to the air curtain.
When the cooking event is stopped, the thermal plume created by the pots of boiling water disappears and the efficiency of both range hood worsens, but again, the containment achieved by the air curtain ensures a better endurance of the performance of the range hood.

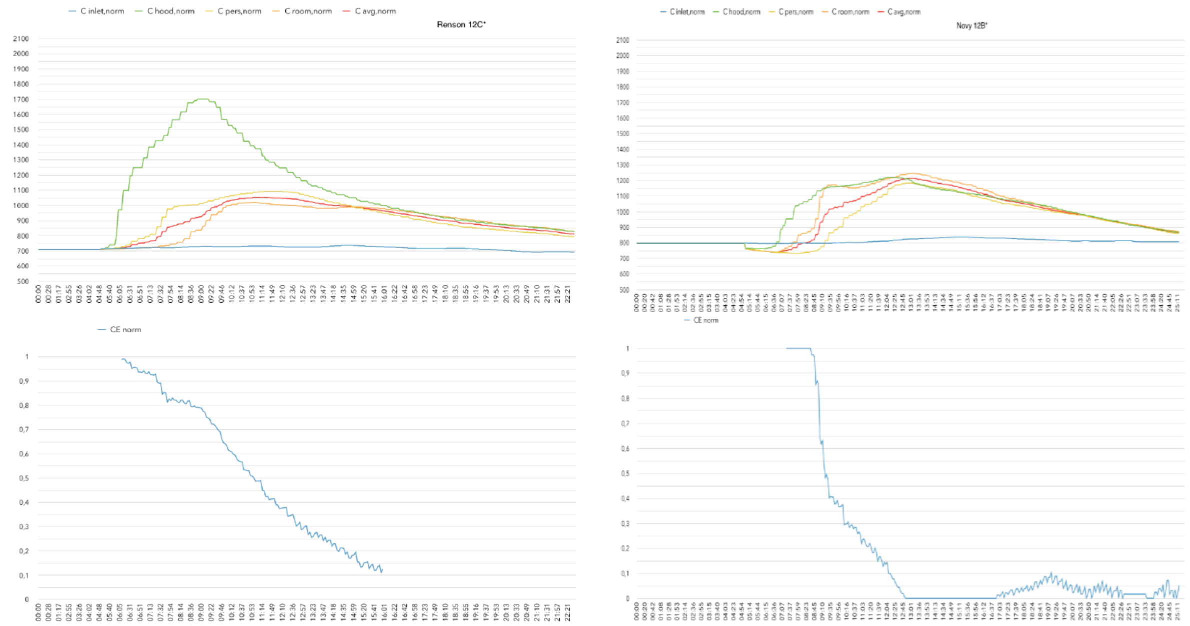


Figure 4: Measured carbon dioxide concentrations and capture efficiencies during a 3 minute cooking event with an air curtain assisted (left) and standard (right) range hood at 175 m^3/h and cooking pots on both burners

With the standard range hood, the concentration in the area near the range is almost immediately as high as above the range, effectively becoming a more or less well mixed environment.

With the air curtain assisted range hood, the excess concentration measured in the exhaust is nicely contained and exhausted, while the room concentrations, both near the range and further away, remain virtually constant and slowly go down.

The capture efficiencies and measured concentrations for the standard range hood remain virtually unchanged if the flow rate is increased by almost 50 %, as is shown in table 1.

4 DISCUSSION

The results clearly demonstrate that the 'post cooking event' period is characterized by reduced capture efficiency. In short cooking events, typical for example in frying events or reheating, this phase dominates the total capture efficiency. In the presented tests, no alternative room exhaust is used. This is rapidly becoming standard practice in kitchen ventilation in western Europe. The range hood is connected to the central exhaust ventilation unit and serves as both the ventilation vent hole and, when activated, as range hood. This configuration is a worst case scenario for the capture efficiency if the post cooking event period is taken into account.

5 CONCLUSIONS

Good capture efficiencies can be obtained with relatively small exhaust flow rates during cooking using air curtain assisted residential range hoods, where standard range hoods do much worse at the same flow rates and are virtually no better at 50% higher flow rates. Nevertheless, the post cooking event period has a large impact on overall capture efficiency in short cooking events regardless of the type of range hood that is used.

6 ACKNOWLEDGEMENTS

The authors would like to acknowledge the technical staff at Renson for their assistance in setting up the climate chamber.

7 REFERENCES

CEN/TC 156, 2014, Equipment for commercial kitchens – components for ventilation in commercial kitchens - Part 9: Capture and containment performance of extraction systems - Test methods, CEN-CENELEC pr EN 16282-9

EN 13141-3:2004 Ventilation for buildings - Performance testing of components/products for residential ventilation - Part 3: Range hoods for residential use

IEC 61591: Household range hoods – Methods for measuring performance

ISO 5167-1:2003 Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full -- Part 1: General principles and requirements

ISO 10140-1:2010 Acoustics -- Laboratory measurement of sound insulation of building elements -- Part 1: Application rules for specific products

Delp, William W. and Brett C. Singer, 2012, "*Performance Assessment of U.S. Residential Cooking Exhaust Hoods*", Environmental Science and Technology, vol 46, no11, pp. 6167-6173

Lunden, Melissa M., William W. Delp, and Brett C. Singer, 2014, "*Capture Efficiency of Cooking- Related Fine and Ultrafine Particles by Residential Exhaust Hoods*", Indoor Air

IMPACT OF STAFF POSTURE ON AIRBORNE PARTICLE DISTRIBUTION IN AN OPERATING THEATRE EQUIPPED WITH ULTRACLEAN-ZONED VENTILATION

Sasan Sadrizadeh* and Sture Holmberg

Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Stockholm, Sweden
**Corresponding email: ssad@kth.se*

ABSTRACT

Airborne particles released from surgical team members are major sources of surgical site infections (SSIs). To reduce SSI risk, ultraclean-zoned ventilation (UZV) systems have been widely applied, supplementary to the main operating theatre (OT) ventilation. Usually, OT ventilation performance is determined without considering the influence of staff-member posture and movements. Whether the surgeon's posture during surgery influences particle distribution within the surgical area is not well analysed and documented.

In this paper, two positions (bending and straighten up) representing the most common surgeon and staff postures in an OT, were analysed. The investigation used computational fluid dynamics to solve the governing equations for airflow and airborne particle dispersion.

Results indicate that bending posture increases the overall number of suspended particles in the surgical area by disrupting the particle-free airflows introduced by the UZV screen. More attention should be paid to staff work practices, since UZV efficiency is highly sensitive to the improper work experience.

KEYWORDS:

posture, ultraclean-zoned ventilation, operating theatre, contaminant distribution, computational fluid dynamics

1. INTRODUCTION

Airborne particles released from surgical team members are the major source of surgical site infections (SSIs). SSIs are the main medical and economic causes of surgical quality problems, with 98 percent of contaminants found in surgical wounds coming from ambient air (Whyte, Hodgson, & Tinkler, 1982). The bacteria suspended in the operating theatre (OT) air may contaminate a wound directly by sedimentation or indirectly by contaminating sterile instruments. Reducing airborne particle concentration is typically managed by increasing staff clothing system performance (Sadrizadeh & Holmberg, 2014c), improving OT ventilation efficiency (Sadrizadeh, Holmberg, & Tammelin, 2014), and reducing the number and activity of people in the OT (Sadrizadeh, Tammelin, Ekolind, & Holmberg, 2014). Currently, the most efficient OT ventilation system is ultraclean laminar airflow (LAF) (Sadrizadeh & Holmberg, 2014a); however, installation is relatively cost-inefficient and the resulting clean zone provided primarily addresses the surgical table and OT personnel, leaving some sterile instruments outside the protection area.

Though an ultraclean-zoned ventilation (UZV) unit was recently examined as an addition to conventional turbulent-mixing OT ventilation both experimentally (S. Friberg, Ardnor, Lundholm, & Friberg, 2003; Pasquarella et al., 2007) and numerically (Sadrizadeh & Holmberg, 2014b), the impact of staff posture on particle distribution within the surgical zone has been rarely investigated. Some previous works did indicate that staff movement might affect pathogen distribution (Chow & Wang, 2012). The on-site measurements and numerical simulation of deposited airborne particles in two OTs by Zhang et al. (Rui, Guangbei, & Jihong, 2008) also showed that team member movement could change surgical area particle distribution. The authors (Sadrizadeh & Holmberg, 2014b) recently examined the UZV as an extension to a turbulent-mixing operating suite. However, the effect of human posture on the examined UZV performance was disregarded.

Therefore, the present study numerically investigated surgical staff posture and UZV unit performance and thus addressing particle distribution within the surgical critical zone.

2. METHODOLOGIES

A square-shaped OT with the physical configuration shown in Fig. 1 was chosen as the physical model. The dimensions were 7 m × 7 m × 3.2 m (H). The OT used a turbulent-mixing ventilation system with total airflow rate of 2 m^3/s, air temperature of 20 °C and turbulence intensity of 5 percent. The air is exhausted out by four floor-level outlets located at the parallel plane of the two side walls. Two medical devices and one medical lamp with heat-load of 350 W and 250 W were considered. In total, nine surgical staff were considered, with each staff member constituting a heat source of 195 W, released from exposed surfaces. The relative positions of furnishings and surgical staff are shown in Fig. 1. A constant skin surface release rate of 5 colony-forming units (CFUs)/s was adopted. Two instrument tables and one surgical table, each equipped with UZVs, were considered. Central air velocity of 0.4 m/s was assigned to the all three UZVs, as suggested by Sadrizadeh and Holmberg (2014b).

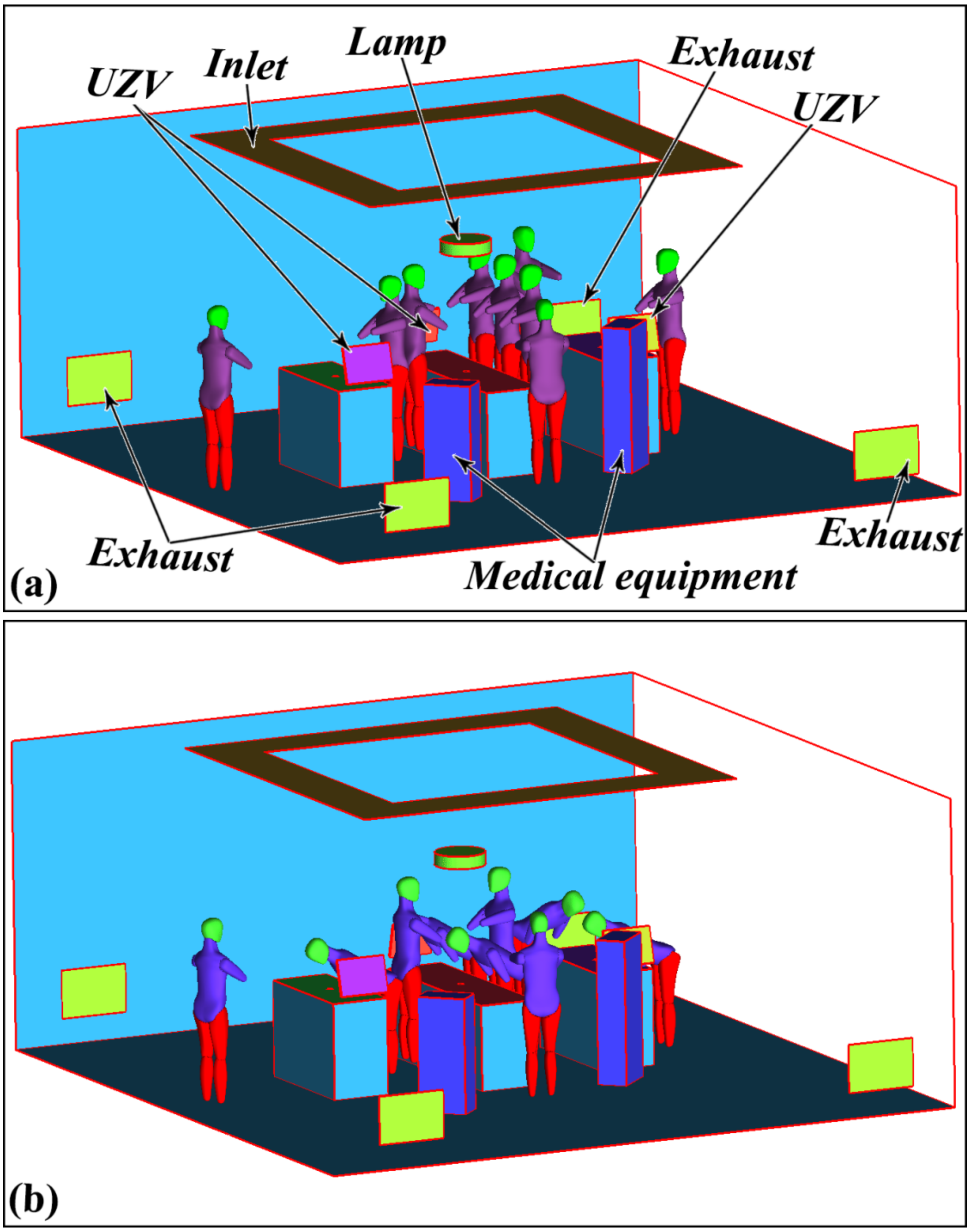


Fig. 37: Isometric view of operating room; staff members in a) upright posture, b) bending posture

ANSYS Fluent software was used to solve the Navier-Stokes and energy equations in a 3D computational domain. The realizable k-ε turbulence model was implemented to predict the airflow and Lagrangian DRW particle tracking method was used to calculate individual trajectories of particles.

To investigate the influence of surgeon posture on the airflow field and airborne particle distribution, two different scenarios were simulated and then compared. The first scenario involved upright posture, with all surgical staff standing upright (see Fig. 1-a). The second scenario considered the bending posture, with some of the surgical staff bent over the tables to represent the most common surgeon and staff-member postures, as is shown in Fig. 1-b. The numerical models were already successfully validated against experimental data available in the literature (Sadrizadeh, Holmberg, et al., 2014; Sadrizadeh, Tammelin, et al., 2014) and thus it is not repeated here.

3. RESULTS AND DISCUSSION

The airflow field for both the examined postures is illustrated by the velocity contour plot in Fig. 2. Bending over the instrument and surgical tables may reduce OT ventilation system performance and more particles can accumulate within the surgical area. SSI risk may drastically increase since the human body, a major contaminant source, is located at the upstream in relation to the instruments and wound area.

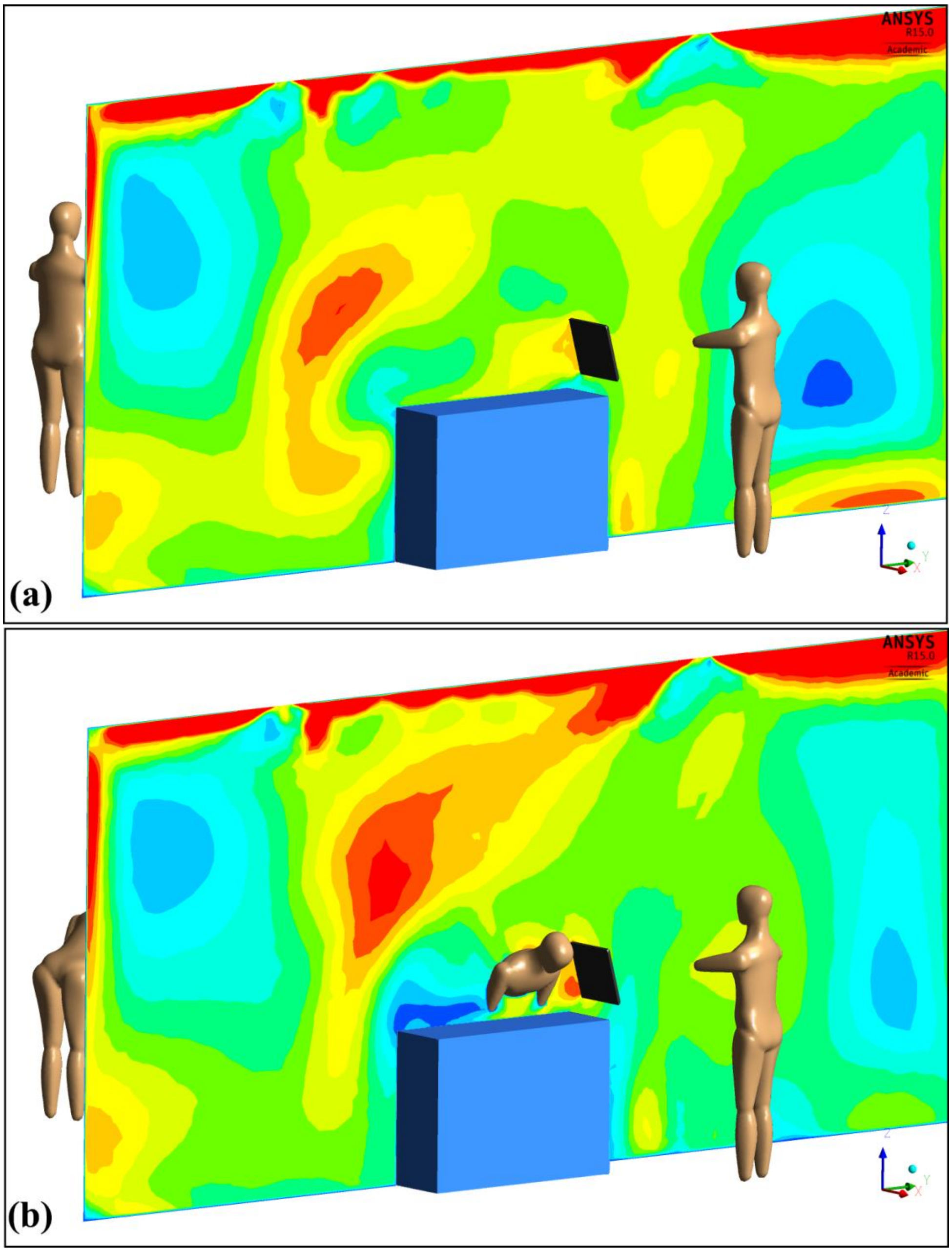


Fig. 38: Velocity contour plot; Staff members in a) upright posture, b) bending posture

Fig. 3 shows the velocity vector plots in the same location as Fig. 2 and documents how unidirectional airflows from the UZV screen could easily disrupt by improper positioning of obstacles such as the human body.

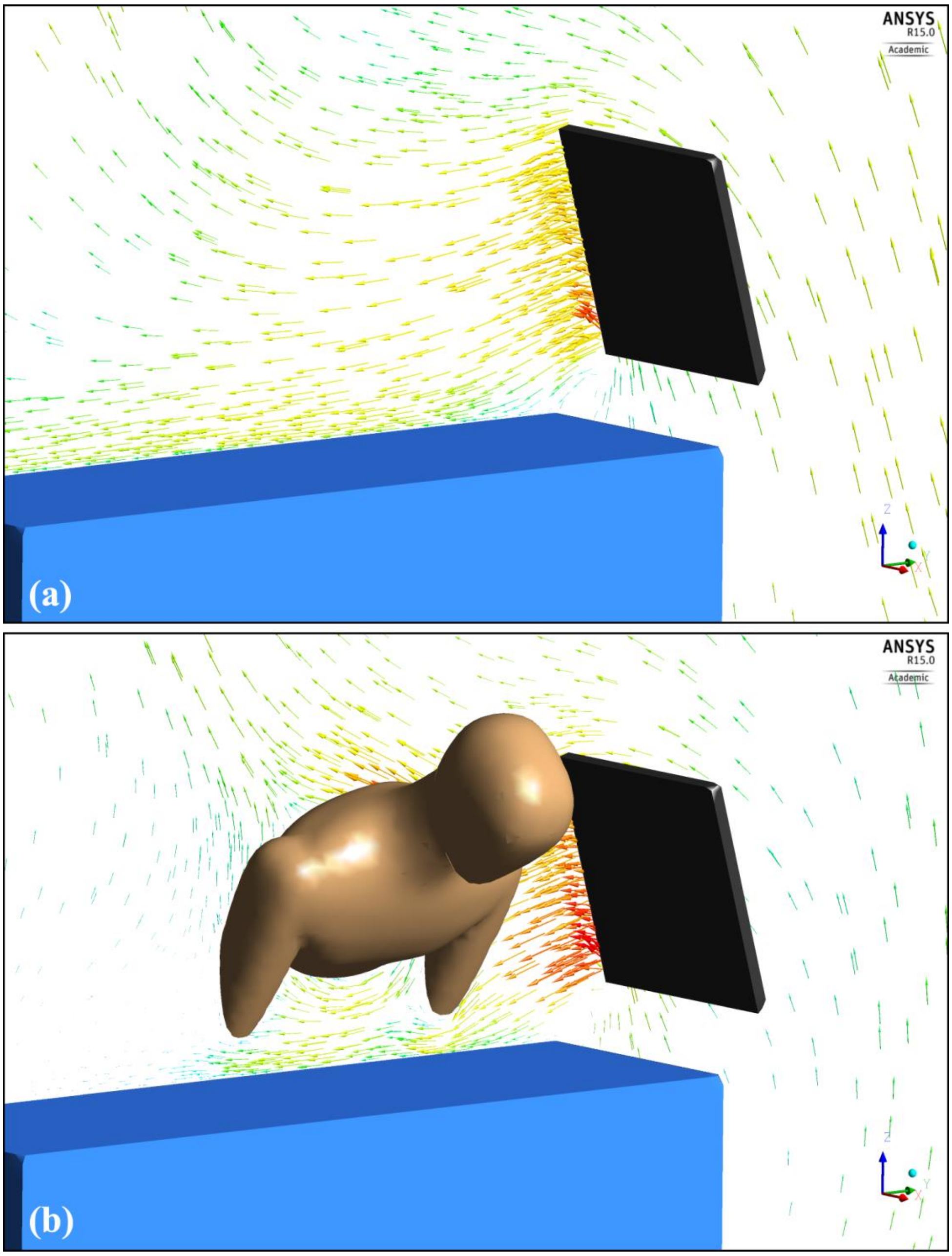


Fig. 39: Velocity vector plot; Staff members in a) upright posture, b) bending posture

Particle simulation, in upright posture scenario, shows that UZV unit can reduce volumetric particle concentration above the tables to less than 2 CFU/m^3, well below the recommended bacteriological OT air limitation of 10 CFU/m^3 for infection-prone surgeries (B. Friberg, Friberg, & Burman, 1999). By contrast, in the bended posture scenario, the CFU concentration increased beyond the CFU limitation, up to six times within the surgical area, thus increasing the chance of SSI.

4. CONCLUSIONS

In this study, the influence of staff posture on airflow structure and contaminant distribution was numerically examined. Two distinct posture scenarios of the surgical team, that is, standing and bending, were simulated and compared.

Results show that improper posture and work experience significantly reduce UZV unit performance. When all surgical staff stands upright, the lateral supply airflow from the UZV screen can introduce particle-free air, resulting in a CFU concentration of less than 2 cfu/m^3. It is highly recommended to limit activities in the area between the UZV diffusers and open wounds or instruments. To have a proper outcome, it is highly important to make sure that all OT personnel understand how work should be performed, and how the ventilation system functions, especially when UZV units are used to supplement the OT main ventilation.

5. REFERENCES

Chow, T.-T., & Wang, J. (2012). Dynamic simulation on impact of surgeon bending movement on bacteria-carrying particles distribution in operating theatre. *Building and Environment*, *57*, 68–80.

Friberg, B., Friberg, S., & Burman, L. G. (1999). Inconsistent correlation between aerobic bacterial surface and air counts in operating rooms with ultra clean laminar air flows: proposal of a new bacteriological standard for surface contamination. *The Journal of Hospital Infection*, *42*(4), 287–93.

Friberg, S., Ardnor, B., Lundholm, R., & Friberg, B. (2003). The addition of a mobile ultra-clean exponential laminar airflow screen to conventional operating room ventilation reduces bacterial contamination to operating box levels. *Journal of Hospital Infection*, *55*(2), 92–97.

Pasquarella, C., Sansebastiano, G. E., Ferretti, S., Saccani, E., Fanti, M., Moscato, U., … Signorelli, C. (2007). A mobile laminar airflow unit to reduce air bacterial contamination at surgical area in a conventionally ventilated operating theatre. *The Journal of Hospital Infection*, *66*(4), 313–9.

Rui, Z., Guangbei, T., & Jihong, L. (2008). Study on biological contaminant control strategies under different ventilation models in hospital operating room. *Building and Environment*, *43*(5), 793–803.

Sadrizadeh, S., & Holmberg, S. (2014a). Comparison of different ventilation principles in an operating suite. In *In Proceedings of 13th SCANVAC International Conference on Air Distribution in Rooms*. October 19–22, São Paulo, Brazil.

Sadrizadeh, S., & Holmberg, S. (2014b). Effect of a portable ultra-clean exponential airflow unit on the particle distribution in an operating room. *Particuology*, [In Press] doi:10.1016/j.partic.2014.06.002

Sadrizadeh, S., & Holmberg, S. (2014c). Surgical clothing systems in laminar airflow operating room: a numerical assessment. *Journal of Infection and Public Health*, *7*(6), 508–516.

Sadrizadeh, S., Holmberg, S., & Tammelin, A. (2014). A comparison of vertical and horizontal laminar ventilation systems in an operating room: a numerical study. *Building and Environment*, *82*, 517–525.

Sadrizadeh, S., Tammelin, A., Ekolind, P., & Holmberg, S. (2014). Influence of staff number and internal constellation on surgical site infection in an operating room. *Particuology*, *13*, 42–51.

Whyte, W., Hodgson, R., & Tinkler, J. (1982). The importance of airborne bacterial contamination of wounds. *The Journal of Hospital Infection*, *3*(2), 123–35.

MECHANICAL VENTILATION PERFORMANCE ASSESSMENT IN SEVERAL OFFICE BUILDINGS BY MEANS OF BIG DATA TECHNIQUES

Ricardo Enríquez[*1], Domingo Bravo[1], José Alberto Díaz[1] and Mª José Jiménez[1]

1: Efficiency in Buildings Research Unit. Renewable Division of CIEMAT
Avenida Complutense 40. Madrid 28040, Spain
Corresponding Author: ricardo.enriquez@ciemat.es

ABSTRACT

Mechanical ventilation performance is a key issue related both to energy efficiency and indoor air quality. There are several techniques for measuring ventilation rates in buildings, such as blower boor tests, flow hoods, VAV box measurements and tracer-gas techniques. From several decades ago, tracer-gas techniques are recognized as the most widely employed method to estimate air exchange rate in buildings. These methods are based on the study of the temporal evolution of the concentration of an injected gas. These methods are usually expensive and do not allow the occupancy of the building during the tests. This issue also limits severely the number of buildings to be evaluated. Despite the natural presence of CO_2 in the atmosphere, there has been a growing interest in using it as a tracer gas. It has been shown recently that it is possible to estimate the ventilation rates by means of the decay method using in-situ CO_2 measurements from transmitters available on the market.
On the other hand, there is a growing tendency to move to the smart paradigm, which usually implies to add more and more sensors to the buildings to achieve better controls, among other applications. Modern Building Energy Management Systems (BEMS) allow the access to a high quantity of data in different formats and resolutions, such as signals from sensors and actuators, set-point temperatures, schedules, digital switches, alarms, plots or reports.
In this work in-situ measurements are combined with data coming from BEMS to evaluate the mechanical ventilation performance of three different office buildings in different regions of Spain. Starting from metabolic CO_2 production from occupants of the buildings, a series of conditions are identified in to be able to evaluate the mechanical ventilation performance. Those conditions are translated into an algorithm, programmed in the Python language, which access the different sources of information to wrangle, cluster and finally calculate the mechanical ventilation performance. This operation is performed for five years of available experimental data and information in different formats, performing a search in more than 100 Gb. of information. This situation falls into the computational framework known as Big Data, as stated by the ICT community.
For the first of the buildings (located at Almería) the methodology is detailed and demonstrated through the complete series of five years of data. Thanks to the application of this technique, a mechanical ventilation rate is obtained in perfect agreement with the design values in different situations. The technique is then applied in two different buildings (located at Madrid and at Valladolid) to assess their respective ventilation rates.
Finally, some conclusions are summarised and possible improvements of the method are pointed out as future work.

KEYWORDS

Tracer-gas measurements, metabolic CO2, cheap ventilation rate assessment, Big Data, ventilation measurements during occupancy, two-point method.

1 INTRODUCTION

Ventilation performance assessment is a key issue regarding both energy efficiency and indoor air quality. Measuring ventilation rates in buildings is usually expensive and prevents from use during the testing periods, which difficult a continuous tracking of the system. On the other hand, there is a growing trend to use the so-called Big Data techniques in the context of smart buildings and cities. From the scientific point of view, there is no precise definition of what Big Data means. The first documented use of the term Big Data in the scientific context comes from NASA scientists (Cox and Ellsworth, 1997), describing a problem with computer graphics. Since then, different definitions have been provided, having all of them three common identified characteristics at the one-machine level: volume (amounts of data beyond the resources), variety (data coming from different sources and formats) and velocity (data needs to be stored in real time and processed in a brief interval). As can be seen, characteristics are still highly context dependent.
The exponential growth in streams of data sensing real world situations and human behaviors - including call detail records (CDRs), social media data (Twitter, Facebook, etc.), traffic data, spending data, government data, satellite data, and others - provides opportunities for carrying out new research and to deal with fundamental problems such as urban planning or healthy living, as stated recently by Antonelli et al. (2015). In the urban environment context, most of the works are related to ICT and mobility, such as that of De Domenico et al. al (2015), where big data is used to reduce the overall traffic in Milan by means of an adaptive routing strategy. Despite the high potential of big data in the field of energy efficiency assessment of the built environment, there are no studies up to the author's knowledge. In this work, it is explored the possibility to assess ventilation rates in buildings in the big data framework.
There are several techniques for measuring ventilation rates in buildings, such as blower boor tests, flow hoods, VAV box measurements and tracer-gas techniques. From several decades ago, tracer-gas techniques are recognized as the most widely employed method to estimate air exchange rate in buildings as stated by Sherman (1989). These methods are based on the study of the temporal evolution of the concentration of a certain injected gas. The quantitative analysis is based on the solution of a mass balance equation for each zone under consideration. Each zone is supposed to be homogeneous (fluid properties such as density and tracer gas concentration are assumed to be the same at every point within the zone), isolated (the zone only exchanges with the "outside", an space whose concentration of tracer gas is unaffected by the zone) and perfectly mixed (the tracer gas becomes instantaneously and homogeneously dispersed within the zone). Assuming that there is no source or sink of air, the ventilation rate can be obtained as (Sherman, 1990):

$$Q = \frac{1}{T}\log\frac{C_{start}}{C_{end}}, \tag{1}$$

Where Q stands for the ventilation flow of air (h^{-1}) C_{start} and C_{end} are the concentrations of the tracer gas at the start and the end of the testing period and T is the duration of the test. This calculation method assures an unbiased estimate of the average (Sherman, 1990).
Another major issue is the selection of the tracer gas. General characteristics for an adequate tracer gas can be drawn: easily measurable in terms of devices cost and non-reactive with the air. For applications involving occupied buildings issues related to toxicity and fire risk must

also be taken into account. To meet perfect-mixing hypothesis a density close to the air is also desirable. In a recent study Cui et. al (2015) performed a brief review of the different tracer gases employed from several decades, to demonstrate that CO_2 can be a good candidate for tracer gas. In this paper metabolic CO_2 is used to estimate the ventilation rate of occupied offices from wall-mounted transmitters' measurements. Information related to the presence of the user and the state of the system is obtained from different sources.
A building located at Almería (Spain) is used to validate the methodology, which is applied to two different buildings located at Valladolid and Madrid.

2 METHODOLOGY VALIDATION

2.1. Experimental set-up and building description

A building located at the Plataforma Solar de Almería (PSA) facilities at the South East of Spain (37º 05' 28'' N, 2º 21' 19'' W) has been selected. The climatic conditions are those of a semi-arid zone with high daily thermal oscillations, hot and dry summers and cold winters. The building is an East-West axis longitudinal 1110 m^2 ground level construction. It was built under a Spanish energy efficiency and solar cooling project called PSE-ARFRISOL (www.arfrisol.es, in Spanish). The building was simulated by means of dynamic building energy simulation software to optimize thermal comfort, energy demand and final consumption. Parameters such as ventilation and infiltration rates or ground reflectance were obtained from Spanish Technical Codes and literature reviews. High thermal inertia south façades, low-emisitivy doble glazing, shadowing structures (including BIPV overhangs) and night ventilation were included in the building after simulation studies. In order to promote architectural integration a double wing structure was designed to allocate solar collectors and solar chimneys. Rooftop shading in summer is also provided form such structure. Heat from solar collector is used for space heating and cooling trough radiant floor systems, assisted by air conditioning. Other elements such as air to earth heat exchanger (buried pipes) and radiant coolers were also included.
With respect to ventilation systems, two air-handling units are operated during the day, with programmed schedules. Natural night ventilation in summer is promoted through solar chimneys.

Figure 1. Different views of the building under study, showing schematically the different bioclimatic strategies implemented. To the right, a floor plan with the fully monitored offices in blue.

Figure 1 shows the constructed building together with its floor plan. Representative rooms have been selected for the study, highlighted in blue. The building has been equipped with a multipurpose monitoring system to investigate different aspects such as HVAC systems performance, modelling of the thermal response of the fabric, thermal comfort assessment or IAQ assessment, among others (Jiménez et al., 2010). The software that manages the data acquisition system is reported by Ferre et al. (2010). The building has served for previous studies such as new methods to measure the ground reflectance by Enríquez et al. (2012a), simulation model calibration in the free running mode by Enríquez et al. (2012b), or the simulation of the performance of solar chimneys by Arce et al. (2013). In addition, the Building Energy Management System (BEMS) data is also accessible at real time through an OPC system or offline through a relational database.

For our purposes, the following measurements are selected for each room: Indoors and outdoors CO_2 concentration, state of door and window (closed/not closed) and recordings on the operation of the mechanical ventilation system (Boolean variable). Indoor CO_2 concentration sensors are wall-mounted at 1.5 m. height from floor. They provide a measurement range of 0 ... 2000 ppm, an accuracy of ±(50 ppm CO_2 + 3 % of reading) and a response time of 1 minute (Vaisala, GMW115). Outdoors CO_2 concentration sensor is located in the weather station at the top of the building, provide a measurement range of 0 ... 5000 ppm, an accuracy of ± 2.5% of reading and a response time of 30 seconds (Vaisala, GMP343).

The occupants of the building are free to use the building at their own preferences; despite they have been encouraged to behave efficiently. The operation of the building is registered from 2009 and on-going, leading to a potential of five years of usable data recorded at a frequency of one minute. Data used in this study comes from two sources: a data acquisition system (DAS) and data coming from BEMS. The DAS stores signals in no-SQL csv-formatted files at a daily basis, reaching a total size of 1 Gb. at the moment of this study. Data coming from BEMS are stored in a relational database storing different kind of information: signals from more than eight thousand sensors and actuators, set-point temperatures, schedules, digital switches, alarms, plots and reports among others. The database is shared between four offices buildings and at the moment of this study reaches a size of 110 Gb. Under this context the problem can be classified in what the ICT community defines as the Big Data framework, with different sources of a big amount of data in different formats and places and in interaction with humans.

2.2. Data analysis and results

The analysis algorithm is now described. Data are separated by days which must fulfill the following criteria to be included: when the user leaves the workplace closes the door and the window (if not closed already) while the mechanical ventilation system is still in operation.

To select data to be included in the study a Python program has been developed, which has been shown to be an appropriate tool to apply under this framework. It is important to remark at this point that the computational efficiency of the algorithm is an issue to take care of, especially for scalability purposes. The following steps are applied, by order, for all the days included in the study (January, 1st, 2009 to February, 28th, 2015):

1. Check if csv-formatted data file contains all the registries (1440 minutes in one day). This can be seen as a very restricted quality filter which could be well relaxed, since one or two minutes can be interpolated. In the case under study, with five years of available data, it is more computationally efficient to discard directly those sets. For situations with less data the cost of the interpolation should be in trade-off with the lesser amount of information to process.
2. Check if the door was closed at the end of the day.
3. Check if the door was opened during the day. If so, store time series for Window/door state and indoor/outdoor CO2 concentration for that day.
4. For the days in which conditions 1-3 hold, query the database for the stop time of the mechanical ventilation. If mechanical ventilation stopped later than the closing door

and window time, retrieve information of the ventilation operation and programmed schedule.

It is worth noticing at this point that with this procedure the queries to the database are optimized, which is seen as a bottleneck in many data wrangling applications. Table 1 show the statistics for the data under consideration, once the program is executed. This operation has been performed for the central office. Different offices will eventually offer different values, mainly due to door/window operation.

At the end, near 40% of the days are potentially useful for this study, leading to 1006 available sets of data. However, additional considerations must be done related to the conditions that both CO_2 concentration and mechanical ventilation must fulfil. CO_2 is present in the atmosphere, so in order to evaluate a perturbation a concentration baseline is to be selected. Two possibilities arise at this point as a reference: indoor or outdoor CO_2. Using outdoor as reference could seem, in principle, the most convenient choice. However, there are issues such as shifting in the measurements due to different sensors and recirculation of air inside the building.

Table 1: Statistics for the data collected in the central office room for the five-years period

Data set	**Number of occurrences**	**Percentage**
Total number of days	2555	100 %
All registries	1950	76.3 %
Closed at the end of day	1768	61.2 %
Opened during the day	1093	42.7 %
Mechanical ventilation working after	1006	39.3 %

In this study both situations have been considered, and the results are presented in figure 2. The outdoors CO_2 reference concentration is taken as the daily mean of the outdoors CO2 concentration. The indoors CO_2 reference concentration has been taken as the mean for the two last hours of the day. It can be seen a bell shape centred around 4 air-changes per hour (ACH). For clarity purposes data are presented in the range 3-5 ACH, but long tail distribution appears up to the range 2-6 ACH. It should also be remarked that in many days (25% outdoors and 5% indoors) the reference concentration is higher than the measured indoors, leading to an error in eq. 1.

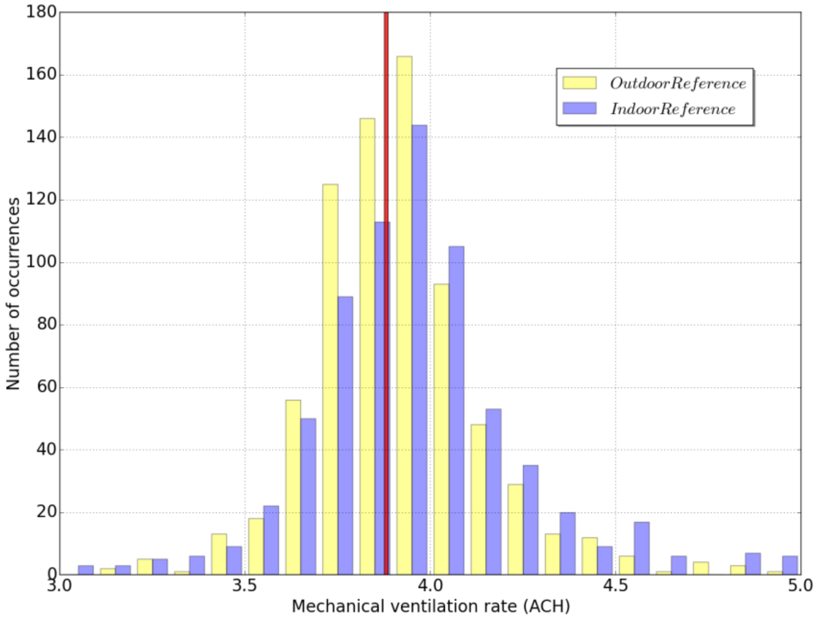


Figure 2. Mechanical ventilation rates estimates taking indoors and outdoors CO2 concentration as references. Some values appear up to 6 ACH and 2 ACH (not shown in the picture). The red bar represents the ventilation rate as designed.

To avoid these situations (long tail distribution and 25% outdoors errors) a new filter is applied to the data. Data are included only if initial indoor CO_2 concentration is 200 ppm higher than the reference one. From sensor technical data and the range of measurements performed the accuracy of the sensor is near 50 ppm. By taking a minimum of 200 ppm of difference for both measurements, initial and reference, will be far distant from accidental difference. In addition, data is considered for the study if mechanical ventilation is in operation after at least three hours since the door was lastly closed. Again, this condition can be relaxed in studies with less available data. Once filtered, outdoors CO_2 concentration reference data are near one half the data for indoors CO_2 concentration reference. Both of them are bell-shaped centred and around 4 ACH. To estimate the value for each series a Gaussian fit has been performed to the data. Table 2 resumes the parameters obtained.

Table 2. Parameters obtained from the gaussian fit for the vetilation rate, together with the design value.

Data series	Mean	Deviation
Outdoors reference	3.96	0.11
Indoors reference	4.01	0.12
Indoors, corner office	3.85	0.15
Design value	3.88	-

It can be seen that outdoors and indoors are close to the design value for the mechanical ventilation system. Both methods give accurate estimates, moreover when variance is taken into account. Due to the accuracy of the sensors, an uncertainty is associated with every value, and can be calculated by means of eq. 3. A mean uncertainty of near 0.2 ACH is obtained when sensor accuracy is considered. It is worth noting at this point that both series and the design values are indistinguishable from this point of view.
Up to now it has been shown that big data analysis techniques and indoors CO_2 concentration measurements derived from metabolic activity can be used to estimate accurately the

ventilation rate of an office. It has also been shown that outdoors or indoors CO_2 concentration can be used as reference values for the calculation with similar precision.

2.3. Cross-validation

The indoors CO_2 concentration reference is of particular interest, since it is a cheaper measurement that requires less sensors. In order to check the validity of the method a different office has been selected to check this. The office selected is that of the corner (see figure 1). Due to problems with the wall-mounted sensor, it was substituted for one of the same kind of the outdoors, so the time series is shorter. The same methodology has been applied taking indoor CO_2 concentration as the reference value. Gaussian fits for both offices are presented in figure 3. Table 2 also includes the results for the Gaussian fit, labelled as corner office.

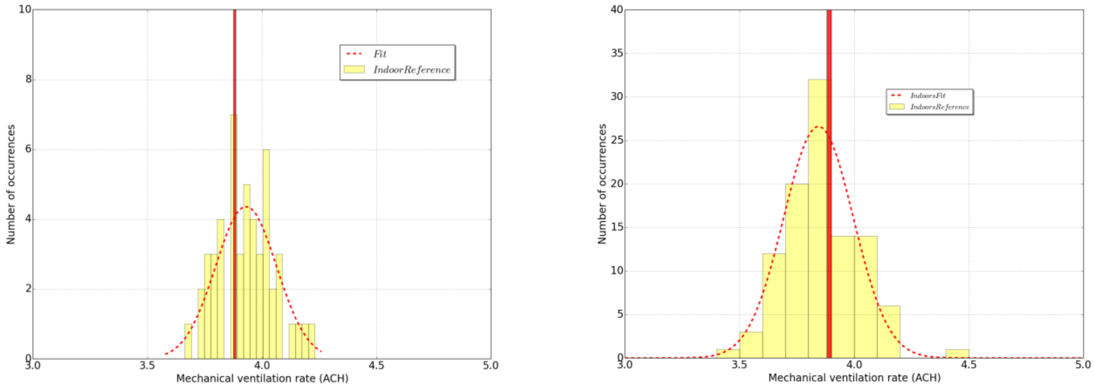


Figure 3. Mechanical ventilation rates estimates taking indoors CO_2 concentration as reference. Left: center office. Right: corner office. Gaussian fit as a red dotted line. The red bar represents the ventilation rate as designed.

It can be seen a good agreement with the situation previously described for the centred office. The ventilation rate estimate is 3.85 ACH, the closest to the design value. Again, the variance of the series and the uncertainty derived from the accuracy of the sensor makes the estimated value and the designed one indistinguishable. Additionally, it should be noticed also that a more accurate sensor is not necessary, since the uncertainty and the variance are quite similar.
It has been proven, then, in two different rooms with data series from five years that ventilation rate of a building can be estimated by means of measurements of indoors CO_2 concentration due to metabolic activity. The methodology employed need to apply techniques coming from the new Big Data field.
The question that arises at this point is that of the optimal size of the sample to estimate the ventilation rate in a confident enough way. From the statistical point of view the previous problem is that of the estimation of the mean for a Gaussian distribution. In the case under study every point can be considered an independent measurement, so a good approximation for the standard deviation of the population is the half of the uncertainty of one single measurement, which can be assessed by error propagation in eq 1.
At a confidence interval of 95% if the error estimation of the mean is to be reduced by a factor k it can be shown, after common algebraic manipulations, that minimum sample size is k^2. In the case under consideration, the uncertainty of one single measurement is close to 0.6 ACH. It can be easily seen that an uncertainty of 0.3 ACH will be reached for a sample of 4 days, 0.2 ACH for a sample of 9 days, 0.15 ACH for a sample of 16 days and so on. On the other hand, if a 0.03 ACH uncertainty should be desired, it can also easily seen that a minimum sample size of 400 would be needed. However, such accuracy is not needed for the most common commissioning applications, and a sample of 16 days is enough for tracking the

ventilation rate in a reliable way. Once the ventilation rate has been estimated for the first time, samples of four to nine days can be enough for testing periodically the building ventilation performance.
Scheduling testing time with the help of the user will reduce dramatically the testing time required. In mechanically ventilated office buildings, for example, user collaboration can be solicited and HVAC programmed properly. In the residential sector, smart meters can be incorporated and user collaboration can also be requested by means of ICT or mobile phone applications. Infiltration and natural ventilation can also be estimated, the only difference will be the selection of the CO_2 decay time according to the lower ventilation rate.

3 METHODOLOGY APPLICATION

In order to further check the usefulness of the methodology two new buildings are considered. Both of them have being designed under high energy efficiency criteria. The first one is located in Madrid and it is mechanically ventilated by means of air handling unit. It was constructed under the same research project that the previously described, so it implements the same monitoring system. The interested reader can obtain complete description of the building and associate studies in Soutullo et al. (2014) and Castillo et al. (2014). One office is selected to estimate the ventilation rate in analogous conditions to the one studied in the previous section.
Table 3 summarizes the data gathered. It can be seen that more than one half of the days the office was occupied and door and windows were closed when user(s) left the room. However, for the decay period selected there are no days where the indoors CO_2 difference is bigger than four times the uncertainty associated with one single measurement. This condition needs to be relaxed in order to have data enough. Several CO_2 differences have been chosen as filter, such as 150 ppm (10 days), 100 ppm (67 days) and 75 ppm (137 days). As it was expected, the lesser the difference the more days included. 100 ppm is selected as a trade-off between the number of days and the amplitude of the decay. It represents about twice the uncertainty of one single measurement and being below that number would compromise the reliability of the method. From the 67 days, 2 of them occur when mechanical ventilation is off, which opens the possibility to estimate natural ventilation (infiltration included).

Table 3: Statistics for the data collected in Madrid's building

Data set	Number of occurrences	Percentage
Total number of days	2526	100 %
All registries	2202	87.17 %
Closed at the end of day	1635	64.73 %
Opened during the day	1508	59.70 %
Ventilation on (100 ppm)	65	2.57 %
Ventilation off (100 ppm)	2	0.08 %

The mechanical ventilation rate obtained from experimental data is close to 3.9 ACH, physically compatible with the designed value for the installation (3.82 ACH). However, these data present variance of 1.4 ACH, which takes into account the fact that the decay amplitude was not as good as desired. If the 75 ppm where to be used, the variance would grow until 2 ACH approximately. When ventilation is off 1.06 ACH is obtained for the two days under consideration. Due to the lack of statistics enough this value should not be considered since its associated error is too big. Anyway, it suggest that under appropriate

conditions such as scheduled measurement campaigns in collaboration with users different regimes could also be explored.
The second building is located in Valladolid and also implements passive strategies. Among them: vegetation open areas under offices and a gardened roof, natural ventilation system composed by a grid system, and distributed lucernaires coupled to the air exchange system. Again, the interested reader is referred to Soutullo et al. (2015) to find a more detailed description.

Table 4: Statistics for the data collected in Valladolid's building

Data set	Number of occurrences	Percentage
Total number of days	1825	100 %
All registries	1146	62.8 %
Closed at the end of day	177	9.6 %
Opened during the day	108	5.9 %
100 ppm CO_2 difference	40	2.2 %

Table 4 summarizes the relevant data related to this building. It is a newer building than the previous two and it was monitored from the very early unoccupied stages, so only a 6% is, in principle, suitable. From the 108 days, 40 hold an indoor CO_2 decay bigger than 100 ppm. The air exchange rate estimate for this dataset is 1.29 ACH with a variance of 1.58 ACH, far from the designed value. When inspected carefully, the data showed that the information of the control system was not being recorded properly.
When applying this methodology, the ventilation rate estimation fail can be seen as a benefit. It helped to identify system malfunctions, a common source of energy waste in a complex environment.

4 CONCLUSIONS

The metabolic CO_2 has been identified as an opportunity tracer-gas to estimate ventilation rate in buildings. Big Data techniques have been identified as a necessary tool to deal with volume, variety of sources and human interaction.
Ventilation rate is identified successfully in two different offices of the same building and the methodology is elaborated, assessing its uncertainty and the minimum testing time required. The methodology is tested against five years of data to get statistics enough. It has been shown that user interaction can enhance the method, opening the gate to non-invasive scheduled testing periods.
Once the methodology has been validated it has been applied in two additional cases. In the first case the ventilation rate is identified successfully, with a high variance due to the statistics. Once more, ICT user interaction can lead to scheduled non-invasive ventilation testing periods. The possibility to study different regimes is also identified in this building. In the second case, the ventilation rate estimation fails. This fact is revealed as a benefit, helping to detect system malfunctioning and could be used as ventilation tracking system.

5 ACKNOWLEGMENTS

This work has been developed in the framework of the OMEGA-CM programme, and one of the ARFRISOL building prototypes has been used as experimental support. The OMEGA-CM programme, reference P2013/MAE2835, is a multidisciplinary R&D programme supported by the Madrid Regional Government approved within the call "Tecnologías 2013", co-financed by EU Structural Funds. The PSE-ARFRISOL, Reference PS-120000-2005-1, is Strategic Singular Scientific and Technological Project supported by the Spanish Ministry of Science and Innovation and former Ministry of Education and Science approved within the National Plan 2004-2007, co-financed by FEDER funds. The authors wish to express their gratitude for this support in the OMEGA-CM programme and the PSE-ARFRISOL project.

6 REFERENCES

Antonelli et al. 2015. Fabrizio Antonelli, Bruno Lepri, Alex 'Sandy' Pentland, Fabio Pianesi. Making Big Data work: Smart, Sustainable, and Safe Cities, EPJ Data Science article collections, 28 January 2015.

Arce et al. al, 2013. J. Arce, J. Xamán, G. Álvarez, M.J. Jiménez, R. Enríquez, M.R. Heras, 2013. "A simulation of the thermal performance of a small solar chimney already installed in a building". Journal of Solar Energy Engineering, Transactions of the ASME. 135(1), art. no. 11005.

Castillo et al., 2014. L. Castillo, R. Enríquez, M.J. Jiménez, M.R. Heras, Dynamic integrated method based on regression and averages, applied to estimate the thermal parameters of a room in an occupied office building in Madrid, Energy and Buildings, Volume 81, October 2014, Pages 337-362.

Cox and Ellsworth, 1997. Michael Cox and David Ellsworth, Application-controlled demand paging for out-of-core visualization Proceeding. VIS '97 Proceedings of the 8th conference on Visualization '97, Pages 235-ff. IEEE Computer Society Press Los Alamitos, CA, USA ©1997.

Cui et al. al, 2015. Shuqing Cui, Michaël Cohen, Pascal Stabat, Dominique Marchio, CO2 tracer gas concentration decay method for measuring air change rate, Building and Environment, Volume 84, January 2015, Pages 162-169.

De Domenico et al. 2015. Personalized routing for multitudes in smart cities. Manlio De Domenico, Antonio Lima, Marta C González and Alex Arenas. EPJ Data Science 2015, 4:1 doi:10.1140/epjds/s13688-015-0038-0.

Enríquez et al. 2012a. R. Enríquez, L. Zarzalejo, M.J. Jiménez, M.R. Heras. 2012. "Ground reflectance estimation by means of horizontal and vertical radiation measurements". Solar Energy. 86(11), pp. 3216-3226.

Enríquez et al., 2012b. R. Enríquez, M.J. Jiménez, M.R. Heras. 2012. "Analysis of a solar office building at the South of Spain through simulation model calibration". Energy procedia. 30, pp. 580-589.

Ferre et al., 2010. J.A. Ferre, M. Pasamontes, M. Castilla, M. Berenguel, M.J. Jiménez, D. Bravo, M. Pérez. 2010. In Spanish. "Desarrollo de un sistema de monitorización y control para edificios bioclimáticos en el marco del proyecto ARFRISOL". Presented at "I Congreso sobre Arquitectura Bioclimática y Frío Solar". Aguadulce, Almería, Spain. 23-26 March 2010. ISBN: 978-84-693-5141-3.

Jiménez et al., 2010. M. J. Jiménez, R. Enríquez, R. Olmedo, N. Sánchez, M.R. Heras. 2010. In Spanish. “Monitorización energética de los C-DdIs del PSE-ARFRISOL. Diseño experimental”. Presented at congress: “I Congreso sobre Arquitectura Bioclimática y Frío Solar”. Aguadulce, Almería, Spain. 23-26 March 2010. ISBN: 978-84-693-5141-3.

Sherman, 1989. M.H. Sherman, On the estimation of multizone ventilation rates from tracer gas measurements, Building and Environment, Volume 24, Issue 4, 1989, Pages 355-362, ISSN 0360-1323, http://dx.doi.org/10.1016/0360-1323(89)90030-9.

Sherman, 1990. M.H. Sherman, Tracer-gas techniques for measuring ventilation in a single zone, Building and Environment, Volume 25, Issue 4, 1990, Pages 365-374.

Soutullo et al., 2014. Soutullo, S.; Enríquez, R.; Jímenez, M.J.; Heras, M.R. (2014). Thermal comfort evaluation in a mechanically ventilated office building located in a continental climate. *Energy and Buildings*,81*, 424-429.*

Soutullo et al., 2015. Silvia Soutullo, Mª Nuria Sánchez, Ricardo Enríquez, Mª José Jiménez and Mª Rosario Heras, Thermal comfort assessment in a sustainable designed office building. Contribution accepted in AIVC 2015.

Vaisala, GMP343. Vaisala, Technical Data Sheet. GMP343 Carbon Dioxide Probe for Demanding Measurements Ref. B210641en rev. A.

Vaisala, GMW115. Vaisala, Technical Data Sheet. GMW115 Carbon Dioxide Transmitter for Demand-controlled Ventilation. Ref. B210568EN-A 2006-03.

RECOMMENDABLE SUPPLY AIR RATES FOR RESIDENTIAL HOUSING – A SIMULATION STUDY CONSIDERING CO_2 CONCENTRATION, RELATIVE HUMIDITY, TVOC EMISSIONS AND MOULD RISK

Gabriel Rojas[*1], Rainer Pfluger[*1] and Wolfgang Feist[1]

[1] *Unit for Energy Efficient Buildings*
Technikerstrasse 13, University of Innsbruck, Austria
**Corresponding author: gabriel.rojas-kopeinig@uibk.ac.at*

ABSTRACT

In an extensive simulation study using a multi-zone airflow and contaminant transport calculation software (CONTAM) recommendations for the supply air rates for residential housing were derived as input for the revision of the Austrian standard ÖNORM H 6038 (2014). The floor plan, the occupancy and the contaminant and humidity sources are modelled to represent a typical Austrian housing situation. A humidity buffering model is also implemented. Based on common thresholds for CO_2, relative humidity (r.h.) and TVOC the so-called relative threshold deviation is determined. It is used as a combined parameter to evaluate indoor air quality in terms bio-effluents, air humidity and pollutants arising from building and interior products. Additionally the potential mould risk due to high air humidity and low surface temperatures is calculated using the isopleth model.

The results suggest a supply air flow into the bedroom of 20 m³/h per person for the chosen reference climate. It represents the best compromise between exceeding the target value of CO_2 and avoiding overly dry periods during winter. If low emitting building products are used, TVOC concentrations seem not to play an important role for the definition of the supply air rates. If the floor plan permits, the implementation of the so-called extended cascade ventilation principle is recommendable. It allows a reduction of the relative threshold deviation with an air exchange rate as low as $0.3\,h^{-1}$. Prerequisite for the implementation of such low air exchange rates is a high thermal quality of the building envelope. It ensures surface temperatures that exclude potential spore germination. The same applies for the use of ventilation systems with humidity recovery. For typical recovery rates of 60%, an air exchange rate as high as $1.0\ h^{-1}$ might be required for the same reference apartment to avoid mould problems, in case the building envelope has a temperature coefficient (f_{Rsi} value) of 0.5 as frequently observed in existing buildings (thermal bridge). For residential housing, humidity recovery should therefore be limited to locations with very cold and dry winters as observed in mountain regions.

KEYWORDS

Mechanical ventilation, ventilation rates, supply air, CONTAM, simulation

1 INTRODUCTION

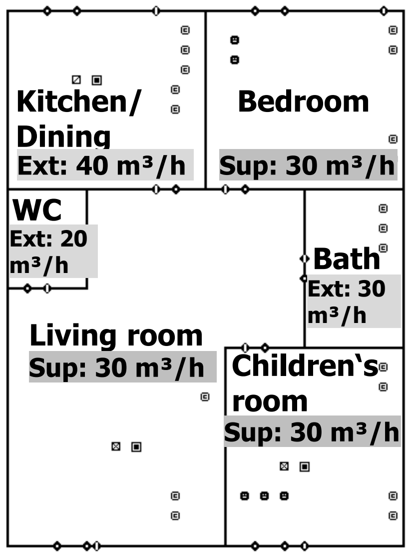
There are several aspects to consider when defining the supply air flow in residential housing. Especially in regions with cold and dry periods contradictory requirements can arise. On one hand bio-effluents from human activities as well as pollutants emitted from building products etc. have to be diluted. On the other hand excessive air exchange rates can decrease the air humidity below recommended limits. This work presents the methodology used for deriving recommendable supply air rates as used as an input for the revision of the Austrian standard for residential ventilation ÖNORM H 6038 (2014).

2 METHOD

This simulation study was performed with the multi-zone airflow and contaminant transport simulation software "CONTAM" (Walton and Dols 2010). MATLAB scripts were used for automated parameter variations (e.g. the volume flow) and for the evaluation of the simulation results.

2.1 Simulation model

A typical Austrian residential living situation was defined based on statistical data (Janik and Vollmann 2001). The floor plan has a living area of 76 m² and is shown in Figure 1 (left). The occupation was modelled with three persons (two adults, one of them working, and one child), the occupation hours and some of the chosen user behaviour are summarized in Figure 1 (right). The indoor air temperatures affect the air exchange between rooms (with open doors). They were modelled constant and they represent the average measured indoor temperature of various Passive House measurement projects. A reference climate dataset for Vienna was used.



	Occupation Schedule[a] [h]			Settings	
	Adult 1 WD/WE	Adult 2 WD/WE	Child WD/WE	Doors	Temp. [°C]
Kitchen	1.75/2	3/3	1.25/1.75	Always open	22[b]
Bed-room	8/9	10/10	--	Day: open Night: closd	21.4[b]
Living room	2.25/6	5/5	3/6.25	n.a.	22[b]
Child. room	--		9.5/10	Day: open Night: closd	21.4[b]
Bath-room	1/1	1/1	0.75/0.75	10 min per use	23
Total	13/18	19/19	14.5/18.75	--	--

Figure 1: CONTAM sketch of the floor plan (Ext=Extract Air, Sup=Supply Air), Occupation schedule and user behaviour in terms of temperature and door position (WD=weekday, WE=weekend).
[a] **based on stat. Data** (Ghassemi-Bönisch and Kronsteiner-Mann 2011)**,** [b] **based on various PH measurement studies** (Rojas, Wagner, et al. 2015; Schnieders and Hermelink 2006)

The human carbon dioxide (CO_2) emission rates were taken from the IEA ECBCS Annex 27 Report (Månsson (Ed.) 2002), being 18 l/h per adult and 12 l/h per child when awake and 2/3 of that when sleeping. The humidity production rate is also literature based (Hartmann et al. 2001; Månsson (Ed.) 2002) and is summarized in Table 1. To depict the humidity buffer potential of the walls a "two layer buffer model" was developed and implemented in CONTAM. The required parameters were calibrated with the results of dynamical hygro-thermal simulations using "Delphin" (Grunewald 2015) of a 20 cm thick concrete wall.

An additional contaminant source was added in living room, bedroom, children's room and kitchen. They represent the emissions that are independent of the occupant-presence, i.e. from building products, furniture, etc. and will be referred as TVOC source. For the reference model, the floor area (FA) specific average source strength was roughly estimated to be 500 µg/m²h. The sources were modelled as constant sources representing long-term emissions. This value was based on older field studies (Mølhave, Sparks, and Wolkoff 1996; Norbäck et al. 1995; Schulz et al. 2010) with limited information regarding the boundary conditions (air exchange rates and floor area per occupant were assumed).

Note that in light of more recent measurement projects (Rojas, Wagner, et al. 2015; Tappler et al. 2014) this value seems rather high. These studies indicate that low emitting building products (as required by the Construction Products Regulation EU 305/2011) are being used for the construction of energy efficient housing in Austria. According to the author of the mentioned study (Tappler et al. 2014), where the VOC concentrations were measured roughly after one year of occupation, a long term emission rate of 115 $\mu g/m^2_{FA}h$ would be a better representative value for occupied new homes. Simulations were also performed using this source strength.

Table 1: Humidity sources

Sources	Magnitude [g/d]	Type	Model parameters
Occupants	1900 [a]	Const.	Adult/Child wake(sleep): 55/45 (30/15) g/h
Cooking	800 [a]	Const.	Morning/Noon/Evening: 110/320/320 g/h [b]
Pers. hygiene	800 [a]	G_0*exp(-kt)	G_0=325 g/h; k=0.12 h^{-}1; 2x /(day pers.)
Plants, etc	670 [a]	Const.	0.365 g/(h m^2_{FA})
Laundry	1150 [a]	G_0*exp(-kt)	G_0=68 g/h; k=0.5 h^{-}1; 1x/(week pers)

[a] based on (Hartmann et al. 2001), [b] daily distribution according to (Månsson (Editor), 2002).

In- and exfiltration through the building envelope was modelled in all rooms with external walls (living room, bedroom, children's room and kitchen) with two "leakage-openings" per room (stack effect) resulting in an exfiltration of 0,6 h^{-1} at 50 Pa over-pressure (n_{50}-value).

2.2 Evaluation method

The simulation results for the various evaluation parameters (CO_2, r.h., TVOC) are plotted room by room for the evaluation period (1. Dec – 1. Mar) and only for hours with occupancy. The resulting area ("C" in Figure 2) between threshold and cumulative distribution function is a measure for the duration <u>and</u> magnitude of the respective threshold deviation. It is equivalent to an area resulting from an (hypothetical) constant value "A" above the threshold (e.g. 162 ppm in Figure 2). For the parameter CO_2 it means, that having a concentration of 700 ppm during the entire evaluation period would be rated equally to having a concentration of 1600 ppm during 10% of the evaluation period. Note that the use of the area as an evaluation measure implies a linear weighting of duration and magnitude of a threshold exceedance. Ideally, a physiological derived weighting (not yet established) should be used.

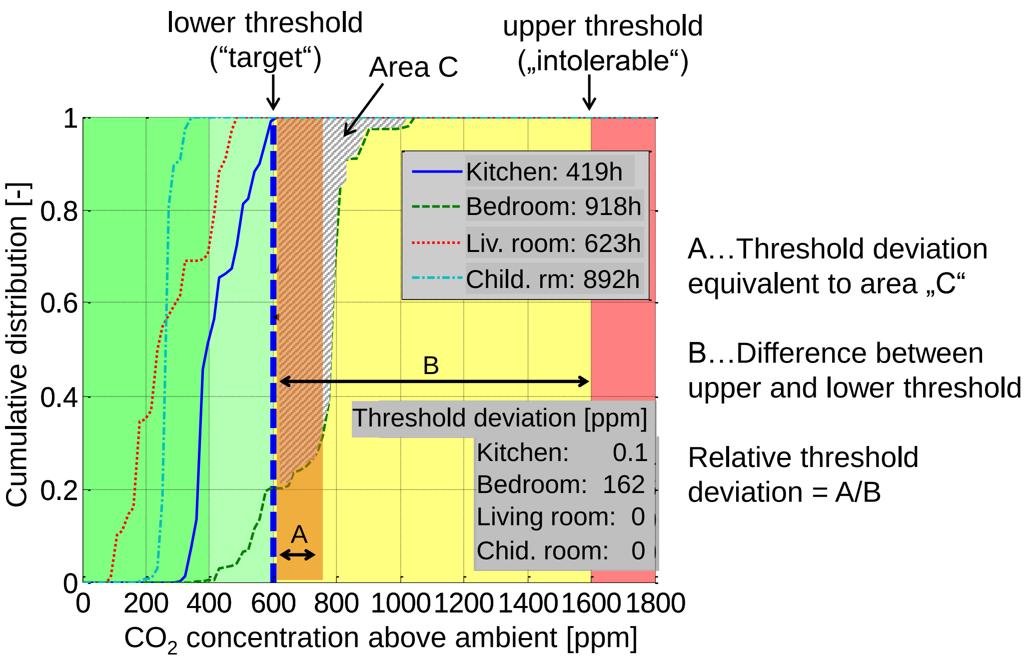


Figure 2: Cumulative distribution of CO_2 concentration of kitchen (blue), bedroom (green), living room (red) and children's room (cyan). Additionally the principle for determination of the threshold deviation is illustrated.

The ratio of this value "A" and the "bandwidth" or the "tolerance" of the threshold of the respective evaluation parameter ("B" in Figure 2) gives the relative threshold deviation. It can be calculated for each evaluation parameter and allows a direct comparison or even summation. For this, different aspects like comfort or physiological impairment (health related aspects) must not be mixed. The herein chosen threshold values (target, intolerable) aim to evaluate possible physiological impairments resulting from a given ventilation situation. They were derived from existing literature and standards and are listed in Table 2.

Table 2

Evaluation criteria[1]	CO_2^{abs} [ppm]	r.h. [%]	TVOC [mg/m³]
Target value	<1000	>30	<0.3
Temporarily tolerable	1000-2000	20-30	0.3-3
Intolerable	>2000	<20	>3

[1] Threshold values for CO_2 and TVOC based on (Heinzow and Sagunski 2007; Lahrz, Bischof, and Sagunski 2008). Target value for r.h. according to EN 13779:2004 and ÖNORM B 8110-2 Bbl 4. Note: The lower r.h. threshold is frequently disputed, but growing evidence of long term effects are being reported, e.g. (Hahn 2007; Pfluger et al. 2013; Wolkoff and Kjærgaard 2007).

To evaluate the mould growth potential, the risk of spore germination is determined as a function of the temperature, r.h. and the exposure time by the isopleth model (Sedlbauer 2001). The isopleth-number is calculated for various ventilation rates and different "qualities" of the building envelope (thermal bridges). The thermal quality of the construction is quantified for this purpose with the temperature coefficient (f_{Rsi}). According to the German standard DIN 4108-2 it is defined as

$$f_{Rsi} = \frac{T_{si} - T_{amb}}{T_i - T_{amb}} \quad (1)$$

with T_{si}, T_i and T_{amb} being the internal surface temperature, the internal room temperature and the ambient temperature. Given T_i and T_{amb}, the internal surface temperature can simply be calculated. To account for damping due to the thermal mass the 24h moving average of the ambient temperature is used for the calculation of T_{si}. The appropriateness of using a 24h moving average was tested by comparing the results of this quasi-stationary approach and a full dynamical hygrothermal simulation (using "Delphin") for two exemplary cases (concrete wall with 3 cm and 20 cm EPS-insulation).

3 RESULTS

Figure 3 shows the relative threshold deviation for CO_2, r.h. and TVOC as a function of the supply air volume flow into the bedrooms (parents and children's room). For the given reference case the target values cannot be met for all evaluation criteria concurrently. The sum exhibits a minimum at around 30-40m³/h for the bedroom (Figure 3, left). It represents the best "compromise" between high CO_2-concentrations (at low supply air rates) and periods with overly dry air (at high supply air rates).

Looking at the living room the minimum is at even lower volume flow rates, indicating that it is being over-ventilated compared to the bedrooms. This is due to the fact that the living room is being indirectly supplied with supply air from both bedrooms via the overflowing air. This floor plan, like most modern floor plans, allows the application of the so-called extended cascade ventilation principle (Rojas, Pfluger, and Feist 2015; Sibille and Pfluger 2013; Sibille et al. 2013).

Figure 4 shows that the sum of the relative threshold deviation is reduced in all rooms when the extended cascade ventilation is applied, i.e. no supply air outlet is provided in the living room. Note that the TVOC target value (<0.3 mg/m³) is not always fulfilled. With the assumed emission sources, the TVOC concentration needs to be considered when defining the supply air rate (see also living and children's room in Figure 3). But in light of recent results of an IAQ measurement study involving newly built energy efficient homes after one year of occupation (Tappler et al. 2014), the modelled emission rate of 500 µg/m²$_{FA}$h seems to be too high to represent a typical Austrian residential housing situation.

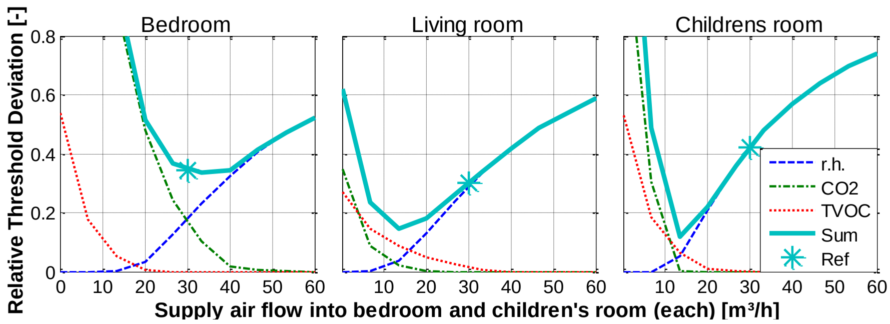


Figure 3: Relative threshold deviation as a function of the supply air flow into the bed and children's room. The supply air flow into the living room was kept constant at 30 m³/h.

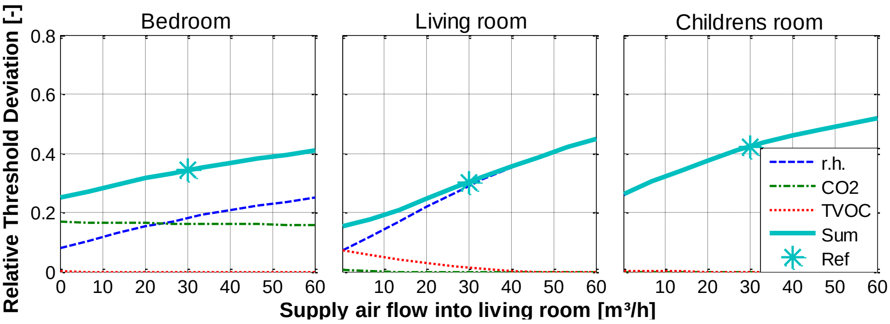


Figure 4: Relative threshold deviation as a function of the supply air flow into the living room. The supply air flow into the bed and children's room was kept constant at 30 m³/h.

Therefore additional simulations were performed with an emission rate of 115 µg/m²$_{FA}$h. As one can see in Figure 5 the target value for the TVOC concentrations are fulfilled even for quite low supply air rates. This means that under these conditions, the supply air rates should be defined considering the necessary dilution of bio-effluents (CO_2 concentration) while maintaining a proper humidity level. The dilution of contaminants from building products does not seem to be the "driving" criteria for the definition of ventilation rates when low emitting building products are used.

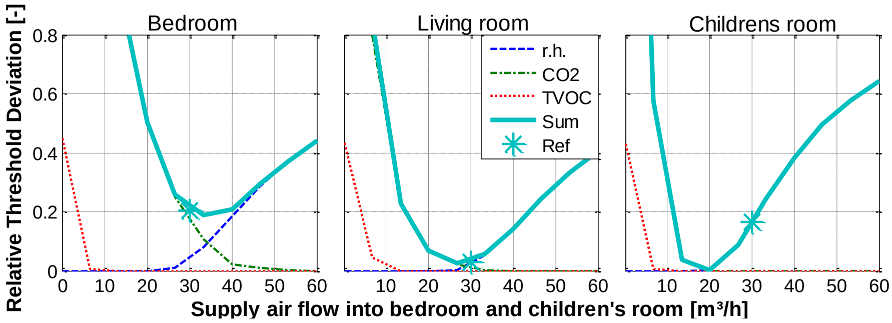


Figure 5: **Relative threshold deviation as a function of the supply air flow into the bed and children's room for TVOC emissions rate** of 115 µg/m²$_{FA}$h. **The extended cascade ventilation principle is applied, i.e. there was no additional supply air flow into the living room.**

A floor plan where the extended cascade ventilation principle cannot be applied, was also investigated by separating the living room from the hallway in the reference model. Here, the sum of the relative threshold deviation has its minimum somewhere between 5 and 30 m³/h depending on the assumed door usage (between living room and hallway). Therefore a supply air flow into the living room of 30 m³/h is recommended for the assumed occupation.

3.1 Variations in climate and occupancy

In a previous study using the same methodology and reference model (Rojas, Sibille, and Pfluger 2012), the sensitivity of the result on various model parameters (user behaviour and climate) was investigated. The climate or more precise the content of water vapour in ambient air influences the resulting threshold deviation strongly (see Figure 6). Note that the results are shown for the supply air rates as defined for the reference model (total 90 m³/h). They might not necessarily represent the setting with the minimal possible relative threshold deviation. Especially for locations with higher vapour content (mild winters) a higher air exchange rate would further reduce the threshold deviation.

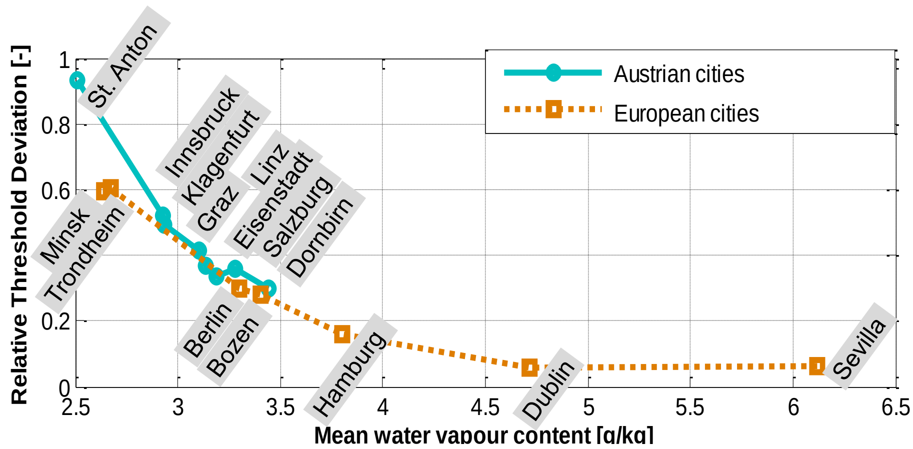


Figure 6: Relative threshold deviation as a function of the mean water vapour content in the ambient during the evaluated winter period. The occupancy-weighted average of kitchen, living-, bed- and children's room is shown. The respective city of the used climate data set is also noted in the graph.

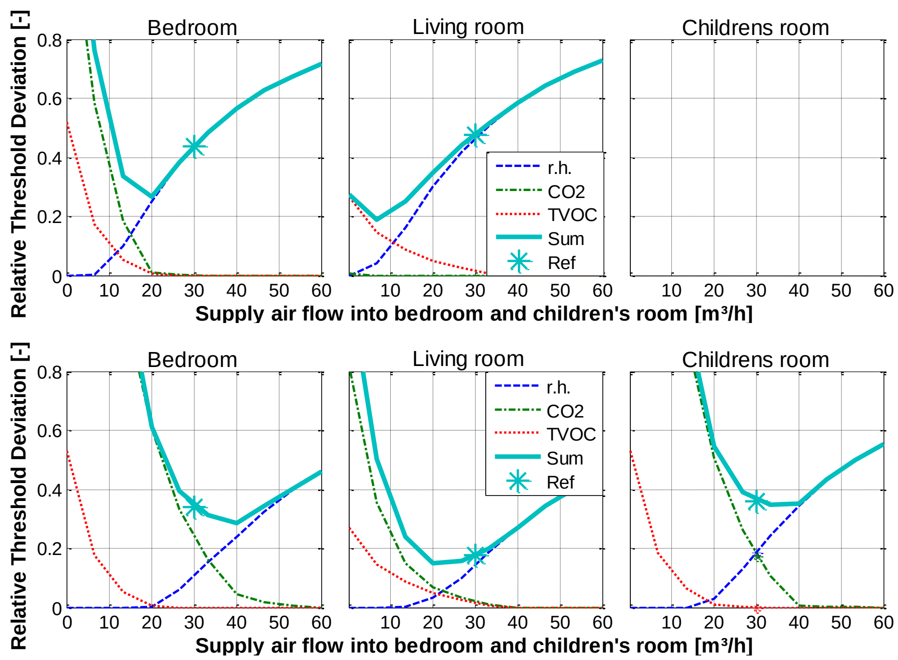


Figure 7: Relative threshold deviation as a function of the supply air flow into the bed and children's room for a one-person household (top) and a five-person household (two adults and three children). The supply air flow into the living room was kept constant at 30 m³/h.

Various simulations with different occupancy profiles were performed. Figure 7 shows the two "extreme" cases with a one and five person occupancy profile. It confirms that a person-specific supply air rate of 20 m³/h reduces the relative threshold deviation in the bedroom. For the children's room a supply air rate of 15 m³/h per child results in the lowest relative threshold deviation (also considering Figure 3). Note that the modelled CO_2 emission represents a 10 year old child.

3.2 Ventilation system with humidity recovery

As seen from the simulation results without humidity recovery it is practically not possible to continuously meet the recommended target values for CO_2 concentration and r.h. especially in climates with cold and dry winters. In principle this contradiction can be relieved by some sort of humidification measures. In the following, the effect of installing a ventilation system with humidity recovery (the most energy efficient form of humidification) was investigated. A typical humidity recovery rate of 60% defined as

$$\eta_x = \frac{x_{\sup} - x_{amb}}{x_{ext} - x_{amb}} \qquad (2)$$

with x_{sup}, x_{ext} and x_{amb} being the vapour content of the supply air, extract air and ambient air respectively, was integrated in the CONTAM model. A humidity dependent control of the ventilation rate was not implemented. The energetic impact of using a ventilation system with humidity recovery is thoroughly evaluated in (Schnieders 2008).

As seen in Figure 8 the relative threshold deviation can be reduced to zero with a supply air flow of 40-50 m³/h for bedroom and children's room (each). From the point of view of meeting the target values for CO_2, TVOC and r.h. a further increase would not be required. It would only increase the ventilation losses and the electric power consumption of the ventilation unit. Nevertheless, depending on the humidity recovery rate and the quality of the thermal envelope a higher ventilation rate might still be needed to avoid the risk of mould growth.

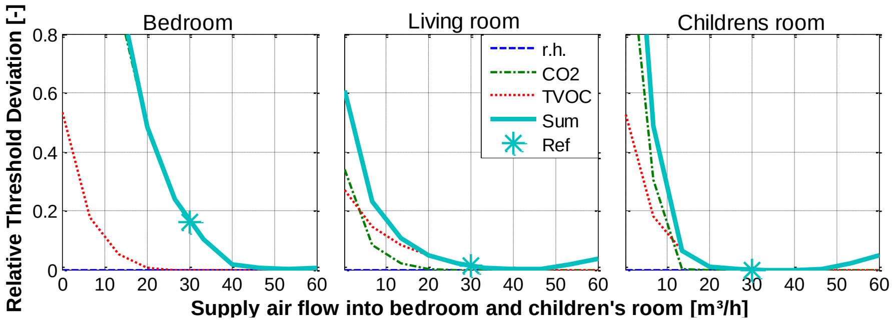


Figure 8: Relative threshold deviation as a function of the supply air flow into the bed and children's room for a ventilation system with a humidity recovery rate of 60%. The supply air flow into the living room was kept constant at 30 m³/h.

3.3 Considering mould risk

Therefore the risk of spore germination was determined as a function of the ventilation rate and the quality of the building envelope (thermal bridges) quantified by the temperature coefficient (f_{Rsi} value). Figure 9 shows the isopleth-number as a function of the total supply air flow assuming a ventilation system with a humidity recovery rate of 60%. The distribution of the supply air into the three supply air rooms was set according to the results shown in Figure 8, i.e. 45.4% into bedroom and 27.3% each into children's room and living room. A minimal supply air flow of 90 m³/h is required to avoid mould growth (in the kitchen) for a f_{Rsi} value of 0.9. This f_{Rsi} value represents an envelope of high thermal quality, i.e. high insulation and only minor thermal bridges as required for the Passive House standard. (An exception is the edge bond of the windows, where f_{Rsi} values around 0.7 are possible even for Passive House certified windows. Here a spore germination is less likely and not critical as it would be easy to clean.) Therefore a ventilation rate of 110 m³/h as derived by the CO_2 and r.h. criteria (see above), would also fulfill the "no-mould" criteria for a humidity recovery rate of 60%. If the envelope (thermal bridge) has a f_{Rsi} value of 0.7 (as required within DIN 4108-2 for new buildings in Germany) the volume flow has to be increased to >130 m³/h to avoid mould growth. For f_{Rsi} value of 0.5 (as often encountered in older buildings) the ventilation rate should be at least 180-200 m³/h to obtain an isopleth-number <1. As a consequence, the use of humidity recovery systems should be limited to buildings with a high thermal quality, i.e. low energy or passive houses.

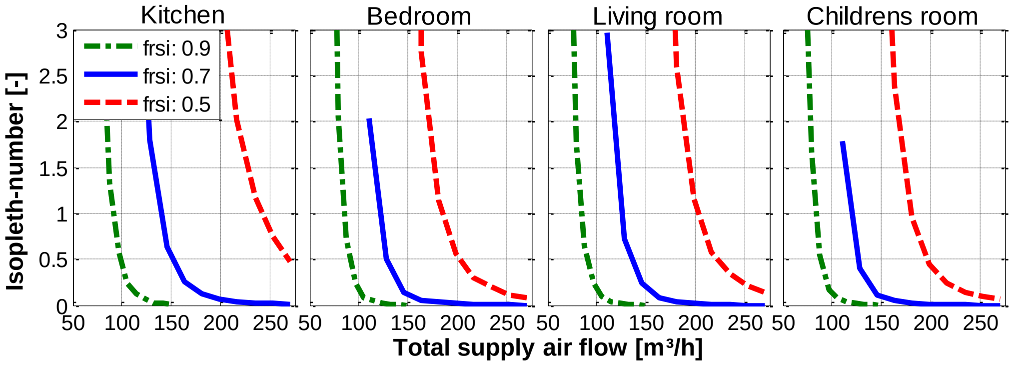


Figure 9: Isopleth-number as a function of the total supply air flow for a ventilation system with a humidity recovery rate of 60% for three different quality levels of the building envelope. A value <1 indicates that the biohygrothermal conditions for spore germination are fulfilled and mould growth is possible.

The mould risk evaluation for the extended cascade ventilation without humidity recovery is shown in Figure 10. According to the relative threshold deviation a total supply air flow of 60 m³/h (40 m³/h into the bedroom and 20 m³/h into the children's room) would be "optimal" (see Figure 5). The results clearly indicate potential mould growth in kitchen and living room for a f_{Rsi} value of 0.5 (e.g. existing building). For buildings with a f_{Rsi} value of >0.7 (which is generally met in new buildings in Austria and Germany) this simulation results show no risk of spore germination at an volume flow of 60 m³/h. Therefore, the cascade ventilation principle is only recommendable for new buildings or existing buildings which have been appropriately refurbished.

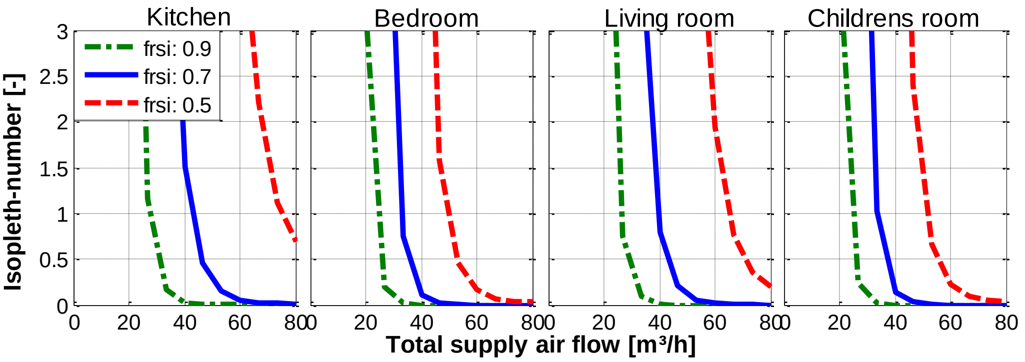


Figure 10: Isopleth-number as a function of the total supply air flow for the extended cascade ventilation principle and no humidity recovery. A value <1 indicates that the biohygrothermal conditions for spore germination are fulfilled and mould growth is possible.

4 CONCLUSIONS

This work derives recommendations for supply air rates in residential housing based on simulations using a multi criteria evaluation method considering bio effluents, emissions from building products and relative humidity. Additionally the mould growth potential is assessed using the isopleth model. The results were incorporated in the revision of the Austrian standard for residential ventilation ÖNORM H 6038 (2014).

Based on this study a supply air flow into the bedroom of 20 m³/h per person is recommended for the chosen reference climate (Vienna). It represents the best compromise between exceeding the target value of CO_2 and avoiding overly dry periods during winter. Pollutants arising from building products and furniture (herein modelled as "TVOC source") seem not to play an important role for the definition of the supply air rates, if low emitting building products are used. For floor plans where the living room is supplied via overflowing air from the bedrooms (more than one bedroom required), no additional supply air into the living room is required. This results in a low air exchange rate for the dwelling. The extended cascade ventilation principle is not recommended for buildings with low internal surface temperatures (thermal bridges) due to mould risk. In this case or simply if the floor plan does not allow the implementation of the extended cascade principle, a supply air rate of 30 m³/h is recommended for the living room (for the investigated 3 person household).

Depending on the water vapour content of the ambient air, these values might need to be adapted. Especially for regions with very cold and dry winters (e.g. mountain regions), ventilation systems with humidity recovery are advisable. Prerequisite is a good thermal building envelope with no heavy thermal bridges ($f_{Rsi} > 0.7$). Otherwise, a potential risk of mould growth exists or a very high (and energy inefficient) increase in air exchange rate is needed.

5 ACKNOWLEDGEMENTS

This work was done within the project “Doppelnutzen” funded by the Austrian Ministry for Transport, Innovation and Technology (bmvit) within the framework of “Haus der Zukunft plus”. For further information see: http://www.hausderzukunft.at/results.html/id6370

6 REFERENCES

Ghassemi-Bönisch, Sonja, and Christa Kronsteiner-Mann. 2011. Zeitverwendungserhebung 2008/09. Retrieved (http://www.statistik.at/web_de/statistiken/soziales/zeitverwendung/index.html).

Grunewald, John. 2015. “Delphin.” Retrieved June 29, 2015 (http://bauklimatik-dresden.de/delphin/).

Hahn, N. Von. 2007. “„ Trockene Luft “ Und Ihre Auswirkungen Auf Die Gesundheit – Ergebnisse Einer Literaturstudie.” Gefahrstoffe - Reinhaltung der Luft 67(3):103–7.
Hartmann, Thomas et al. 2001. “Bedarfslüftung Im Wohnungsbau.”

Heinzow, Birger, and Helmut Sagunski. 2007. “Beurteilung von Innenraumluftkontaminationen Mittels Referenz- Und Richtwerten [Evaluation of Indoor Air Contaminants Based on Reference and Guide Values].” Bundesgesundheitsblatt, Gesundheitsforschung, Gesundheitsschutz 50:990–1005.

Janik, Wilhelm, and Kurt Vollmann. 2001. “Wohnungen Nach Haushaltsmerkmalen: GWZ-VZ 2001.” Retrieved (http://www.statistik.at/web_de/statistiken/wohnen_und_gebaeude/index.html).

Lahrz, Thomas, Wolfgang Bischof, and Helmut Sagunski. 2008. “Gesundheitliche Bewertung von Kohlendioxid in Der Innenraumluft.” Bundesgesundheitsblatt, Gesundheitsforschung, Gesundheitsschutz 51(11):1358–69. Retrieved June 13, 2012 (http://www.ncbi.nlm.nih.gov/pubmed/19043767).

Månsson (Ed.), Lars-Göran. 2002. Simplified Tools Handbook - Annex 27. edited by Lars-Göran Månsson. FaberMaunsell Ltd on behalf of the International Energy Agency.

Mølhave, L., LE Sparks, and P. Wolkoff. 1996. "The Danish Twin Apartment Study–Part II: Mathematical Modeling of the Relative Strength of Sources of Indoor Air Pollution." Indoor … 18–30. Retrieved September 1, 2012 (http://onlinelibrary.wiley.com/doi/10.1111/j.1600-0668.1996.t01-3-00003.x/abstract).

Norbäck, D., E. Björnsson, C. Janson, J. Widström, and G. Boman. 1995. "Asthmatic Symptoms and Volatile Organic Compounds, Formaldehyde, and Carbon Dioxide in Dwellings." Occupational and environmental medicine 52(6):388–95. Retrieved (http://www.pubmedcentral.nih.gov/articlerender.fcgi?artid=1128243&tool=pmcentrez&rendertype=abstract).

Pfluger, R., W. Feist, A. Tietjen, and A. Neher. 2013. "Physiological Impairments at Low Indoor Air Humidity." Gefahrstoffe - Reinhaltung der Luft 03:107–8.

Rojas, Gabriel, Rainer Pfluger, and Wolfgang Feist. 2015. "Cascade Ventilation – Air Exchange Efficiency in Living Rooms without Separate Supply Air." Energy and Buildings. Retrieved (http://dx.doi.org/10.1016/j.enbuild.2015.02.014).

Rojas, Gabriel, Elisabeth Sibille, and Rainer Pfluger. 2012. "Sensitivitätsanalyse Zur Raumluftqualität Mit Wohnraumlüftung." Pp. 323–30 in enova - Nachhaltige Gebäude. Forschungs- und Studienzentrum Pinkafeld.

Rojas, Gabriel, Waldemar Wagner, Jürgen Suschek-Berger, Rainer Pfluger, and Wolfgang Feist. 2015. "Applying the Passive House Concept to a Social Housing Project in Austria – Evaluation of the Indoor Environment Based on Long-Term Measurements and User Surveys." Advances in Building Energy Research (June):1–24. Retrieved (http://www.tandfonline.com/doi/full/10.1080/17512549.2015.1040072).

Schnieders, Juergen. 2008. Energetische Bewertung von Wohnungslüftungsgeräten Mit Feuchterückgewinnung. Darmstadt.

Schnieders, Juergen, and Andreas Hermelink. 2006. "CEPHEUS Results: Measurements and Occupants' Satisfaction Provide Evidence for Passive Houses Being an Option for Sustainable Building." Energy Policy 34(2):151–71. Retrieved June 2, 2014 (http://linkinghub.elsevier.com/retrieve/pii/S0301421504002708).

Schulz, Christine et al. 2010. Kinder-Umwelt-Survey (KUS) 2003/06.

Sedlbauer, Klaus. 2001. "Vorhersage von Schimmelpilzbildung Auf Und in Bauteilen." TU Muenchen.

Sibille, Elisabeth, and Rainer Pfluger. 2013. "Optimization of Dwelling Floor-Plan Configuration for Cascade Ventilation." in Tagungsband:17. Int. Passivhaus-Tagung, edited by Wolfgang Feist. Passivhaus Institut.

Sibille, Elisabeth, Gabriel Rojas, Mattias Rothbacher, R. Pfluger, and H. K. Malzer. 2013. "'Doppelnutzen' - Komfort- Und Kostenoptimierte Luftführungskonzepte Für Energieeffiziente Wohnbauten [Final Report: Extended Cascade Ventilation - 'Double Use']." Retrieved (http://www.hausderzukunft.at/publikationen/view.html/id1158).

Tappler, Peter, Hans-Peter Hutter, Herwig Hengsberger, and Wolfgang Ringer. 2014. Lüftung 3.0 - Bewohnergesundheit Und Raumluftqualität in Neu Errichteten, Energieeffizienten Wohnhäusern [Ventilation 3.0- Occupant Health and Indoor Air Quality in Newly Build Energy Efficient Housing]. Vienna: Institut für Baubiologie und Bauökologie. Retrieved (http://innenraumanalytik.at/pdfs/lueftung_2014.pdf).

Walton, George N., and W. Stuart Dols. 2010. CONTAM User Guide and Program Documentation. NIST. Retrieved (http://www.bfrl.nist.gov/IAQanalysis/CONTAM/userguide.htm).

Wolkoff, Peder, and Søren K. Kjærgaard. 2007. "The Dichotomy of Relative Humidity on Indoor Air Quality." Environment International 33:850–57.

VENTILATION, INDOOR AIR QUALITY AND LEARNING IN SCHOOLS

Pawel Wargocki

International Centre for Indoor Environment and Energy, DTU Civil Engineering, Technical University of Denmark, Nils Koppels Alle, 2800 Kgs. Lyngby
paw@byg.dtu.dk

ABSTRACT

Studies show that environmental conditions in schools are often inadequate, even in developed countries, and that they are frequently much worse than in office buildings. Outdoor air supply rates in schools are considerably lower than in offices, in many cases even lower than those observed in dwellings. Research studies show that inadequate classroom ventilation can reduce learning abilities of pupils in elementary school as measured in form of improved performance of typical school tasks requiring skills in maths and language and in form of the tests used by educational departments to examine the progress in teaching. Inadequate ventilation rates will also decrease illness absence, which may be detrimental for learning process. Children in classrooms with poor ventilation are less attentive and less concentrated. The negative results on learning may have significant economic implications, which are not immediate and can be first harnessed in the future when children join the job market, but some of them may be also seen immediately in form of the reduced absence rate of teachers as well as parents or caregivers. The ventilation solutions to improve classroom ventilation will largely depend on the climatic conditions but in the case of moderate climates mechanical ventilation and hybrid systems seem the most attractive solutions.

KEYWORDS

Elementary schools; indoor air quality; ventilation; learning

1 INDOOR AIR QUALITY AND VENTILATION IN SCHOOLS

Studies show that the environmental conditions in schools are often inadequate, even in developed countries, and that they are frequently much worse than in office buildings. For example, measurements in 39 schools in Sweden showed that 77% of schools did not meet building code regulations (Smedje and Norbäck, 2000). The most common defects in schools include insufficient outside air supplied to occupied spaces; water leaks; inadequate exhaust air flows, poor air distribution or balance; and poor maintenance of heating, ventilation and air-conditioning (HVAC) systems, as indicated by the analysis of 88 National Institute of Occupational Safety and Health (NIOSH) Health Hazard Evaluation Reports for educational facilities in the USA where formal complaints had been registered (Angell and Daisey, 1997; Daisey et al., 2003).

Outdoor air supply rates in schools are considerably lower than in offices, in many cases even lower than those observed in dwellings (Brelih, 2012; Dimitroulopoulou, 2012). They are also often much lower than they should be according to current recommendations for classrooms

(Daisey et al., 2003, Dijken et al, 2005). For example, ASHRAE Standard 62.1 (2014) recommends for classrooms 5 L/s per person plus 0.6-0.9 L/s per m^2floor. Low ventilation rates often lead to carbon dioxide (CO_2) levels being well above the recommended level of 800-1,000 ppm (sometimes 1,400 ppm) during school hours (Sowa, 2002; Dijken et al., 2005; Boxem et al., 2006; Santamouris et al., 2008; Wyon et al., 2010; Gao et al., 2014), implying that the concentration of other pollutants, not only the bioeffluents from children for which CO_2 is a good indicator, will be high, and that classroom air quality is consequently poor.

In recent air quality measurements in 320 schools in Denmark, CO_2 concentrations exceeded 1,000 ppm in more than 50% of classrooms (Menå and Larsen, 2010, Clausen et al., 2014; Toftum et al., 2015). The air quality in these classrooms did not meet the requirements of the Danish Building Code nor the Danish Working Environment Authority because the outdoor ventilation rates were too low. For comparison, similar measurements in Norway and Sweden showed that only in no more than 20% of classrooms were the CO_2 concentrations above 1,000 ppm. Many studies have also reported high concentrations of particles in classrooms (e.g., EFA, 2001; Dijken et al., 2005; Simoni et al., 2010).

2 EFFECTS ON LEARNING

Majority of studies examining the effects of indoor air quality and ventilation on the performance of schoolwork and learning used psychological and neurobehavioral tests. These tests examine different skills needed for proper learning, such as the ability to concentrate and memorize (Myhrvold et al. 1997; Bako-Biro et al. 2010; Ribic 2008). They used also shorter tests examining the ability to read, comprehend and calculate (Wargocki and Wyon 2013). For example, a classroom study by Myhrvold et al. (1996) found a weak association between CO_2 levels and simple reaction time suggesting a positive effect of increased ventilation on performance. In studies reported by Wargocki and Wyon (2013) pupils performed arithmetical calculations and language based tasks under different conditions of air quality achieved by changing ventilation rate between 3 and about 10 L/s per person (Figure 1). The speed at which the tasks were solved was improved with increased ventilation, but there were no effects on errors. Similar results were observed by a recent study, which copied the experimental approach of Wargocki and Wyon (Petersen et al., 2015). Also in a study of Bako-Biro et al. (2010), the performance of range of cognitive tasks was improved as well as time needed to solve simple math tests was reduced when ventilation rate was increased from about 0.3-0.5 to 13-16 L/s per person. Ribic (2008) observed improved performance on d2-test, a standard test for measuring concentration, when CO_2 concentration was reduced from around 3,800 to 870 ppm (absolute level). In contrast to these observations, Mattsson and Hygge (2005) did not observe any positive effect of operation of particle air cleaners on psychological tests despite the measureable effect on classroom levels of particles.

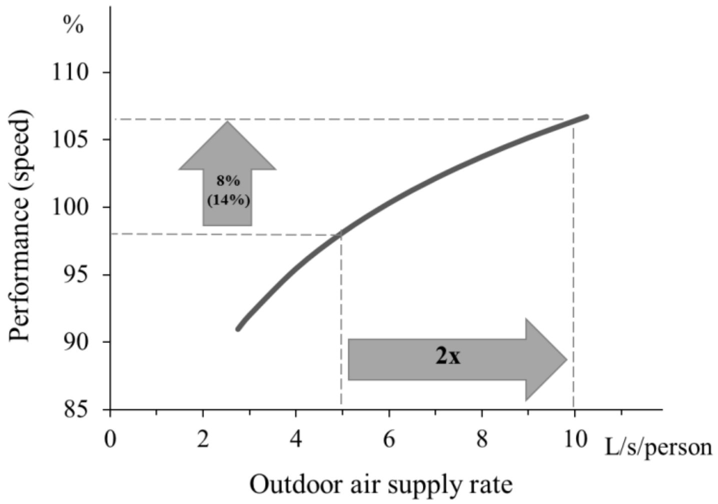


Figure 1: Performance of math and language-based tasks under different levels of outdoor air ventilation rates (Wargocki and Wyon, 2013)

Although the long-term learning outcomes are expected to be affected by the absence of abilities to perform simple psychological tests and ability to read, calculate and comprehend, the connection between the progress in learning and these abilities is not very well documented. Therefore, some studies monitored long-term learning using standardized tests, which are often developed by national education departments. These tests monitor the progress in learning and benchmark both individual pupils and schools, as well as evaluate the effectiveness of teaching methods and curricula over time. Haverinen-Shaughnessy et al. (2011) showed that poor ventilation in classrooms reduced the number of pupils just passing language and math tests (Figure 2). In another study by Toftum et al. (2015), academic achievement was evaluated with the scores from a standardized Danish test scheme, adjusted for a socioeconomic reference score. The lowest national test scores were generally found for pupils in classes with CO_2 concentrations above 2,000 ppm, although the association was not significant. Pupils in schools with some means of mechanical ventilation scored on average higher in the national tests than pupils in schools with natural ventilation, probably because the efficacy of ventilation was higher.

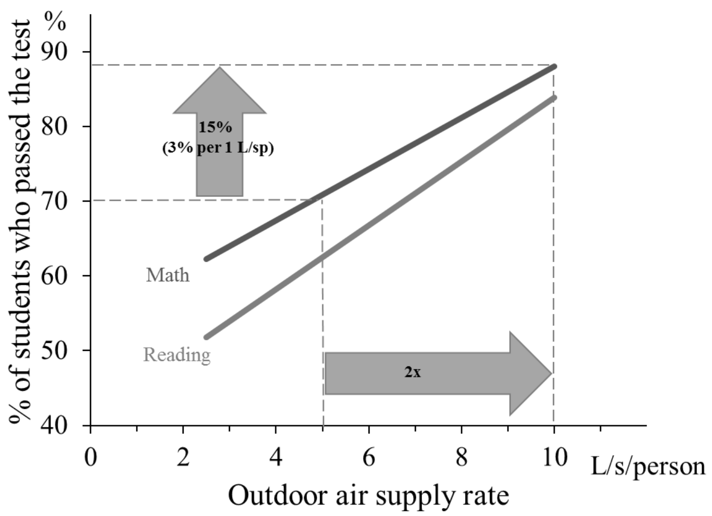


Figure 2: Performance of standard tests examining progress in math and language proficiency at different rates of ventilation with outdoor air (Haverinen-Shaughnessy et al., 2011)

The results from the previous experiments on the effects of classroom air quality on the performance of schoolwork do confirm that these effects are systematic and suggest that improving classroom air quality will have significant positive effect on some aspects of learning, both on cognitive skills and academic attainment, as well as on academic achievements. The level of this effect is not the same across different studies as might be expected but with reasonable confidence it can be assumed that doubling ventilation rate would improve the performance of schoolwork by up to 14% and each additional 1 L/s per person would increase number of students passing the tests by 3% and would reduce the absence rates by at least 1.6% (Figures 1-2).

Some studies measured illness absence and associated it with poor air quality or poor classroom ventilation to examine indirectly the effects on learning outcomes. Pilotto et al. (1997) showed in a cohort study that air pollutants from gas heaters had a negative effect on attendance at school, which was presumed to be due the result of negative effect on children's health. Berner (1993) showed an association between poor maintenance of schools and the poor academic achievement of the children attending them. Ervasti et al. (2012) found increased short-term sick leave among teachers in schools with poorly perceived air quality, while Simons et al. (2010) found high student absenteeism to be associated with poor ventilation in 2751 New York schools. Shendell et al. (2004) found student absence to decrease by 10-20% when CO_2 concentration decreased by 1,000 ppm in 434 American classrooms. However recent study of Gaihre et al. (2014) in Scottish schools showed that an increase of 100 ppm of CO_2 corresponds only to 0.2% increase in absence rates (roughly one order of magnitude lower than the data of Shendell et al. (2004)) corresponding to about 0.5 day a year in a 190–days long school year. The study of Gaihre et al. (2014) did not however find any relationship between air quality approximated by the levels of CO_2 and educational attainment measured as the percentage of class attaining the average level expected for this group. Another recent and very comprehensive study in 162 Californian classrooms observed that illness absence decreased by as much as 1.6% for each additional 1 L/s per person (Mendell et al., 2013); this is again lower than the data of Shendell et al. but higher than the data of Gaihre et al. Finally a recent work, though performed in day care centres equipped with the balanced mechanical ventilation system, found that increasing air change rate by 1 h^{-1} would reduce the number of sick days by 12% (Kolarik et al., 2015); these results were observed even though the ventilation rates were quite high in these day-care centres (CO_2 levels were <1,000 ppm, and on average around 640 ppm). The above findings suggest that increasing classroom ventilation may substantially decrease illness absence and may affect the learning experience though it is not clear whether or to which extent the short-term absence of pupils would affect academic performance (Mendell and Heath, 2005).

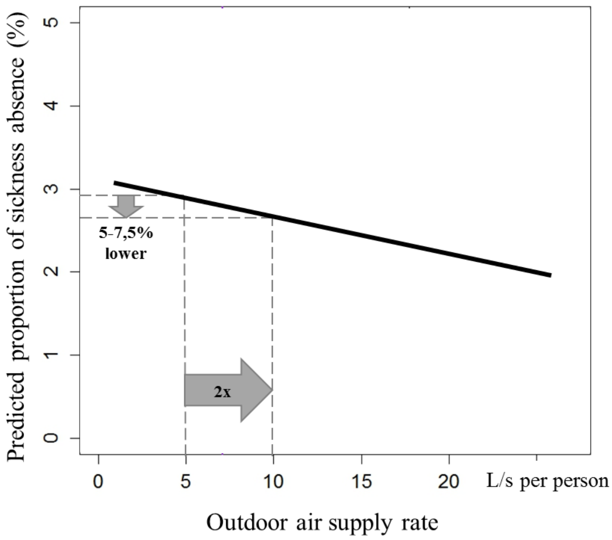


Figure 3: Absence rate as a function of carbon dioxide concentration (ventilation rate) in different school districts in California (Mendell et al., 2013)

3 ECONOMIC IMPLICATIONS

Wargocki et al. (2014) attempted to estimate future socio-economic consequences of improved indoor air quality in Danish primary schools. Assuming that increased school performance will improve productivity, will reduce the duration of primary education (in Danish system the pupil can take either 9 or 10 grades in the elementary education depending on the educational attainment) and absenteeism of teachers, the macro-economic effects were estimated of increasing ventilation rates from 6 L/s per person required by the Danish Building Code to 8.4 L/s per person required by the Swedish Code. Modelling of benefits showed that increasing ventilation would yield an average annual increase in GDP of €173 million due to increased productivity of the workforce and more pupils leaving the school earlier, and an average annual increase in the public budget of €37 million again through improved productivity and shorter stay in primary school as well as lower teacher sick leave. These effects correspond to no more than 0.07% of Danish GDP in 2011 thus seems reasonable in magnitude. All effects are expected to increase (being higher and higher from year to year over the 20 years for which the analyses were performed) the more students leave the schools where the air quality and ventilation are improved.

A different estimation of the effects of reduced absence rates was performed by Mendell et al. (2014). They assumed that ventilation rates in Californian K-12 schools will be increased from current level of 4 to 7.1 and 9.4 L/s per person, and estimated the benefits from decreased illness absence to school districts (i.e., increased revenue from the State for student attendance which is the model adopted by the schooling system there), and the benefits to families as a result of decreased costs from lost caregiver wages/time. These benefits yielded from US$33 to US$66 million from increased revenue and from US$80 to US$160 million from reduced losses to caregivers.

Both estimations show that the benefits and potential losses due to reduced learning ability because of poor air quality could be considerable and although not providing immediate tangible benefits (except the effects on absence rates) they should not be neglected.

4 POSSIBLE VENTILATION SOLUTIONS

Gao et al. (2014) examined how different types of ventilation systems influence classroom temperature and air quality, and whether there is any impact on window opening behaviour of pupils and teachers. The classrooms were selected in the same school located in rural Copenhagen with different methods of ventilation obtained as follows: by manually operable windows, by automatically operable windows with and without exhaust fan, and by balanced mechanical ventilation system. Measurements were performed for one month in the selected classrooms during non-heating season when outdoor temperatures were 10-30℃ (during school hours) and in the heating season when outdoor temperatures were when the outdoor temperatures ranging from -7.3 to 10.9℃. The results showed that classrooms with automatically operable windows and exhaust fan and with mechanical ventilation systems achieved the best thermal environment and air quality during both heating and non-heating season among all classrooms examined, while classroom with manually operable windows had the highest carbon dioxide concentration levels more so during heating season (Figure 4). These results may suggest that hybrid ventilation and mechanical ventilation system are the most feasible solutions for moderate climates typical for Denmark. These results should not be generalized for other climates such as e.g. cold or Tropical climates without providing sufficient research evidence.

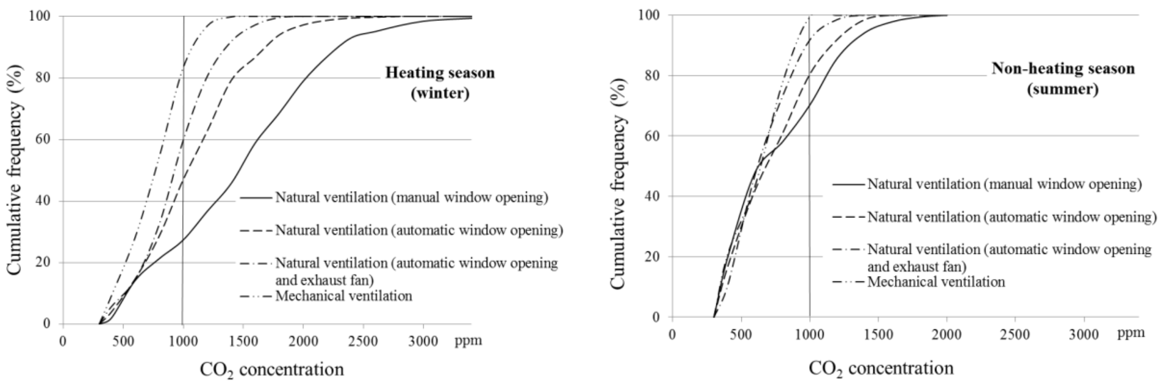


Figure 4: Cumulative curves showing measured carbon dioxide concentration during school hours in classrooms with different ventilation systems during heating and non-heating season (Gao et al., 2014)

Wargocki and da Silva (2015) examined whether providing pupils in elementary school with visual feedback on the classroom concentration of CO_2 and instructing them to open the windows in response to this indication would results in changes in window opening behaviour and in improved classroom ventilation. The study was performed in the heating and non-heating season, in the latter case with and without split air-conditioner securing classroom temperature at comfortable level. During a two-week periods in both seasons, teachers and students were instructed to open the windows in response to the CO_2 feedback in one week and open them, as they would normally do, without feedback, in the other week. Providing CO_2 feedback caused that more windows were opened in this condition, as expected and reduced CO_2 levels (Figure 5). This increased energy use for heating and reduced the cooling requirement. Split cooling reduced the frequency of window opening when no CO_2 feedback was present, suggesting that classroom temperature is the driving factor for this behavioural response. Despite many limitations, da Silva et al. recommended the use of CO_2 feedback as a feasible solution for controlling classroom air quality in rural schools with natural ventilation when ambient climate conditions are mild. The Authors cautioned that cooling of naturally ventilated classrooms may reduce window opening and therefore will result in poor classroom air quality.

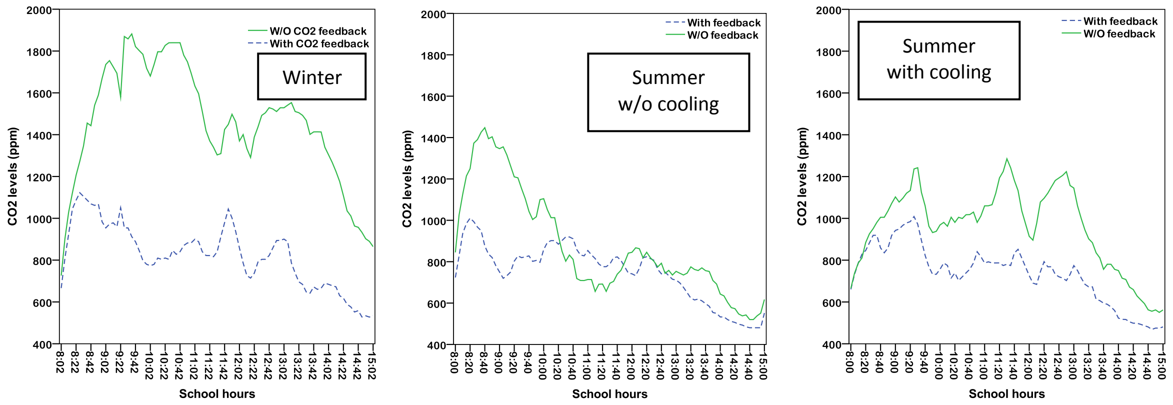


Figure 5: Carbon dioxide concentration in classrooms during school hours when pupils were able and not able to use visual feedback; measurements are shown for heating season and non-heating season, in the latter case for two instances when the temperature in the classroom was not controlled and could drift according to the temperature outdoors and when it was controlled with split cooling unit to avoid thermal discomfort (Wargocki and da Silva, 2015)

5 ACKNOWLEDGEMENTS

Present work was supported through grant IEE/131786/SI2.675580 awarded by the European Commission through Intelligent Energy Europe Programme for the project titled "Sustainable school building renovation promoting timber prefabrication, indoor environment quality and active use of renewables", RENEW SCHOOL.

6 REFERENCES

Angell, W.J., Daisey, J. (1997) "Building factors associated with school indoor air quality problems: A perspective" Proceedings of Healthy Buildings/IAQ'97, Washington DC, Vol. 1, 143-148. Virginia Polytechnic Institute and State University.

ASHRAE(2014) ASHRAE Standard 62.1-2014, Ventilation for acceptable Indoor Air Quality, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta

Bakó-Biró Z., Clements-Croome., D.J., Kochhar, N., Awbi, H.B., Williams, M.J. (2012) Ventilation rates in schools and pupils' performance. Build. Environ. 48, 215–223.

Berner, M. (1993) "Building conditions, parental involvement, and student achievement in the District of Columbia public school system", Urban Education, 28(1), 6-29.

Boxem, G., Joosten, L., Bruchem, M.v. and Zeiler, W. (2006) "Ventilation of Dutch schools; an integral approach to improve design", In: Schwarzer, J. and Lain, M. (eds) Proceedings of 17th Air-conditioning and Ventilation Conference 2006, Prague, Czech Republic, Society of Environmental Engineering, pp. 31-36.

Brelih N. (2012) Ventilation rates and IAQ in national regulations. REHVA J;1:24e8.

Dimitroulopoulou C. (2012) Ventilation in European dwellings: a review. Build Environ; 47:109e25.

Clausen, G., Toftum, J., Andersen, B. (2014) Indoor environment in classrooms – Results of the mass experiment 2014 (Indeklima i klasseværelser – resultater af Masseeksperiment 2014). Danish Science Factory, Copenhagen, 19 pages (in Danish).

Daisey, J., Angell, W.J. and Apte, M.G. (2003) “Indoor air quality, ventilation and health symptoms in schools: an analysis of existing information”, Indoor Air, 13, 53-64

Danish Housing and Building Agency (1995) Building Regulations, Danish Ministry of Housing, Copenhagen, Denmark.

Dijken, F.V., Bronswijk, J.V. and Sundell, J. (2005) “ Indoor environment in Dutch primary schools and health of the pupils”, In Yang, X., Zhao, B. and Zhao, R. (eds) Proceedings of Indoor Air 2005, Beijing, China, Tsinghua University Press, Vol. I(1), pp. 623-627.

EFA, (2001) Indoor Air Quality in Schools. European Federation of Asthma and Allergy Associations, Helsinki, Finland.

Ervasti, J., Kivimäki, M., Kawachi, I., Subramanian, S., Pentti, J., Oksanen, T., Puusniekka R., Pohjonen ,T., Vahtera, J., Virtanen, M., (2012) School environment as predictor of teacher sick leave: data-linked prospective cohort study. BMC Public Health 12, 770 (http://biomedcentral.com/1471-2458/12/770 accessed 30 Oct 2014)

Gaihre, S., Semple, S., Miller, J., Fielding, S., Turner, S. (2014) Classroom carbon dioxide concentration, school attendance, and educational attainment. J. School Health, 84(9), 569-574.

Gao, J., Wargocki, P., Wang, Y., (2014) Ventilation system type, classroom environmental quality and pupils' perceptions and symptoms. Build. Environ., 75, 46-57.

Haverinen-Shaughnessy, U., Moschandreas, D.J., Shaughnessy, R.J. (2011) Association between substandard classroom ventilation rates and students’ academic achievement. Indoor Air, 21(2), 121–131.

Kolarik, B., Andersen, Z. J., Ibfelt, T., Engelund, E. H., Møller, E., & Bräuner, E. V. (2015). Ventilation in day care centers and sick leave among nursery children. Indoor air.

Mattsson, M. and Hygge, S. (2005) “Effect of articulate air cleaning on perceived health and cognitive performance in school children during pollen season”, In Yang, X., Zhao, B. and Zhao, R. (eds) Proceedings of Indoor Air 2005, Beijing, China, Tsinghua University Press, Vol. I(2), 1111-1115

Mendell, M.J. and Heath, G.A. (2005) “Do indoor pollutants and thermal conditions in schools influence student performance? A critical review of the literature”, Indoor Air, 15, 27-52.

Mendell, M.J., Eliseeva, E.A., Davies, M.M., Spears, M., Lobscheid, A., Fisk, W.J., Apte, M.G. (2013) Association of Classroom Ventilation with Reduced Illness Absence: A Prospective Study in California Elementary Schools. Indoor Air, 23(6), 515-528 (doi:10.1111/ina.12042).

Menå HM and Larsen EM (2010). Indoor environment in schools. MSc thesis, Technical University of Denmark.

Myhrvold, A.N., Olsen, E. and Lauridsen, Ø. (1996) “Indoor environment in schools – Pupils’ health and performance in regard to CO2 concentration”. In: Yoshizawa, S., Kimura, K., Ikeda, K., Tanabe, S. and Iwata, T. (eds) Proceedings of Indoor Air '96, Nagoya, The 7th International Conference on Indoor Air Quality and Climate, Vol. 4, pp. 369-374.

Petersen, S. Jensen, K.L, Lønbæk Sandvang Pedersen, A., and Smedegaard Rasmussen, H. (2015) The effect of increased classroom ventilation rate indicated by reduced CO2-concentration on the performance of schoolwork by children, Indoor Air.

Pilotto, L.S., Douglas, R.M., Attewell, R.G. and Wilson S.R. (1997) “Respiratory effects associated with indoor nitrogen dioxide exposure in children”, International Journal of Epidemiology, 26(4), 788-796.

Ribic, W. (2008) Nachweis des Zusammenhanges zwischen Leistungsfähigkeit und Luftqualität. Heizung, Lüftung/Klima, Haustechnik, 59(7), 43-46 (in German).

Santamouris, M., Synnefa, A., Asssimakopoulos, M., Livada, I., Pavlou, K., Papaglastra, M., Gaitani, N. Kolokotsa, D, and Assimakopoulos, V. (2008). Experimental investigation of the air flow and indoor carbon dioxide concentration in classrooms with intermittent natural ventilation. Energy and Buildings, 40(10), 1833-1843.

Shendell, D.G., Prill, R., Fisk, W.J., Apte, M.G., Blake, D., Faulkner, D. (2004) Associations between classroom CO2 concentrations and student attendance in Washington and Idaho. Indoor air, 14(5), 333–341.

Simoni, M., Annesi-Maesano, I., Sigsgaard, T., Norbäck, D., Wieslander, G., Nystad, W., Canciani, M., Sestini, P., Viegi, G. (2010) School air quality related to dry cough, rhinitis and nasal patency in children. Eur. Respir. J., 35, 742-749. Smedje, G. and Norbäck, D. (2000) “New ventilation systems at select schools in Sweden--effects on asthma and exposure”, Archives of Environmental Health, 55: 18-25.

Simons, E., Hwang, S.A., Fitzgerald, E.F., Kielb, C., Lin, S. (2010) The impact of school building conditions on student absenteeism in Upstate New York. Am. J. Public Health, 100, 1679–1686.

Sowa, J. (2002) “Air quality and ventilation rates in schools in Poland – requirements, reality and possible improvements”, In: Proceedings of Indoor Air 2002, Monterey, CA, USA, 2, 68-73.

Toftum, J., Kjeldsen, B.U., Wargocki, P., Menå, H.R., Hansen, E.M.N., Clausen, G. (2015) Association between Classroom Ventilation Mode and Learning Outcome in Danish Schools, Building and Environment.

Wargocki, P., Wyon, D. (2013) Providing better thermal and air quality conditions in school classrooms would be cost-effective. Build. Environ., 59, 581-589.

Wargocki, P., Foldbjerg, P., Eriksen, K.E., Videbæk, L.E. (2014) Socio-economic consequences of improved indoor air quality in Danish primary schools. In: Proceedings of Indoor Air 2014, paper no. HP0946.

Wargocki, P., & Da Silva, N. A. F. (2015). Use of visual CO2 feedback as a retrofit solution for improving classroom air quality. Indoor air, 25(1), 105-114.

Wyon, D.P. , Wargocki, P., Toftum, J. and Clausen, G. (2010) Classroom ventilation must be improved for better health and learning, REHVA Journal, 6, 35-39.

EVALUATION OF VENTILATION SOLUTIONS FOR RETROFITTING OF SCHOOLS

Christian A. Hviid*[1] and Steffen Petersen[2]

1 Department of Civil Engineering
Technical University of Denmark
Brovej building 118
DK-2800 Kgs. Lyngby
**Corresponding author: cah@byg.dtu.dk*

2 Department of Engineering
Aarhus University
Inge Lehmanns Vej 10
DK-8000 Aarhus C, Denmark

ABSTRACT

In 2011, the Danish Energy Agency initiated a study into ventilation solutions for the retrofit of schools to identify the most promising technologies. The reason was an increasing awareness that the ability of school children to absorb, adapt and use knowledge was affected negatively by inadequate ventilation rates. This paper presents an output of this study. A method for evaluation of the ventilation systems is proposed. The method consists of three categories with a clear separation to create a scoring board that facilitates transparent and unbiased evaluation. The method was applied in an analysis of ventilation solutions in schools and the finding was that centralized or decentralized balanced mechanical ventilation activated by CO2 and temperature sensors in the individual classrooms has the highest score and has the best market maturity. Other solutions are possible but unacceptable performance on some performance issues thus has to be accepted. Furthermore, a set of design rules for ventilation solutions in the case of school retrofit is presented. The design rules require an analysis of the building typology before the best solution can be chosen for a specific school.

KEYWORDS

Ventilation, indoor air quality, schools, retrofitting

1 INTRODUCTION

Children spend more time in school than in any other environment besides their home. It is therefore thought-provoking that several investigations have shown that the ventilation rate in schools across the world is below recommended levels (Corsi et al., 2002; Daisey et al., 2003; Sohn 2009; Andersen et al., 2009) and that there is solid scientific evidence that this affects the ability of school children to absorb, adapt and use knowledge (Wargocki, 2015).
As a consequence of this, the Danish Energy Agency initiated a comprehensive study (Hviid et al., 2014) to identify well-functioning ventilation solutions for retrofit of schools. A similar German study was published by FGK (2004). The present study included a survey of existing knowledge, analysis and qualification of different solutions and specific recommendation of the solutions that best meet a number of criteria, such as indoor climate, energy consumption, costs and aesthetics. This paper presents the method for evaluation of the ventilation systems

used in this study and the main findings from a study where the method was applied for analysis of the ventilation solutions in schools.

2 METHOD FOR EVALUATION OF VENTILATION SYSTEMS

The primary motivation for establishing ventilation in schools is to ensure a healthy and productive learning environment by providing good air quality. The secondary objective of ventilation is to provide a mean to minimise overheating. However, there are other important issues to consider in relation the performance of a ventilation solution. The section provides a method for performance evaluation of ventilation systems.

The key performance issues of a ventilation solution are listed in the first column in Table 1. Each key performance issue has an assessable performance criterion. These performance criteria are divided into three categories with a clear separation to create a scoring board that facilitates transparent and undisputable evaluation. A newly installed ventilation system in a school is in general expected to be within category in performance issues. If the ventilation solution demonstrates *significantly* better performance than legal requirements, current standards and practice – that is for example the case if the ventilation system meets the future energy requirements of the legislation – then the category becomes "excellent". The score becomes "Unacceptable" if a performance issue is outside the range of the category "Acceptable".

There is not suggested any weighing of the performance issues in this method because it is a subjective task. The authors recommend that all evaluation parameters are considered to be equal because they all are highly relevant to any well-functioning ventilation system in a classroom. However, the user of the method is free to assign any greater weight to an issue if desired.

Table 1: Three levels of evaluation criteria ranging from unacceptable to excellent

Performance issues	Unacceptable	Acceptable	Excellent
Indoor air quality, CO_2	Above 1500 ppm	1000-1500 ppm	Below 1000 ppm
Temperature	Regularly outside 20-26 °C	Within 20-26 °C	20-23 °C
Cooling by night ventilation	-	Night ventilation is a possibility	Night ventilation by default
Draught risk	Occurring	Low risk	Not occurring
Noise, mechanical (traffic)	Above 30 (33) dB(A)	27-30 (30-33) dB(A)	Below 27 (30) dB(A)
Aesthetics	-	(Very) visible	Integrated or invisible
Filtration	-	Filtration is a possibility	Filter class F7
Specific fan power, SFP	Above 2100 W/(m³/s)	1000-2100 W/(m³/s)	Below 1000 W/(m³/s)
Heat recovery	Below 70 %	70-85 %	Above 85 %
Installation costs	-	On par with central mech. vent.	30 % cheaper
Maintenance costs	-	On par with central mech. vent.	30 % cheaper

3 ANALYSIS OF VENTILATION SOLUTION IN SCHOOLS

The method presented in section 2 was used for analysis of the ventilation solutions for schools. The analysis was carried out on the basis of data from CO_2 and temperature logs of 85 different schools, and on data from manufacturer's data sheets. The CO_2 concentration was logged in selected classrooms equipped with different types of air exchange principles:

- 31 with balanced central mechanical ventilation

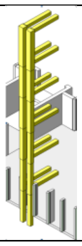
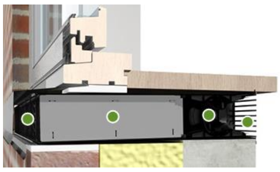
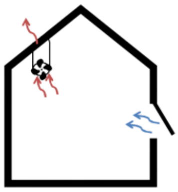
- 11 with mechanical exhaust
- 42 with manual airing
- 1 with automatic airing in a combined stack and cross ventilation configuration

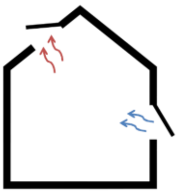
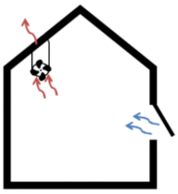
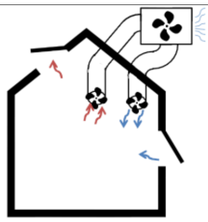
The actual air exchange (ventilation rate) was approximated from the CO_2 peak concentration and the number of students in the class.

The analyses of draught risk, aesthetics and costs were based on qualitative data collected by interviewing a number of practioneers, e.g. contractors, manufacturers, engineers, and service personnel in the European school ventilation industry. A full list of the interviewees is available in Hviid et al. (2014). The assessment of aesthetic performance was based on discussions with an architect with substantial experience in school refurbishments.

3.1 Ventilation principles

Various types of ventilation principles have been identified, examined and assessed, see the tables below. These systems cover what is considered by the authors to be representative for up to 95% of the types of ventilation solutions that can be found in schools today.

Central ventilation	**Decentral ventilation**	**Micro ventilation**	**Mechanical exhaust**	**Exhaust w/ pre-heat by air-to-water heat pump**
				
Dual ducting	Compact unit in room	Micro units in facade	Central fan and intake in facade	Heating coil in facade intake

Airing	**Automatic airing**	**Fan-assisted automatic airing**	**Hybrid ventilation**
			
Manual opening of windows	Automatic windows in cross flow configuration	Auto. window opening with fan-assistance	Airing and central mechanical

4 RESULTS

The results presented constitute only a short exempt of the comprehensive study presented by Hviid et al. (2014).

Figure 2 depicts the ventilation rate in 78 of the logged 85 schools sorted by the ventilation installation or renovation year. The figure illustrates the tendency that schools with manual airing or simple mechanical exhaust, even in newer schools, are not able to meet the current target value in the Danish Building Code. For comparison, the legal requirement in Belgium is on par with Denmark, whereas Sweden, Norway and Holland require approx. 40% more fresh air per pupil. Only newer central mechanical systems perform satisfactorily today and seem able to meet the expected future demands.

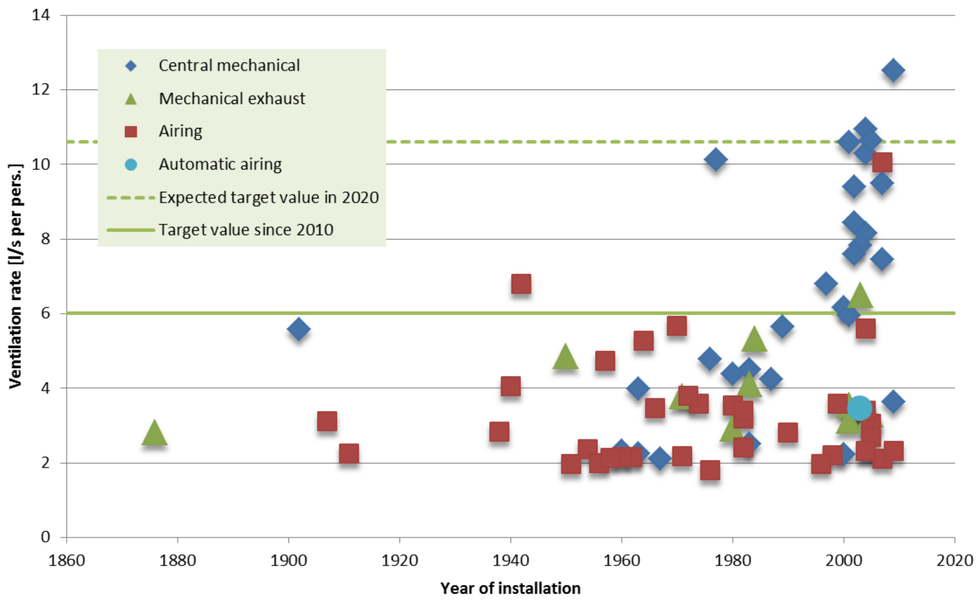


Figure 2: Ventilation rates from classrooms in 78 different schools. The target values are approximately identical to the current and the future requirement in the Danish Building Code

The costs of installing and maintaining different ventilation systems over a period of 20 years are shown in Figure 3. Airing has no costs because it only requires the openable windows to be serviced. The cost for this is considered to be part of the general building maintenance. Installing systems for automatic window opening only requires a rather small extra installation and service costs whereas combining automatic airing with mechanical ventilation in a hybrid solution is quite expensive. The maintenance costs for micro ventilation reflects the expected lifespan of the mini fans which by build quality and sheer number poses a significant service risk.

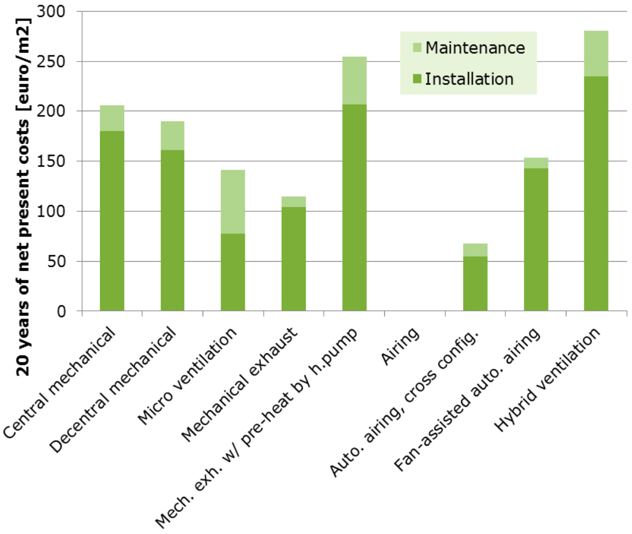


Figure 3: The costs of installing and maintaining different ventilation systems over a period of 20 years in a classroom. Discount rate 2.0%

It has been difficult to measure the exact energy consumption of the logged ventilation systems and the information sources were error prone. Instead we presumed that future installations are more energy efficient than the old ones due to continuous product development. Figure 4 shows the yearly heating and electricity consumption per m^2 in a classroom on the basis of the expected efficiencies of new installations listed in Table 2. The simulation was carried out in a simple building simulation tool based on the quasi-static method in ISO 13790 (2008). Cooling down the building thermal mass by night ventilation was crucial to keep the temperature within comfortable range and it is in general recommended to use night ventilation whenever possible. Furthermore, figure 4 shows that heat recovery should be of major concern in a new ventilation installation and that care should be taken not to let the introduced extra electricity consumption exceed the heatingsavings.

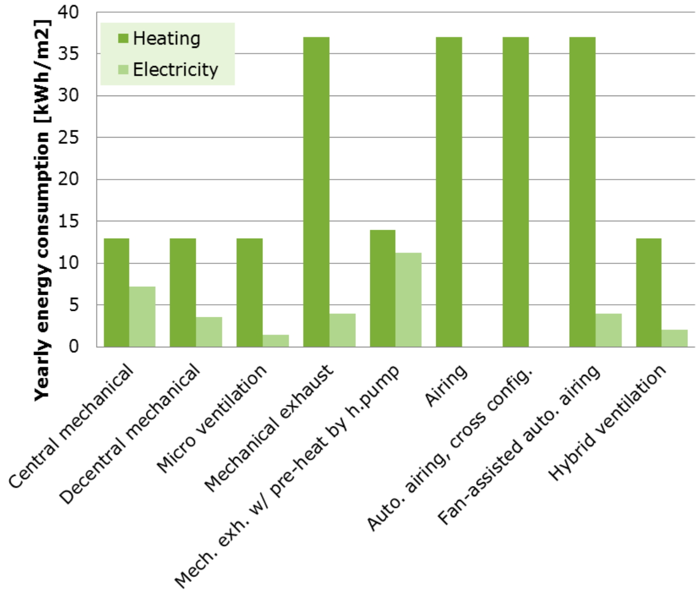


Figure 4: The yearly energy consumption for different ventilation principles in a classroom

Table 2: The energy-related ventilation properties used to simulate the total energy consumption in a classroom in a Danish climate. Specific fan power

	SFP W/(m^3/s)	Heat recovery	Night ventilation
Central mechanical	1.500	85 %	Yes
Decentral mechanical	700	82 %	Yes
Micro ventilation	300	85 %	Yes
Mechanical exhaust	800	0 %	Yes
Mech. exch. w/ pre-heat by h.pump	2.300	80 %	Yes
Airing	0	0 %	No
Automatic airing	0	0 %	Yes
Fan-assisted automatic airing	800	0 %	Yes
Hybrid, two-mode	1000	80 %	Yes

Table 3 depicts the scoring chart of different ventilation systems based on the logs in 85 different schools, interviews with practioneers at all levels in the European school ventilation industry, energy simulation results based on the expected or required performance of future installations and other national and international sources.

Most noticeable are the draught risks that were reported for some of the systems. The common denominator of these systems is that they supply air directly into the comfort zone and even when the air was pre-heated in the façade, significant problems with draught risks were still reported. Some of the systems do not filter the air (marked with a white cell and a "d"), and this could pose a problem in areas with a lot of pollen allergies or in polluted urban areas.

Table 3: The scoring chart of different school ventilation systems. Dark indicate excellent performance, light is acceptable (legal), and stripes indicate illegal or troubling performance. White cells mean either that the variable has no meaning in this specific context, or that is does not constitute a violation of the legislation

a) evidence suggests that simple mech. Exhaust performs usually ok to keep the co2 level acceptable.
b) without special measures the draught risk is eminent. The risk should be evaluated carefully in each application with detailed simulations
c) acceptable performance requires silencer devices in the facade (hviid et al., 2014), if the facade openings are oriented towards traffic. School yards my also require similar sound dampening devices to be installed.
d) for ventilation principles with facade openings filtration is not a legal requirement.
e) hybrid two-mode uses facade openings in the summer but for a large part of the year, the intake air is filtrated through the mechanical ventilation system. Consequently the performance is acceptable.
f) heat recovery is not a legal requirement in this type of ventilation system

5 DESIGN RULES

The evaluation method described in section 2 covers the aspects of ventilation in schools However; in the case of school retrofit the building's design typology is crucial to the appropriate choice of ventilation system. Especially the constructive principle is often decisive when the optimal ventilation solution is to be recommended as changes that compromise the structural system are expensive. For example, it is expensive (maybe even impossible) to install a central balanced ventilation system in a building with heavy load-bearing interior walls due to the need of ducts penetrating the existing bearing structure. Table 4 provides a design for choosing the most appropriate ventilation solution for different building typologies. For further details see Hviid & Petersen (2012).

Table 4: Recommended design rules for retrofitting of schools with ventilation

Principle	Recommended design rules
Central mechanical	Best suited for buildings with mainly light partitions that are cheap to penetrate with ducts or one-storey buildings where ducts can be routed on the roof. If the systems are visible, aesthetics should be considered. Height in corridor > 3.2 m Room height for diffusers in the ceiling > 2.5 m: Room height for inlets in the wall > 2.8 m, if height from inlet to ceiling is < 0.3 m
Decentral mechanical	Best suited for schools with many load-bearing walls that blocks easy routing of ducts Does take up floor space or space in the façade which will reduce the available daylight. Internal and external aesthetics must be considered. Room height > 2.8 m Room depth > 5x room height To ensure the ventilation effectiveness and minimize the risk of draught, the inlet should be placed close to the ceiling. Distance from inlet to ceiling < 0.3 m Avoid objects in the jet path, e.g. lighting fixtures or beams A mockup is recommended for assessing noise issues
Micro ventilation	Best suited for schools with many load-bearing walls that blocks easy routing of ducts Should be placed as high as possible above the comfort zone to avoid draught. Distance from inlet to ceiling < 0.3 m
Mechanical exhaust	The largest potential is in schools with existing vertical shafts but the air distribution is critical and the intake through openings in the façade creates significant draught risks. The analyses shows inadequate performance Room depth < 5x room height
Mechanical exhaust with pre-heating by air-to-water heat pump	Intake through openings in the façade creates significant draught risks, even if the intake air is pre-heated. Façade coils suffers from freezing if the heating and ventilation system controls are not aligned
Airing	Should be avoided or at least supplemented with nudging features, e.g. a CO2 traffic light that compels the user to open the window (Wargocki and Da Silva, 2015).

	Room height > 2.7 m Room depth < 2x room height Not possible in facades towards traffic
Automatic airing, cross configuration	Requires special room designs, e.g. large room volumes and ventilation between facade-facade or facade-skylight. Detailed simulations of the performance is recommended to assess the indoor air quality and to avoid the risk of draught Room height > 3.2 m Room depth < 5x room height Not possible in facades towards traffic
Fan-assisted automatic airing	Room height > 2.7 m
Hybrid, two-mode	The design rules for airing and central mechanical apply.

6 CONCLUSIONS

This paper presents a score chart for transparent and unbiased evaluation of different school ventilation solutions. Nine different ventilation principles has been evaluated and compared on the basis of quantitative and qualitative analyses. The main conclusion is that centralized or decentralized balanced mechanical ventilation activated by CO_2 and temperature sensors in the individual classrooms has the highest score in the method and has the best market maturity. Automatically controlled hybrid ventilation, i.e. automatically controlled airing and mechanical ventilation in combination, is performing excellent on many performance issues except costs. The cheapest solution is airing in combination with some nudging feature that persuades the user to open the windows. Another promising principle with very low installation costs and low energy is micro ventilation. However, it suffers from lack of filtration and from uncertainties concerning the exact service costs and lifespan.

A set of design rules for ventilation solutions in the case of school retrofit is presented. The design rules require an analysis of the building typology, e.g. structural principle, room height, room depth, possible location of ventilation inlet and routing, before the best solution can be chosen for a specific school.

7 ACKNOWLEDGEMENTS

The present work was supported through grant IEE/131786/SI2.675580 awarded by the European Commission through Intelligent Energy Europe Programme for the project titled "Sustainable school building renovation promoting timber prefabrication, indoor environment quality and active use of renewables", RENEW SCHOOL.

8 REFERENCES

Andersen, B., Clausen, G., Larsen, E.M., Menå, H.R. (2009). Masseeksperiment 09. Indeklima i klasselokaler. Available from: http://masseeksperimentet.danishsciencefactory.dk/sites/default/files/files/fagligrapportmassex2009.pdf

Corsi, R.L., Torres, V.M., Sanders, M. and Kinney, K.L. (2002). Carbon dioxide levels and dynamics in elementary schools: results of the TESIAS Study. In: Proceedings of Indoor Air ’02, Vol. 1, pp. 74–79.

Daisey, J.M., Angell, W.J. and Apte M.G. (2003). Indoor air quality, ventilation and health symptoms in schools: an analysis of existing information, Indoor Air, 13, 53–64.

FGK, Fachverband Gebäude-klima e.V. (2004) Lüftung von Schulen, Status-report 22 Arbeitsgruppe Raumklimageräte und Behaglichkeit, Bietigheim-Bissingen, Germany

Hviid, C.A., Petersen, S. (2012). Implementation of ventilation in existing schools – a design criteria list towards passive schools, in: Proceedings of Passive House Symposium, Bruxelles, October 5

Hviid, C.A., Wargocki, P., Mortensen, J.D., Lindgreen, T.S., Holst, M.K. (2014). Velfungerende løsninger til ventilationssystemer i skoleklasser, Danish Energy Agency, Copenhagen. Available from: http://pcb-guiden.dk/file/460142/ventilationssystemer_skoleklasser_katalog.pdf

ISO 13790. 2008. Energy performance of buildings – calculation of energy use for space heating and cooling. International Organization for Standardization. Geneva Switzerland

Sohn J., Yang W., Kim J., Son B., and Park, J. (2009). Indoor air quality investigation according to age of the school buildings in Korea, J Environ Manage, 90, 348–354

Wargocki, P. (2015). Ventilation, indoor air quality and learning in schools. In: Proceedings of the 36th AIVC conference, Madrid, Sept. 23-24

Wargocki, P., & Da Silva, N. A. F. (2015). Use of visual CO2 feedback as a retrofit solution for improving classroom air quality. Indoor air, 25(1), 105-114

MINIMAL INVASIVE VENTILATION SYSTEMS WITH HEAT RECOVERY FOR SCHOOL BUILDINGS

Rainer Pfluger*[1], Kai Längle[1], Gabriel Rojas[1]

1 Unit for Energy Efficient Builidngs
University of Innsbruck
Technikerstrasse 13
Innsbruck, Austria

2 Organisation
Street address
Address, Country
**Corresponding author: e-mail address*

ABSTRACT

This paper presents different ventilation solutions for the retrofit in existing school building with a special focus on historic buildings. These building usually pose a major challenge for the integration of energy efficient ventilation systems, i.e. with heat recovery. For decentralized systems, the ductwork can be minimized by wall integrated heat recovery units, whereas for central systems, a horizontal air distribution in the attic combined with vertical ducts was found to be a possible solution for listed buildings. Former is problematic for aesthetic reasons due to visibility of the outer wall inlets, especially when no exterior insulation layer can be applied. A central unit with vertical ducts has the drawback of requiring many fire protection flaps, which are associated with high maintenance costs. Therefore a new ventilation system was designed and tested for a listed school building in Innsbruck (Austria), which is one of eight case study buildings within the EU-project 3ENCULT. In order to minimize the ductwork within the building, an active overflow system takes the air from the corridor to the class room and vents the extract air back to it. A central heat recovery system ventilates the staircase and the corridors with preheated fresh air. The prototype of the active overflow system, the control strategy and the simulation results are presented. To show the potential variants when implementing the active overflow principle, a further school building which will be energetically refurbished within the EU-project SINFONIA is presented. Here the HVAC-planning foresees the implementation of the active overflow concept with and without dedicated extract air ducting.

KEYWORDS

Active overflow ventilation, school ventilation, heritage protected, minimal invasive

1 INTRODUCTION

There is an overwhelming amount of literature showing the negative impact of poor classroom ventilation on student performance, e.g. (Toftum et al., 2015). Therefore it is important to equip all schools with a mechanical ventilation system. This way good indoor air quality can be ensured independently of the pupils and teachers manual ventilation habits. Many school buildings are now being energetically refurbished to lower the emissions of CO_2-emissions. Only a ventilation system with heat recovery can fulfil energy efficiency and good indoor air quality. It can also help to protect the building construction from a building physics point of view and it enhances the thermal comfort for the users. For heritage protected buildings the intervention should be as reversible and minimal invasive as possible.

In case of standard ventilation systems for new buildings, ductwork is applied to guide the air to the occupied space. These ducts are installed in vertical shafts or behind the suspended ceiling. The construction of new shafts and suspended ceilings is not possible in most cases for historic buildings. This paper shows several solutions how to avoid ducts, shafts and suspended ceilings as far as possible. Much of the content is based on a previous publication (Pfluger & Längle, 2013). This work was performed within the EU-project 3ENCULT, where a wide range of energy efficient solutions for cultural heritage are investigated and developed. The following strategies for implementation of heat recovery were found to be especially adapted for historic buildings:

- decentralized wall integrated systems
- central system with direct vertical supply air ducts
- central system with active overflow principle

The pros and cons of central and decentralized systems are described hereunder. The active overflow system was implemented and tested in two class rooms in the 3ENCULT case study CS5 (school building in Hötting, Innsbruck, Austria) and will be fully implemented in another listed school building in Innsbruck.

2 DECENTRALIZED VERSUS CENTRALIZED SYSTEMS

The most important decision in terms of the choice of the ventilation system is, whether a central unit or decentralized systems is the most appropriate solution for the individual building.

The pros and cons of central systems can be summarized as follows:

The central heat exchanger needs a plant room (or space in the attic). If the ambient air intake and the exhaust air outlet is placed at the roof or conducted underground, no openings in the façade are necessary. This is very important for architectural reasons. However, the planning and installation costs are relatively high. Especially in historic buildings, individual solutions have to be found in any case – the system is tailor-made for the building, respecting the architectural and historic value of the monument. The horizontal as well as the vertical ducts need space in the building. Sometimes unused chimneys can be used for the vertical ducts, but also horizontal distribution is necessary. In this paper solutions are shown, how to avoid or reduce the ductwork. If any duct passes through a fire zone boundary, additional fire-protection appliances are necessary.

In case of decentralized systems almost no duct work and no fire protection equipment are necessary; however, the following disadvantages have to be kept in mind:

Ambient air intake and exhaust air outlet has to be conducted through the façade. In case of listed buildings, the aspect of the historic façade may not be changed. Solutions have to be found, which are acceptable respecting the preservation issues. In case of a system placed directly in the occupied zones of the building, the sound protection is difficult. The same holds for the maintenance and the aesthetics of the decentralized units in the occupied zones.

There is no general recommendation for the type of the system, neither in terms of preservation issues nor in terms of costs. The following sections may help to find the optimum solution for the individual building.

2.1 Decentralized Wall Integrated Ventilation Systems

For school building various products for decentralized ventilation systems are available on the market. Most of them are constructed for integration at the ceiling. In order to avoid long cold ducts, the unit should be placed close to the window. On the other hand, from a design point of view, this position has a lot of drawbacks. It is not good for an efficient use of daylight and it significantly affects the appearance of the room. These disadvantages can be avoided by placing the ventilation unit at the wall under the window (at the parapet). A design study for this version of a wall integrated ventilation system was performed within the EU-project 3ENCULT.

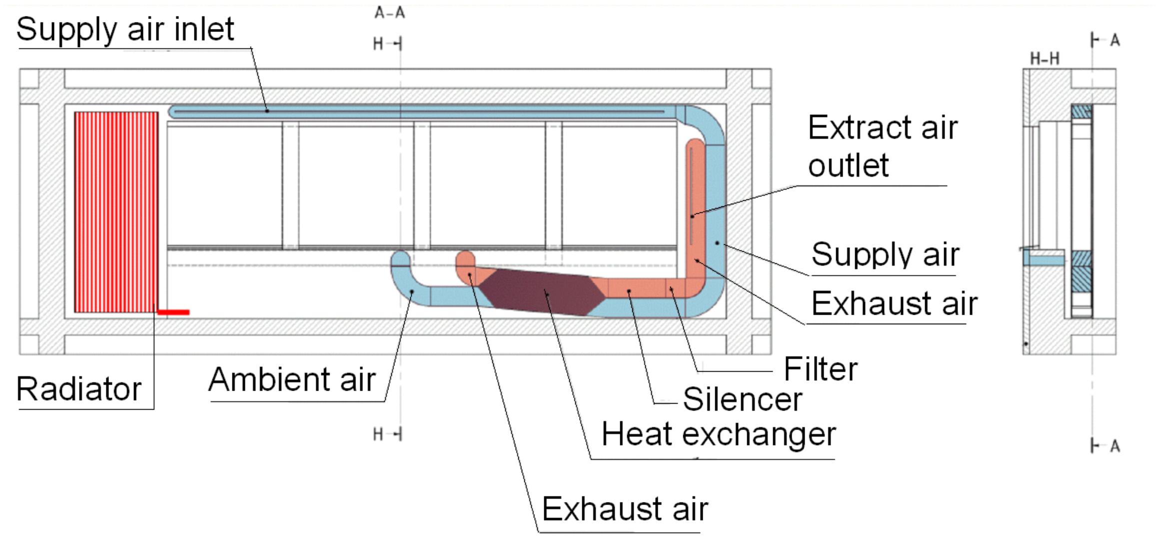


Figure 1: View from inside: Counter flow heat exchanger mounted at the parapet; supply air inlet above the window; extract air outlet besides the window.

As shown in Figure 1, a flat counter flow heat exchanger can be mounted at the parapet, whereas the supply air inlet and the exhaust air outlet are placed above and besides the window respectively. In order to avoid any grill at the façade for ambient air intake, a slit below the window sill can be applied. For exhaust air outlet, this possibility is not valid, because condensation and freezing problems at the wall surface would occur. Therefore a perforated plate or a cover plate in front of the window post is suggested as exhaust air outlet (see Figure 2).

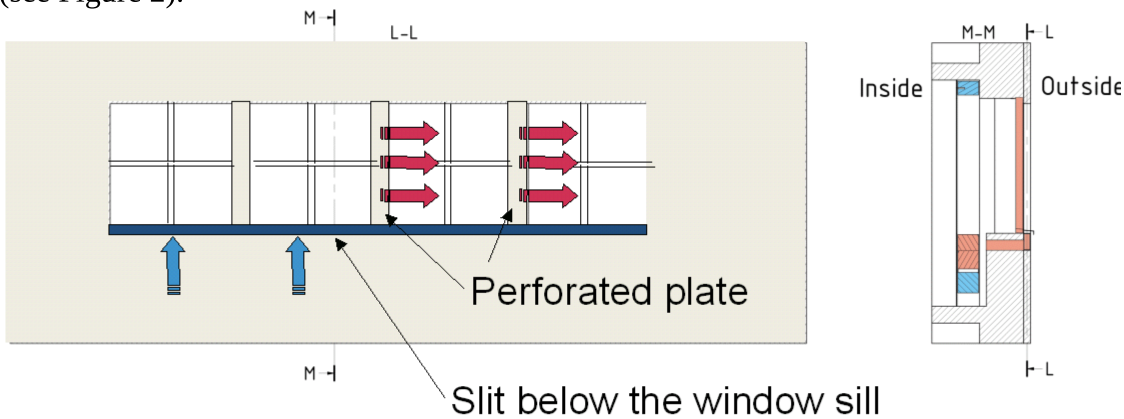


Figure 2: View from outside: Ambient air intake via slit below the window sill, exhaust air outlet via perforated plate at the window post.

The suggested design can be realized for buildings where an external insulation is applicable. In this case, the flat air ducts for ambient and exhaust air can be integrated in the insulation layer. After finishing the plaster (outside) and dry walling (inside), no ductwork is visible from both sides. For most of the listed buildings however, no external insulation is appropriate. In those cases a central heat recovery should be preferred.

2.2 Central Ventilation with Direct Vertical Ducts

In case of a central system for new buildings or refurbished buildings without special historic or architectural value, the most convenient way to mount the supply air ducts is to fix them at the ceiling in the corridor and to cover it by a suspended ceiling. For architectural reasons or aspects of preserving, this might not be possible in some listed buildings. In that case, a vertical supply air duct can be placed directly in the supply air rooms, whereas the horizontal distribution duct is mounted in the attic or cellar.
The disadvantage of this solution is, that the vertical ducts are crossing the ceilings, which are working as separation between different fire protection areas. For this reason, at each ceiling a fire flap has to be placed, resulting in high costs for investment, operation (pressure drop) and maintenance. The solution described in the next sections works without the supply air distribution duct.

3 ACTIVE OVERFLOW PRINCIPLE (AOP)

The AOP was developed and tested for the application in residential buildings by "Hochbaudepartement, Amt für Hochbauten, Stadt Zürich". A design competition for active overflow ventilation systems was launched and published in (Sprecher & Estevez, 2011). Within this document the active overflow principle is described as follows:
"Apartments with mostly open doors can be vented with a very simple ventilation system: It must only exhaust washrooms and kitchen, the whole supply air flow may be introduced at any place into the flat, e.g. into the corridor. A structural realization is simple because bathrooms and kitchen are often located close together and are accessible by riser shafts. In reality room doors remain mostly closed at night. To maintain acceptable air quality in these rooms, an active overflow element must ensure that the air from the corridor is vented into the rooms and back into the corridor - in compliance with all comfort criteria. The return flow of the air into the corridor can be realized via the door gap or via an overflow element (passive or active)" (translation by author). The applicability of the AOP in residential settings was also investigated in (Sibille & Pfluger, 2015).
As the AOP works successfully in refurbishing of residential buildings, the author decided to investigate, if the principle is also applicable for school buildings. The major difference compared to residential buildings is the higher flowrate, which is more difficult to distribute without draft risk and low sound emission. Airborne sound transmission from the class room to the corridor and vice versa can be minimized as described in the next section.

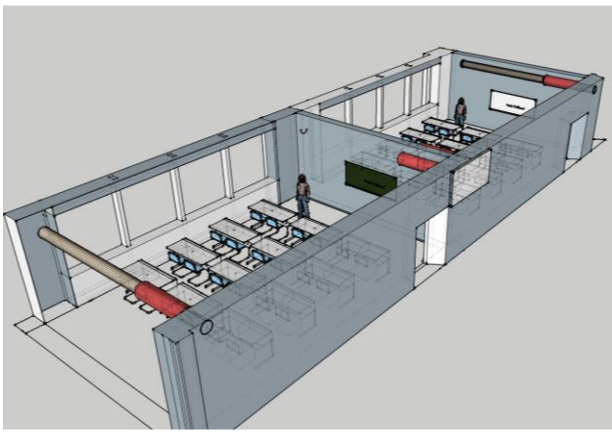


Figure 3: Active overflow system and air distribution from corridor to class room via textile hoses.

3.1 Control Strategies for Central Fans and Active Overflow Fans

As described above, the active overflow fans are responsible for the air change from the corridor to the classrooms whereas the central fans are venting fresh and preheated (by heat recovery) air to the staircase. The electricity consumption depends on the operation time and flow rate of each of the fans. As the occupation time is only a small fraction of the total time, an occupation dependent control strategy is necessary to save operational costs.

The most simple control strategy is to control the fans (both, the active overflow as well as the central fans) depending on a fix time schedule. The advantage is the low installation costs, because no sensor is necessary. The disadvantage is that this system is not flexible in terms of changes related to the real occupation and the time schedules.

If the CO_2-concentration is measured in the corridors or in the staircase, the central fans can be controlled via a Proportional-Integral (PI) controller to a set point of e.g. 600 ppm in order to keep high air quality in the staircase and corridor zone for ventilation of the class rooms. The concentration in the corridors will vary according to the occupation of the adjacent class rooms. Hence at least one CO_2-sensor per corridor should be installed; the maximum value measured by all of the sensors compared to the set point (error signal) is used as input signal for the PI-controller.

In general, the start time for operation of the fans should be at least one hour before pupils enter the school. This guarantees a good indoor air quality already at the beginning of the occupation time. Otherwise the accumulation of contaminants throughout the nighttime would result in low air quality within the first hour of occupation in the morning.

Keeping this in mind, a switch-on signal for all of the fans (both, active overflow and central fans) for one hour (e.g. from 6:45 to 7:45 a.m. at each working day) by time schedule is necessary in any case. As the air quality rating from emissions which are independent of occupation cannot be detected by CO_2-measurements, the flow rate of the central fans should be controlled additionally by TVOC-concentration measurement or simply by time schedule. For simplicity and robustness, the latter option is preferred.

In order to control more flexible in respect to changing occupation, the on/off signal for the active overflow fans could come from presence-control sensors in each room, which is considered a rather robust and low cost solution. However, even for this control strategy the pre-ventilation before occupation has to be controlled by time schedule.

To prevent bad odor within the time after the occupation, a time delay of one hour after the switch-off signal for the active overflow fan helps to bring down the contamination concentration.

In case of fire, any signal from a sensor for smoke or fire will switch off all fans, the central fan as well as all of the active overflow fans in order to avoid any active smoke distribution.

The control scheme as summarized in this section is displayed in Figure 4.

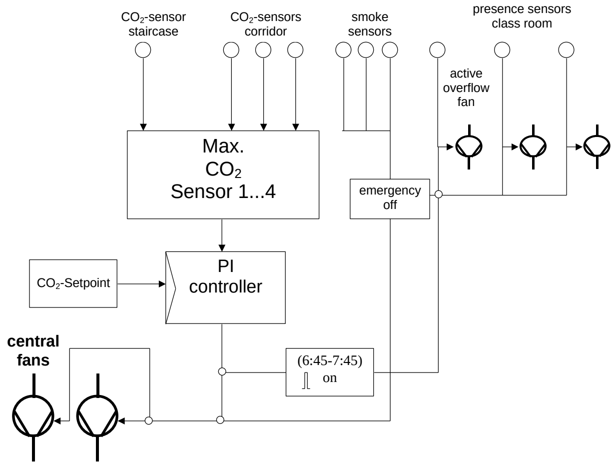


Figure 4: Control scheme for Central Fans and Active Overflow Fans.

3.2 Dynamic Simulation of Indoor Air Quality

In order to simulate the CO_2-concentration as well as the indoor air humidity within the classrooms, corridors, staircase, cloakroom and toilets etc., a multi-zone model was set up with the simulation software CONTAM 3.0 (Walton & Dols, 2010). Figure 5 shows an extract (ground floor only) of the 52-zone model (4 floors). 48 zones are considered as well mixed and four zones (i.e. three corridor zones and the stair case zone) are modeled as 1-D-convection-diffusion zone. The latter was necessary because of the large extent of the corridors in longitudinal direction (length of the corridor 39.5 m in the ground floor, 45.3 m in the first and second floor and height of the staircase 13.1 m). “Contaminant flow in one direction consists of a mixture of convection, the bulk movement of air, and diffusion, the mixing of the contaminant within the air. CONTAM’s primary 1-D convection diffusion model is taken directly from the finite volume method (Patankar, 1980; Versteeg & Malalasekera, 1995). This model divides the zone into a number of equal-length cells and uses an implicit method (with a fast tri-diagonal equation solver) to guarantee stability in computing the contaminant concentrations” (Walton & Dols, 2010).

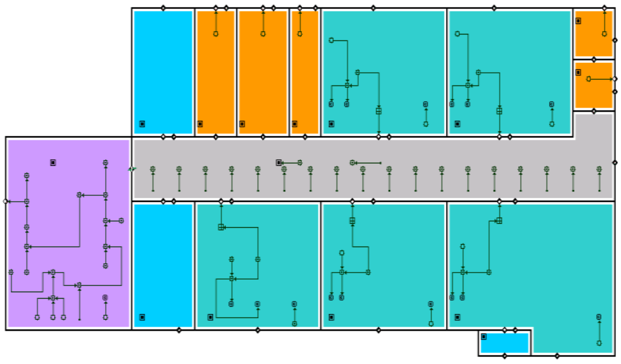


Figure 5: Extract of the 52-zones CONTAM model (sketch-pad); ground floor with detailed 1D-convection-diffusion corridor zone (grey) and staircase zone (purple); mixed zones for class room (cyan), toilets/wardrobes (orange) and storage rooms (blue)

The time schedules of occupation for all occupied zones are implemented in the model. The occupation of the classrooms is mostly five hours a day, starting from 7:45 a.m.. A number of 20 pupils per class at the age of 10 to 14 years (CO_2-source of 12 L/h and H_2O-source of 90 g/h per pupil) were assumed for the simulations.

The calculated results for these boundary conditions (CO_2-concentration of ambient air was assumed to be a constant value of 400 ppm) are shown in Figure 6. The CO_2-concentration in the corridor is limited to values of around 600 ppm. With a flow rate of 700 m³/h per classroom, the CO_2-concentration in the class rooms is limited to peak values of around 1000 ppm.

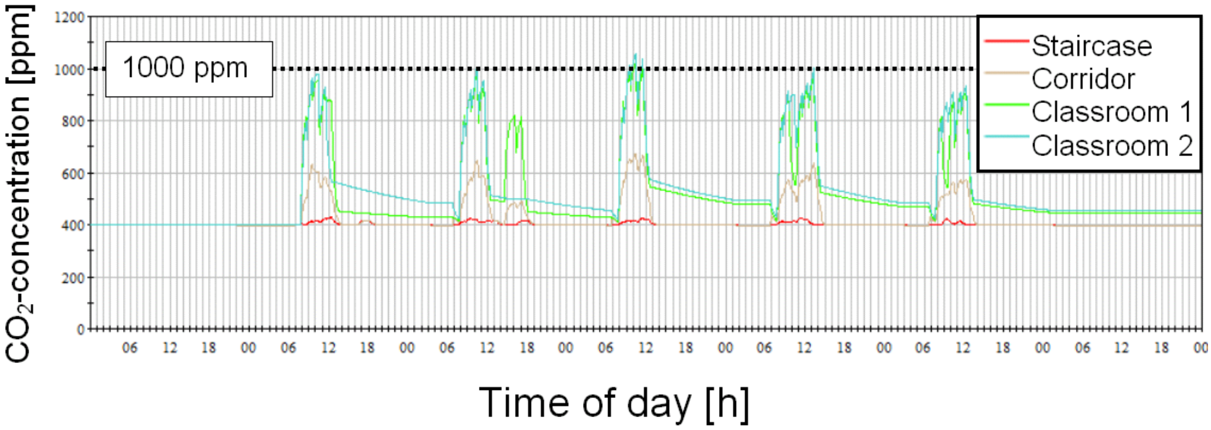


Figure 6: One week of simulation results of CO_2-concentrations in the staircase, the corridor and the class rooms.

3.3 Implementation of the Active Overflow Principle – Example 1

Within the FP7 project "3ENCULT – Efficient Energy for EU Cultural Heritage", the school building "NMS Hötting" in Innsbruck (Austria) is one of the eight case studies for demonstration and verification of energy efficient solutions (Troi & Lollini, 2011). Besides the reduction of thermal losses, a special focus will be on adaptation and optimization of the ventilation system. The active overflow principle as described above was transferred to school buildings. In this case, the high flow rate (around 700 m³/h) calls for a dedicated air distribution system to avoid complains due to draft risk and airborne noise. This was realized by textile hoses for supply air distribution as shown in Figure 7. The air passes (driven by fan) from the corridor via silencers into that hoses, which are perforated by laser for uniform flow distribution. To minimize the sound transmission between the class rooms and corridor, also the passive overflow openings are equipped with silencers.

Figure 7: Silencer and fan-box prototype manufactured by ATREA (top-left), mounting of silencer in the wall (top-right), fan box and textile diffuser in operation (bottom).

The building under investigation is a listed four-story school building (year of construction 1929/30). Figure 8 shows the ground floor plan with four class rooms, a library as well as the toilets and cloakrooms etc. There is a hydraulic heating system with radiators. The cooling in summer is realized by night ventilation via the windows, no mechanical cooling is necessary.
The staircase is directly linked to the open space of the corridors. The fire doors will only be closed in case of emergency. A central heat recovery system ventilates the staircase and the corridors with preheated fresh air. The active overflow system (one for each class room) takes the air from the corridor to the class room and vents the extract air back to it. Finally the air is sucked to the toilets and cloakrooms and from there, via vertical ducts, back to the central heat recovery system located at the attic.

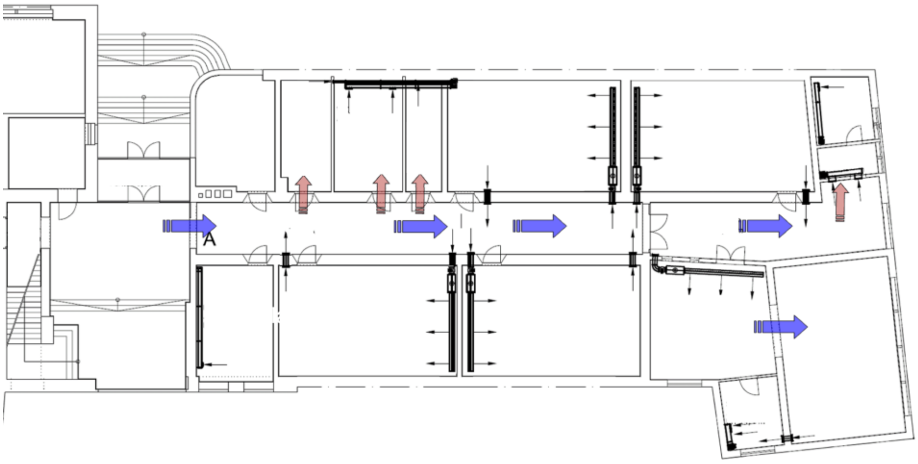


Figure 8: Ground floor (NMS Hötting, Innsbruck, Austria) showing the AOP installation.

Within this demonstration project two class rooms were fully refurbished (internal insulation, advanced lighting system acoustic enhancement) and were equipped with prototypical AOP's built by ATREA®. This ventilation concept is being tested in real use since 2014 and measurements were performed, see Figure 9. It shows the CO_2-concentration in two classrooms with a very similar occupancy profile in direct comparison. In class room No. 1 the overflow ventilation was switched out. In class room No. 2 the overflow fan was in operation with a reduced flow rate of 450 m³/h. Even with this low flow rate, the IDA class 3 (CO_2 concentration below 1400 ppm) was maintained, whereas in the classroom without ventilation the limit value was exceeded within the first hour of lessons.

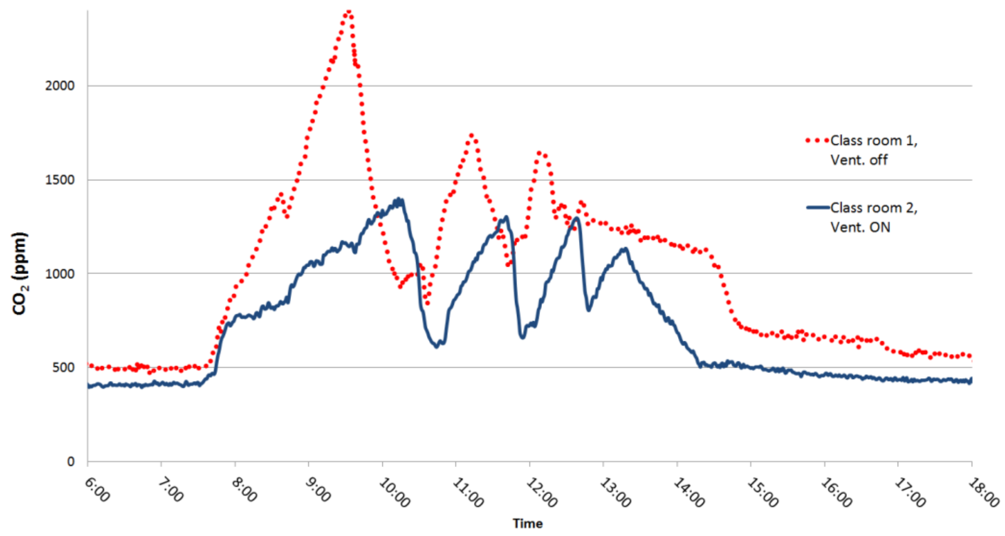


Figure 9: Simultaneous measurement of CO_2-concentration in class room 1 (active overflow fan off) and class rom 2 (active overflow fan running at 450 m³/h)

3.4 Implementation of the Active Overflow Principle – Example 2

Within the FP7 project "Sinfonia - Smart INitiative of cities Fully cOmmitted to iNvest In Advanced large-scaled energy solutions" (Sinfonia, 2014) another school building in Innsbruck (Siegmairschule) will be refurbished. It was inaugurated in 1960 and it is also heritage protected which makes the retrofit of a regular centralized or decentralized ventilation system difficult or even impossible. Based on the positive experiences from the former project (NMS Hötting) the building owner decided to also implement the AOP for the entire building. Only the lower floor will be equipped with a regular centralized system having separate supply and extract air channels. The lower floor hosts mostly workshops, storerooms, etc. and is therefore not well suited to implement the AOP (due to possible odour transport between the rooms). The HVAC-planning for building phase 1 is completed and bidding is underway. 18 classrooms and various other rooms (offices) will be ventilated via the AOP. The respective ventilation unit will have a nominal ventilation rate of 14,000 m³/h. The second building phase will be of equal size is also planned with the AOP. The ventilation unit is placed in the attic above the staircase. The full nominal supply air will be directed into the main staircase via a perforated ceiling. The extract air inlets are place at the end of the corridors (and at the toilets at the end of the corridors). In the west wing the extract air ducts run through the attic back to the central unit. Due to geometric constraints the extract air ducts of the east wing are guided through the basement floor (suspended ceiling) and back to the central unit via vertical shafts in the staircase. A certain number of fire protection flaps are unavoidable, but by guiding the supply air through the staircase and corridors, the number is at least reduced by a factor of two.

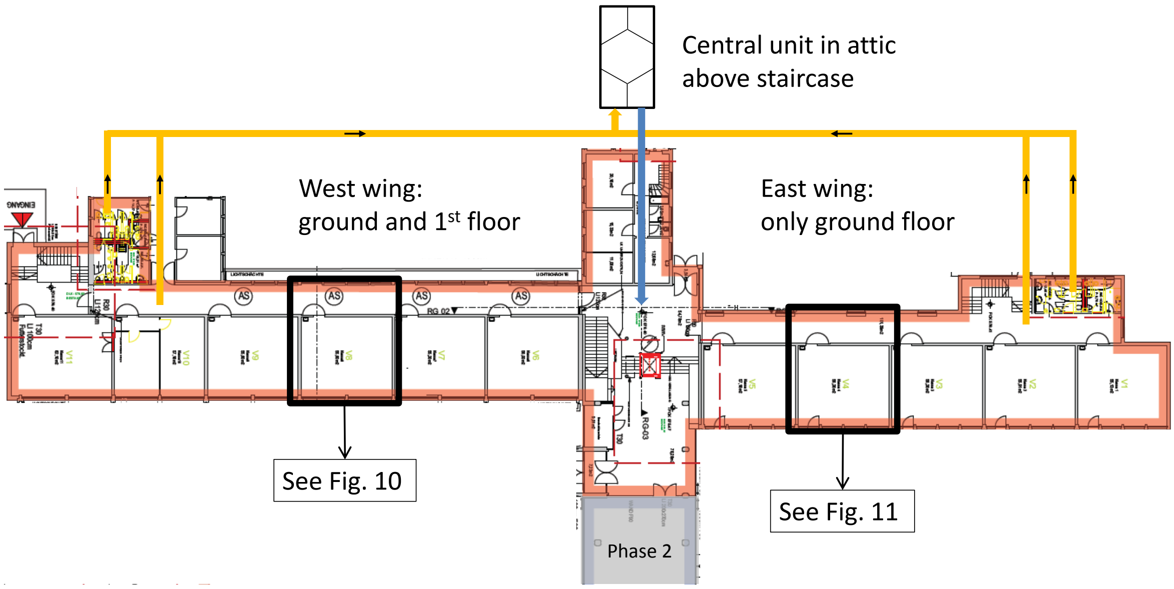


Figure 10: Floor plan of the Siegmairschule including a schema of the AOP as planned by Stiefmüller Hohenauer & Partner, Kundl, Austria (assigned HVAC planner).

This real-case example shows that refurbishment projects often pose a major challenge when it comes to routing the ventilation ducts. Here a combination of two different variations of the AOP is being implemented. The corridor in the ground floor of the west wing allows for a suspended ceiling. Therefore the extract air of the six classrooms can be collected and fed into the extract air duct without mixing with the supply air in the corridors (see Figure 11).

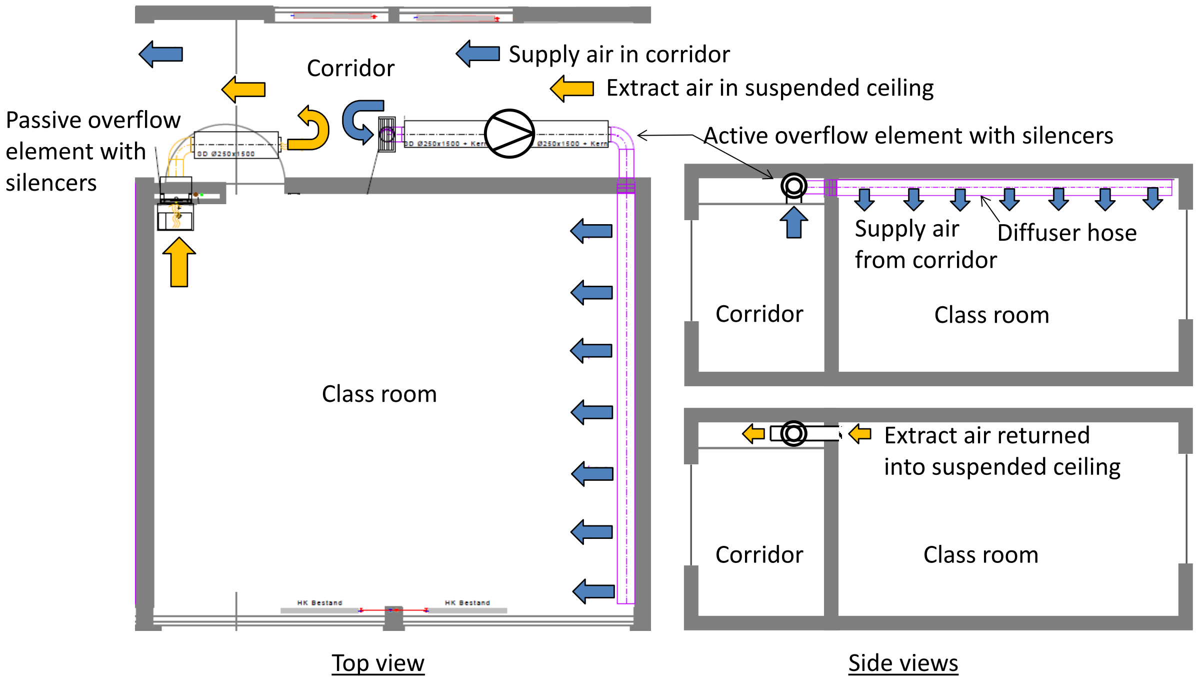


Figure 11: Detail views of one classroom of the west wing corridor. In this AOP variant the extract air from the classroom does not flow back to the corridor. This way good separation of supply and extract air is possible.

This is the ideal situation in terms of ventilation effectiveness and allows for a reduced volume flow into the classroom compared to the AOP-variant where supply and extract air are mixed. In the other two corridors an extract air duct could not be place within the same fire protection area. Besides geometric and spatial challenges, a fire protection flap would be required for each classroom. Therefore the mixed air AOP was chosen (see Figure 12). Here the extract air is guided via a passive overflow element (with silencer) back into the corridor, where it will mix with the supply air. If supply air for the classroom is drawn from the bottom and extract air is returned at the top of the corridor as planned for the east wing, stratification could potentially reduce the mixing. More investigations are needed to quantify this effect and deduce the optimal ventilation rates for the classrooms.

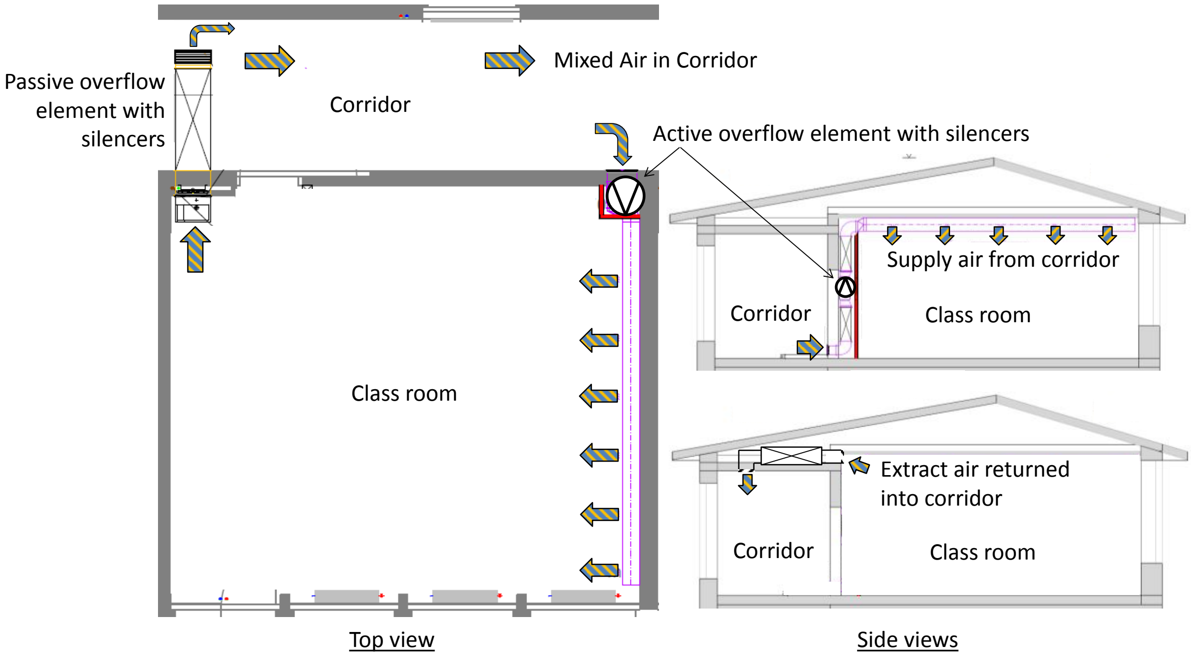


Figure 12: Detail views of one classroom of the east wing corridor. In this AOP variant the extract air from the classroom flows back to the corridor mixing with the supply air. The active overflow element has to be designed for higher volume flow rates since the “supply” air is pre-contaminated.

4 SUMMARY AND CONCLUSION

This paper discusses different ventilation concepts for school buildings with a focus on retrofit-solutions. Here, routing the ducting usually poses a major challenge, especially when restrictions due to heritage protection exist. Depending on the floor plan and the restrictions, either a central or a decentralized system can be chosen. In case of a central system, either vertical ducts into the supply air rooms or active overflow from the corridor can be applied. The vertical ducting solution usually results in high maintenance costs due to the high number of fire protection flaps required. For this reason this work investigates the active overflow principle (AOP) and presents exemplarily cases where it has been or it will be implemented. If realized with no dedicated extract air ducting (for the air leaving the classroom), it has the lowest ventilation efficiency of the compared concepts. If the extract air can be routed all the way to the classroom, no major drawback in terms of ventilation efficiency is expected. As the second example shows, different variants of the AOP can be put into practise within one building. Further investigations are needed to analyse the effect of the possible variants on the ventilation efficiency in more detail.

From the architectural and/or preservation point of view, the AOP is preferable since it reduces the ductwork to a minimum. The control strategy for the central fan as well as the active overflow fans is rather simple and effective. To reduce installation and maintenance

costs, only one CO_2-sensor per corridor and one presence-control sensor per class room can be installed. Simulations and measurements in a prototypical setup indicate that acceptable air quality can be achieved with this ventilation concept. Further studies are needed to put this concept to the proof when implemented on a large scale (full building) and to determine the ventilation efficiency depending on the possible realisation.
As a conclusion the AOP seems to be a very attractive solution for retrofitting school buildings with a mechanical ventilation system with heat recovery, especially when limited space for routing the ducts is available or heritage protection calls for a minimal invasive retrofit.

5 ACKNOWLEDGEMENTS

Investigations were granted by EU-project 3ENCULT: Efficient Energy for EU Cultural Heritage Contract No. 26016.

6 REFERENCES

Patankar, S. V. (1980). Numerical Heat Transfer and Fluid Flow (1st ed.). London: Hemisphere Publishing.

Pfluger, R., & Längle, K. (2013). Minimal Invasive Ventilations Systems with Heat Recovery for Historic Buildings. In 11th REHVA world congress Clima 2013. REHVA.

Sibille, E., & Pfluger, R. (2015). The use of active overflow systems for the distribution of fresh air in apartments. In W. Feist (Ed.), 19th International Passive House Conference (pp. 407–412). Darmstadt, Germany: Passive House Institute.

Sinfonia. (2014). Sinfonia - Projekt Hompage. Retrieved January 10, 2015, from http://www.sinfonia-smartcities.eu/

Sprecher, F., & Estevez, M. (2011). Aktive Überströmer, Bericht des Preisgerichts [Active Overflow Elements, Report from the Jury]. Zurich. Retrieved from https://www.stadt-zuerich.ch/hbd/de/index/hochbau/beratung/gebaeude_energietechnik/projekte_realisiert.html

Toftum, J., Kjeldsen, B. U., Wargocki, P., Menå, H. R., Hansen, E. M. N., & Clausen, G. (2015). Association between classroom ventilation mode and learning outcome in Danish schools. Building and Environment, 92, 494–503. doi:10.1016/j.buildenv.2015.05.017

Troi, A., & Lollini, R. (2011). With a Holistic Approach and Multidisciplinary Exchange towards Energy Effi ciency in Historic Buildings Respecting their Heritage Value. Paris: International Federation of Library Associations and Institutions (IFLA). Retrieved from http://www.ifla.org/publications/international-preservation-news

Versteeg, H. K., & Malalasekera, W. (1995). An Introduction to Computational Fluid Dynamics: The Finite Volume Method (1st ed.). Essex, England: Longman Group Ltd.

Walton, G. N., & Dols, W. S. (2010). CONTAM User Guide and Program Documentation. NIST. Retrieved from http://www.bfrl.nist.gov/IAQanalysis/CONTAM/userguide.htm

INDOOR CLIMATE ASSESSMENT OF A CLASSROOM WITH MECHANICAL VENTILATION AND OPERABLE WINDOWS

Yacine ALLAB[1, 3*], Andrea Kindinis[1], Francesco Causone[2], Sophie Simonet[3], and Annie-Claude Bayeul-Lainé[3]

1 Université Paris-Est, Institut de Recherche en Constructibilité, ESTP, F-94230, Cachan, France.
**Corresponding author: yallab@adm.estp.fr*

2 end-use Efficiency Research Group, Department of Energy, Politecnico di Milano, Via Lambruschini 4, 20156, Milano, Italy

3 Ecole Nationale Supérieure d'Arts et Métiers, Laboratoire de Mécanique de Lille UMR 8107, France

ABSTRACT

Ventilation air may be provided in buildings by means of natural or mechanical strategies. When a HVAC system is installed, thermal comfort and indoor air quality (IAQ) may be controlled with higher precision. However, especially between the 70s and the 90s, mechanical ventilation systems have been installed on formerly naturally ventilated buildings without providing any control for natural ventilation. The two ventilation systems are therefore overlapping, without any energy or comfort oriented control strategy, and the occupants are operating windows without any consistent understanding of IAQ and comfort conditions, neither of energy consumption. It may both lead to a substantial energy waste and to low indoor climate conditions. A typical university classroom with exhaust mechanical ventilation and operable windows without switching control has been assessed, in order to understand how occupant behaviour and different ventilation scenarios may influence indoor climate conditions.

KEYWORDS

Thermal comfort; indoor air quality; ventilation strategies

1 INTRODUCTION

In European countries, 40% of the consumed energy is dedicated to buildings. Whereas, with 46% of the total energy consumption, the building sector remains the largest consumer in France. More than half of this energy is dedicated to ventilation and air conditioning systems. The application of energy policies developed to reduce energy consumption, led in past decades to reduce ventilation rates, decreasing indoor climate quality (ICQ) and causing, especially in the 80s, the appearance of the so called sick building syndrome (SBS). New strategies were nevertheless developed to minimize energy consumption while meeting the requirements for indoor climate quality: CO_2 or humidity driven ventilation systems, high-efficiency heat recovery units and hybrid ventilation strategies.

Low ventilation airflow rates are often associated with problems of IAQ. A ventilation system is designed to ensure a sufficient air change to remove the pollutants contained within the building. In addition, ventilation is closely linked to thermal comfort. In the early 90s, the relationship between thermal comfort and air distribution in indoor environments has been reduced to the notions of «draught rate» and «draught risk». A criterion was included in the standards for thermal comfort through the DR index that estimates the percentage of persons exposed to draught through a combination of air speed, temperature and turbulence (Fanger et al., 1985). In 1998, ASHRAE has launched a study to investigate the effect of DR on thermal comfort. The occupants showed a tendency to prefer the presence of air movements. Other studies have proven that in warm microclimates, occupants felt comfortable with air velocities between 1 m/s and 1.5 m/s (Candido et al., 2010). The distribution of air temperature and velocity is strongly related to the ventilation system. According to the characteristics of the ventilation system (inlet and outlet air flow, inlet air temperature, location of vents) discomfort effects may occur (vertical temperature gradients, draught) (Tomasi et al., 2013).

This study focuses on a school classroom and concerns users thermal perception and thermal comfort measurements; it includes also IAQ measurements, considering CO_2 as an indirect indicator of contaminants related to human activities. The main objective of the present work is to assess ICQ (thermal comfort and indoor air quality) according to various ventilation strategies in a typical French classroom. The investigated classroom is equipped with mechanical exhaust and manually operable windows. The field study includes both physical measurements on thermal comfort and CO_2 concentration, and subjective assessment by means of survey questionnaires.

2 CASE STUDY

The study was performed at ESTP, a civil engineering school located in Cachan (5 km south of Paris). According to the Köppen-Geiger climate classification, the climate in Cachan is defined as Cfb (warm temperate, fully humid and warm summer) with a moderate seasonality. The annual average temperature is around 11°C. In winter the average temperatures are typically below 5°C, and in summer they do not exceed 25°C.

The investigated classroom (Figure 1) is inside Laplace building, a newly constructed building (2008) devoted to teaching. This classroom is equipped with water-filled radiators which operate only during the heating season (early November to mid-April). The classroom is not equipped with a cooling system. In order to guarantee thermal comfort in summer and minimum air changes throughout the year, the classroom is equipped with a mechanical ventilation system (extraction only) and natural ventilation (manually operable windows). The classroom is characterized by a large glazing area (18 m^2) spread over 15 double glazing windows. However, only two windows (facing north and west respectively) are operable. The operable windows can be used both in tilted or swing position. In the present study, the opening of the two windows is performed only in tilted position with an opening area of 0.45 m^2 (for each window).

Table 1: Classroom's characteristics

Classroom	V[m^3]	A[m^2]	WFR	WO	NS	NW	W_w[m]	W_h[m]	W_{oa}[m^2]
Laplace 21	216.4	81.3	0.21	N-W	40	2	0.84	1.39	0.90

Note: V=volume [m^3]; A= area [m^2]; WFR=window to floor ratio; WO= windows orientation; NS= number of seats; NW=amount of operable windows; W_w=windows width; W_h=windows height; W_{oa}=windows opening area

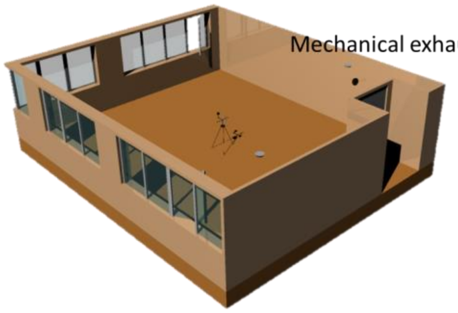


Figure 1: Investigated classroom

3 METHODOLOGY

3.1 Measurement protocol

The field study was performed during the winter period for 12 weeks (from January 5th to March 27th). Before it, a preliminary campaign was conducted in order to analyze the spatial distribution of the indoor climate parameters and to establish a reference point for the following assessment campaign. Homogeneity tests were performed using different spatial measurements at different levels and zones in the classroom according to ISO 7726. The classroom was divided into four zones according to occupation, façades orientation and location of heating terminals. For each zone the operative temperature was recorded at four levels (10 cm; 60 cm; 110 cm; 170 cm). After an analysis of the results, no significant differences were observed. The reference point was then chosen at the center of the effective occupied zone on a level of 110 cm.

The actual assessment campaign was then divided into two sessions. The first session was carried out from January 20th to February 27th and it was dedicated solely to thermal comfort assessment using both physical measurements for PMV calculations, and subjective survey questionnaires. A dedicate indoor climate station was used to record air temperature, globe temperature, air velocity and relative humidity. Table 2 shows the characteristics of the indoor climate station (LSI portable thermal environment monitoring system). The measurements were performed continuously during the occupancy period (from 8:00 am to 5:00 pm) with 1 minute time step. Due to the globe thermometer time response, the measurements were extended 30 minutes before and after the measurement period.

The second session was carried out from March 2nd to March 27th and was dedicated to the study of ventilation strategies effect on ICQ. Four ventilation strategies were considered:

- M1: mechanical extract and infiltration supply (windows closed)
- M2: mechanical extract and windows supply
- N1: natural ventilation on single side (without mechanical extract)
- N2: natural ventilation on two sides (without mechanical extract)

This session included two sets of parallel measurements: long term measurements (continuous measurements for the whole period) and short term measurements (measurements during the occupancy period only). The long term measurements were recorded using a KTHCO2-E data logger (Table 2) including the measurement of air temperature, relative humidity and CO_2 concentration (5 minutes time step). As for the first session, the short term measurements were recorded using the LSI portable thermal environment monitoring system, including the measurements of air temperature, globe temperature, air velocity and relative humidity with 1 minute time step. The outdoor physical parameters were recorded using a KTHCO2-E (air temperature, relative humidity and CO_2 concentration). Wind speed and direction were obtained from the weather database of Météo France.

Table 2: characteristics of measuring instruments

	Quantity	symbol	Measuring range	Accuracy
LSI portable thermal environment monitoring system	Air temperature	T_a	-50-70°C	± 0.1 °C
	Globe temperature	T_g	-40-80°C	± 0.15°C
	Relative humidity	RH	0-100%	± 1.5 %
	Air velocity	V_a	0.01-20 m/s	± 0.05+0.05V_am/s
KTHCO$_2$-E dataloger	Air temperature	T_a	-20-70°C	± 0.5 °C
	Relative humidity	RH	0-100%	±0.88 RH %
	Carbon dioxide	CO_2	0-5000ppm	±50 ppm

3.2 Indoor climate assessment

The thermal comfort assessment is based on the calculation of PMV and PPD indexes according to ISO 7730 methodology. The analysis is carried out comparing the calculated PMV values to the limits fixed by the EN 15251 and ISO 7730 standards. The indoor air quality assessment (second session) is carried out by comparing the measured CO_2 concentrations above outdoor level with the limits fixed by EN 15251 standard. Table 3 summarizes the limits fixed by the EN 15251 for thermal comfort and indoor air quality.

Table 3: The EN 15251 classification

Category	PMV	PPD [%]	CO_2 max above outdoor
I	-0.2<PMV<0.2	< 6	350
II	-0.5<PMV<0.5	< 10	500
III	-0.7<PMV<0.7	< 15	800
IV	PMV<-0.7; or +0.7>PMV	> 15	<800

The second session aims to study the impact of ventilation strategies on both thermal comfort and indoor air quality. Thus, in addition to the analysis of thermal comfort and indoor air quality, the air change rate, i.e. air change per hour (ACH) is estimated for each ventilation strategy. Usually, the ACH estimation is made by means of tracer gas measurements adopting the concentration decay, the constant emission, or the constant concentration method (Sherman, 1990) with SF_6, CO_2 or another tracer gas. Despite the uncertainties of the measurements related to its presence in the air, carbon dioxide is not less interesting for the ACH estimations. In fact, the CO_2 produced by occupants in a building decays exponentially after the occupancy period. For a decay period $t-t_0$ corresponding to the end of occupancy period (C_0, maximum CO_2 concentration) and the end of the decay period, the integral form of the mass balance is obtained as:

$$C(t)- C_{ext}= (C_0-C_{ext}) * \exp [-n (t-t_0)] \quad (1)$$

where, C(t) is the measured CO_2 at time t, C_{ext} the outdoor CO_2 concentration and *n* the ACH. The ACH is then obtained by fitting the concentrations against time with an exponential regression. It is therefore possible to estimate the ACH adopting CO_2 produced by occupants.

3.3 Subjective assessment

A questionnaire survey was performed in order to assess the occupants' perception in parallel to short term measurements, both for the first and for the second session. The questionnaire was set up in accordance with standard EN ISO 10551 and adapted to school environments according to previous studies on subjective assessment in schools (Pereira et al., 2014). Thus, students and teachers were invited to express their perception on ICQ by filling a questionnaire twice a day, according to the school time schedule. The questionnaire was filled by the occupants at the end of each period, respecting the 30 minutes adaptation time, required by the standard EN ISO 10551 for the relevance of the vote.
The questionnaire was divided into three parts. The first part concerns personal information (age, sex, height, weight, clothing for PMV calculation, possible diseases). The second part focuses on thermal comfort perception including:

- overall thermal perception (based on the ASHRAE 7 points scale);
- thermal discomfort on a five point scale: comfortable, slightly uncomfortable, uncomfortable, very uncomfortable, extremely uncomfortable;
- thermal preference on a 7 points scale: a lot warmer, warmer, a bit warmer, no change, a bit colder, colder, a lot colder;
- thermal acceptability on a 2 points scale: acceptable, not acceptable.

The questionnaires, which showed contradictions between the thermal perception, discomfort and preference, were removed. The Thermal Sensation Index (TSI) was calculated as the mean value of the vote on the overall thermal perception, whereas the thermal dissatisfaction was estimated as a combination of the thermal perception (persons who answered ± 2 and ± 3) and the thermal preference (persons asking for a change).
The third part of the survey focuses on the perception of indoor air quality based on the judgment of the odor, and on the air quality and movement perception. Due to the limited amount of space, the analysis of this part is not included in the present paper.
In total, more than 390 subjects were interviewed and 350 questionnaires were validated. In fact, only questionnaires which present no contradiction between the TSI, thermal discomfort index (TDI) and thermal preference index (TPI) were considered reliable. Table 4 lists the occupant's typology (general details) in the investigated classroom.

Table 4: statistical summary of the subjective assessment

Sample size	Gender		Age (years)				Height				Weight (kg)			
	M	F	Max	Min	Mean	SD	Max	Min	Mean	SD	Max	Min	Mean	SD
350	238	112	40	19	22	2	1.97	1.49	1.77	0.9	120	48	71	13

Note: M=Male; F=Female; Max=Maximum; Min=Minimum; SD=Standard Deviation

4 RESULTS AND DISCUSSIONS

4.1 Weather data analysis

Table 5 summarizes the averages values of outdoor parameters (air temperature, wind speed and relative humidity) for the two sessions.
The recorded values correspond to the local winter climate. The average outdoor temperatures were respectively 3.8 °C and 5.7°C for the first and the second session. The average CO_2

concentration was 386 ppm. However, some peak values (501 ppm) were recorded and may be due to the urban effect (pollution peak).

4.2 First session: thermal comfort analysis

Figure 2 shows the cumulative frequency distribution of indoor air temperature, mean radiant temperature and operative temperature. The mean value of operative temperature (20.9 °C) was within the comfort range fixed by the standards. However, considerable differences were remarked during some periods. In fact, the maximum and minimum recorded values were respectively 26.1 °C and 16.5 °C (Table 6). 20 % of the recorded operative temperatures values were below the lower limit (19 °C) fixed by ISO 7730 for school environments in winter periods (22 ± 3 °C).

Table 5: summary of the weather data during the assessment campaign

Parameter/session Symbol	Air temperature T_{out} [°C]	Relative humidity RH_{out} [%]	Wind speed W_s [m/s]	CO_2 concentration [ppm]
Session 1				Not recorded
Mean	3.8	82	3.94	/
Min	-2.6	43	0.00	/
Max	13.1	98	10.00	/
Session 2				
Mean	5.7	85	4.49	386
Min	-4.2	48	0.05	295
Max	17.6	98	16.65	501

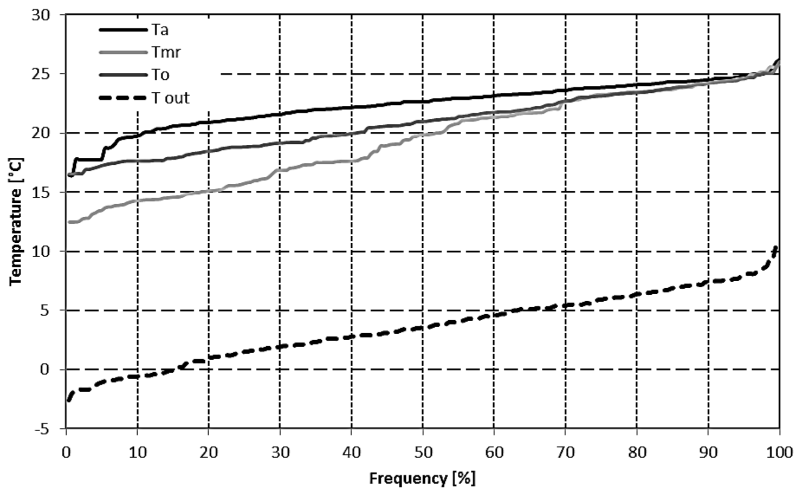


Figure 2: cumulative frequency distribution of air temperature (Ta), mean radiant temperature (Tmr), operative temperature (To) and outdoor temperature (Tout)

Figure 3 shows the cumulative frequency of PMV. Almost 40 % of the PMV values during the occupancy period were below the lower limits fixed by the standards for a classroom in category III, adequate for existing buildings, whereas the value rises to 55% if category II limits are adopted and to 70 % for category I. Thus, the classroom may be assumed to be a cold environment according to the first session of physical measurements.

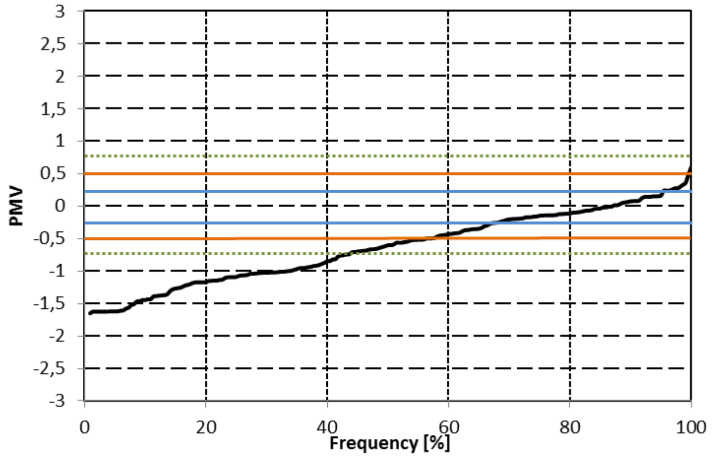


Figure 3: cumulative frequency distribution of PMV for the whole first session plotted with the limits corresponding to the three categories fixed by the EN15251 (blue for category I, orange for category II and dotted green for category III)

Table 6: statistical summary of indoor measured physical parameters

Parameter	RH	T_a	T_{mr}	T_o	V_a
Max	67	26.2	25.9	26.1	0.11
Min	15	16.5	12.5	16.5	0.00
Mean	36	22.4	19.5	21.0	0.05
SD	14	1.9	3.8	2.4	0.07

Note: Max=Maximum; Min=Minimum; SD=Standard Deviation

The analysis of the subjective responses leads to some different conclusions. The mean TSI was, in fact, around -0.3, i.e. within category I limits. In order to better understand the reported discrepancy, the TSI and the PMV were contrasted against the operative temperature measurements (Figure 5) and the adaptive PMV (aPMV) proposed by Yao (Yao et al., 2009) was additionally calculated, according to the cold climate:

$$aPMV=PMV \div (1+a\times PMV) \quad (2)$$

In his adaptive PMV, Yao have estimated the "a" factor to be 0.293 and -0.125 for warm and cool conditions respectively. According to the results, the "a" factor was set as -0.125.

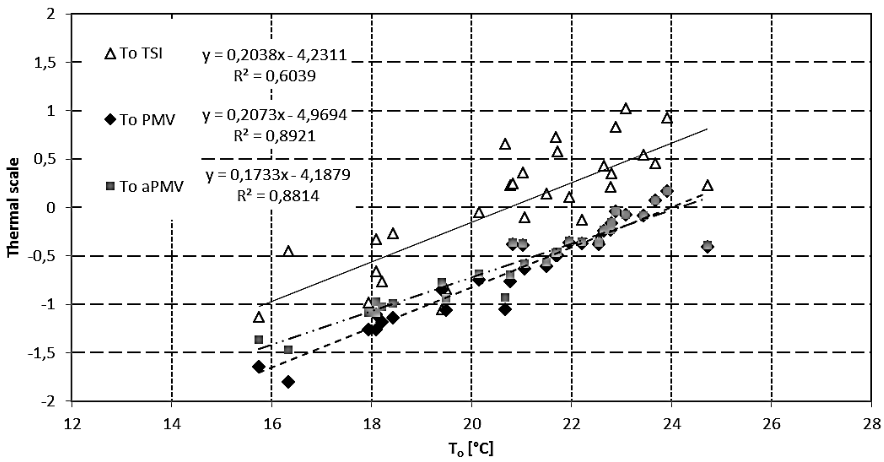


Figure 4: comparison between TSI; PMV and $_a$PMV with T_o

The results plotted in Figure 4 shows that occupants are less sensitive to cold condition compared to the prediction made on the basis of standard PMV. In particular thermal neutrality results between 20 and 22 °C for the real occupants, whereas it is around 24 °C when applying PMV and aPMV models.

4.3 Second session: indoor climate quality analysis

According to the outdoor data analysis, a total of 12 days were selected for the analysis of the ventilation strategies (three days for each strategy). Figure 5 shows the cumulative frequency distribution of operative temperature for the four ventilation strategies, while table 7 gathers average values of indoor and outdoor physical parameters for each ventilation strategy. As expected, the single side natural ventilation strategy (N1) shows a slight overheating. On the contrary, double side natural ventilation (N2) shows 80% of the occupied period below 19°C. The two mechanical strategies have a considerable gap. In fact, the mechanical exhaust with window supply (double side) M2, shows the lowest temperatures. The mechanical exhaust coupled with infiltration supply (M1) shows temperatures in the middle of all the other configurations. These variations can be explained by the different amount of fresh air supplied by each strategy. However, also night time ventilation could affect the results, since, to be able to estimate the ACH, windows were left open in scenario N1, N2 and M2 also at night time, and this may have affected thermal mass activation and temperatures drop.

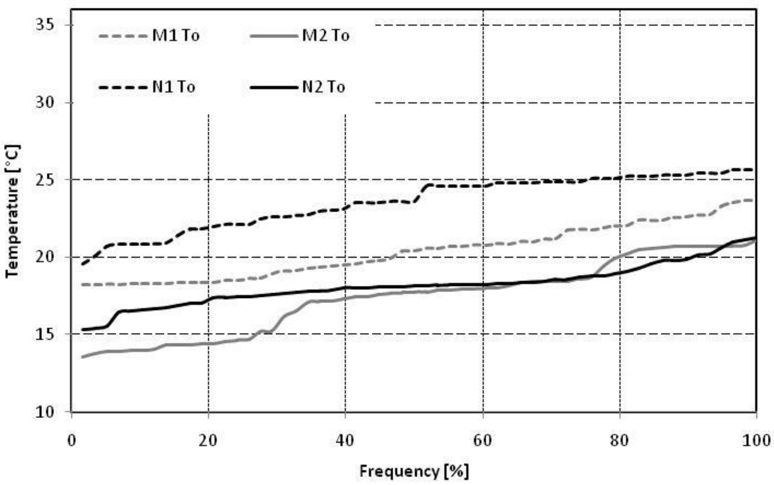


Figure 5: cumulative frequency distribution of the operative temperatures for the four strategies

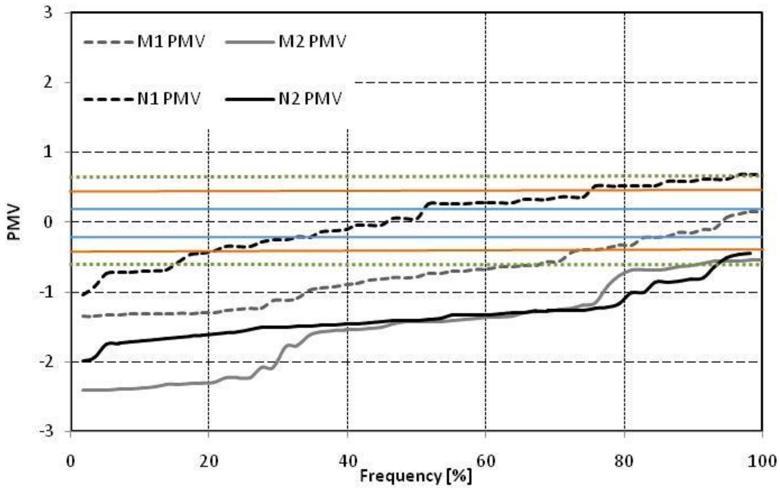


Figure 6: cumulative frequency distribution of the predicted mean vote for the fourth strategies with the three limits corresponding to the three categories fixed by the EN15251 (blue for category I, orange for category II and dotted green for category III)

Figure 6 shows the cumulative frequency distribution of the PMV for the four strategies. The trends for each strategy correspond to the trends shown for operative temperatures and it confirms the substantial influence of operative temperature on PMV. Strategy N1, i.e. natural ventilation on one side, shows the warmest conditions, with 40% of the values out of category II. M2 and N2 strategies show, instead, 100% of the values below the limits of the category II.

CO_2 concentration plotting (Figure 7) clearly shows the influence of occupancy on IAQ. Peak values are, in fact, recorded during the occupation time whereas the lowest values happen during non-occupied periods (nights and weekends). Students and teachers are the only source of CO_2 in the room, so CO_2 may be assumed as an indirect indicator of human related bio-effluents or contaminants related to human activity. During the occupancy period, the CO_2 concentration inside the classroom reaches 4900 ppm (Figure 7) an extremely high value. Figure 8 shows the cumulative frequency distribution of CO_2 concentration above outside. More than 20% of the measured concentrations exceeded outside values for more than 2500 ppm, with a peak value of 4500 ppm. Quite beyond the limits reported in standard EN 15251.

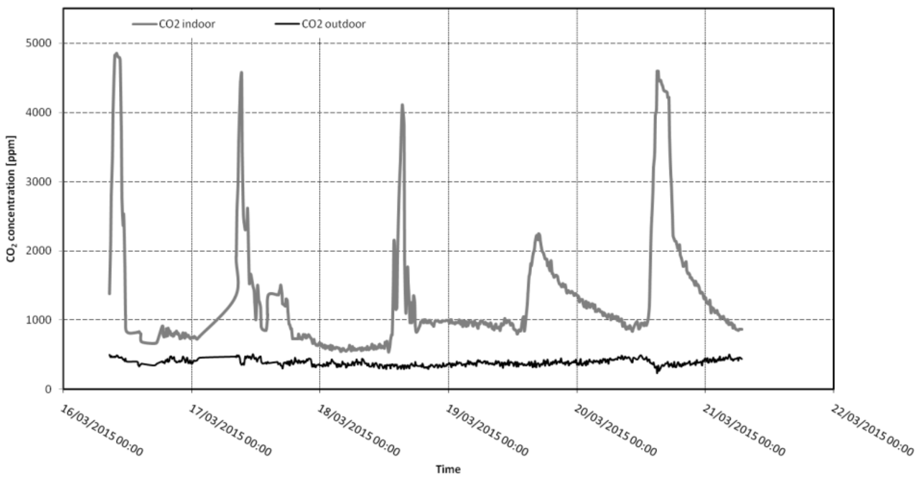


Figure 7: inside, outside CO_2 concentration during a reference week

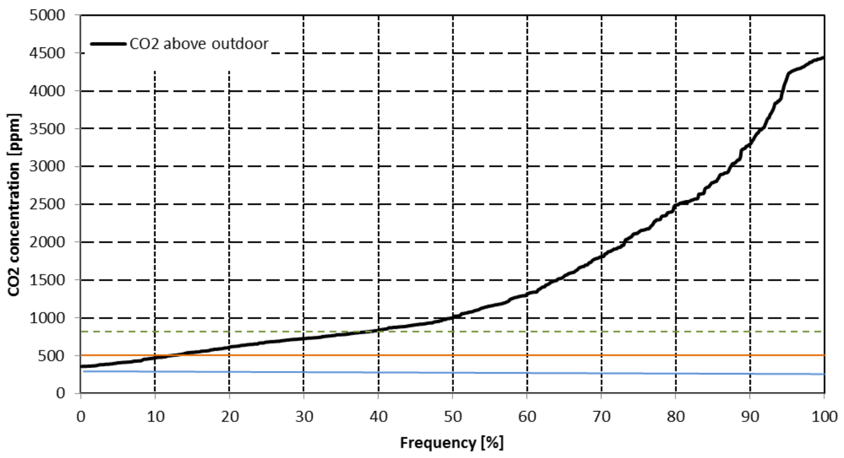


Figure 8: cumulative frequency distribution of CO_2 concentration above outdoor value, plotted with the three limits corresponding fixed by EN15251

As explained in section 3.2, the ACH was estimated from the end of the occupancy period, assuming a maximum value of CO_2 concentration, until the decay to the outdoor level. Table 7 shows the mean values of ACH for each strategy. In fact, three decay periods were selected for each strategy. As expected, the most effective strategy is M2, i.e. mechanical exhaust and windows supply. The ACH values for the two mechanical strategies M1 and M2 were respectively 0.42 h^{-1} and 0.69 h^{-1}. The natural strategies show lower values, 0.06 $h^{-1}$and 0.19 h^{-1} for N1 and N2 respectively.
The ACH values are nevertheless quite low. Previous studies which were performed on mechanically ventilated classroom shows ACH values within a range of 0.73-1.91 h^{-1}(You et al., 2011). Others studies (Dorizas et al., 2015) presented values around 0.29 h^{-1}in a naturally ventilated classrooms.

Table 7: averages values of indoor and outdoor physical parameters for each ventilation strategy

Parameter/strategy	N1	N2	M1	M2
Operative temperature	23.6	18.2	20.4	17.4
Outdoor temperature	8.6	8.6	8.9	8.1
Wind speed	3.13	3.35	3.84	4.23
ACH	0.06	0.19	0.42	0.69

5 CONCLUSIONS

This paper presents the results of an in-situ experimental campaign and survey related to ventilation strategies and thermal comfort in a typical French classroom during the heating season. The survey includes thermal comfort and indoor air quality assessment using four ventilation strategies. Long term and short term experimental measurements were made in parallel to the survey. The results on thermal comfort show that the classroom was slightly cold. However, the subjective results collected through 350 questionnaires, showed a sensible adaptation of occupants to cold conditions, whereas they show more sensitive to warm conditions.
The results of indoor air quality based on the CO_2 concentrations show values quite above the thresholds suggested by standards. The CO_2 concentration peak values reach 4500 ppm above outdoor level, which is consequence of the poor ventilation and of the high occupation value.
The results of the second experimental session, which focused on the assessment of various ventilation strategies, showed significant variations among them. As expected, the strategies

based on mechanical exhaust showed the best results in terms of ventilation rates, although the air change rates resulted always very low. It was not instead possible to establish whether mechanical or natural ventilation performs better in terms of thermal comfort. Further investigations on ventilation air flow distribution (tracer gas measurements) will be undertaken to better characterize the ventilation effectiveness of the investigated ventilation strategies.

6 REFERENCES

Candido, C; de Dear, R J; Lamberts, R; Bittencourt, L. (2010). Air movement acceptability limits and thermal comfort in Brazil's hot humid climate zone. *Building and Environment* (45), 222-229.

Carrilho, J D; Mateus, M; Batterman, S; da Silva,M G. (2015). Air exchange rates from atmospheric CO2 daily cycle. *Energy and Buildings*, (92), 188-194.

Dorizas P V; Assimakopoulos, M; Helmis, C; Santamouris, M. (2015). An integrated evaluation study of the ventilation rate, the exposure and the indoor air quality in naturally ventilated classrooms in the Mediterranean region during spring. *Science of the Total Environment* (502), 557-570.

Fanger, P O; Ipsen, B M; Langklide, G; Olesen, B W; Christensen, N K; Tanabe, S. (1985). Comfort limits for asymmetric thermal radiation. *Energy and Buildings*, 225-236.

Pereira, L D; Raimond, D; Corgnati, S P; da Silva,M G. (2014). Assessment of indoor air quality and thermal comfort in Portuguese secondary classrooms: Methodology and results. *Building and Environment* (81), 69-80.

Sherman, M H. (1990). Tracer gas techniques for measuring ventilation in a single zone. *Building and Environment* (25), 365-374.

Tomasi, R; Krajkic, M; Simone, B; Olesen, B W. (2013). Experimental evaluation of air distribution in mechanically ventilated residential rooms: Thermal comfort and ventilation effectiveness, *Energy and Buildings* (60), 28-37.

Yao, R; Li, B; Liu, J. (2009). A theoretical adaptive model of thermal comfort – Adaptive Predicted Mean Vote (aPMV) Running, *Building and Environment* (44), 2089-2096.

You, Y; Niu, C; Zhou, J; Liu, Y; Bai, Z; Zhang, J; He, F; Zhang, N. (2012). Measurements of air exchange rates in different indoor environments using continuous CO2 sensors. *Journal of Environmental Sciences*, (24), 657-664.

VENTILATION STRATEGIES FOR THE DEEP ENERGY RETROFIT OF A KINDERGARTEN

Francesco Causone*[1], Amin Moazami[1], Salvatore Carlucci[2], Lorenzo Pagliano[1], and Marco Pietrobon[1]

1 end-use Efficiency Research Group,
Department of Energy, Politecnico di Milano,
Via Lambruschini 4, 20156, Milano, Italy
**Corresponding author:*
francesco.causone@polimi.it;

2 NTNU Norwegian University of Science and Technology, Department of Civil and Transport Engineering, Høgskoleringen 7A, NO-7491 Trondheim, Norway

ABSTRACT

The scientific literature often reports example of educational buildings with extremely poor ventilation performance. An in-field investigation for the environmental and energy assessment of a kindergarten in Milano, confirmed that operable windows were not operated when the average daily temperature dropped below 14 °C, jeopardizing indoor air quality and kids learning performance. Seven different ventilation strategies were therefore simulated, in order to evaluate the one that better fitted a general project of deep energy retrofit of the building, including building envelope and systems. The best scenario resulted to be the one using hybrid ventilation at nighttime and mechanical ventilation at daytime. Both energy and thermal comfort conditions were evaluated and a tradeoff between them was established. Nighttime ventilation showed to be extremely effective in improving thermal comfort conditions, during the cooling season. It resulted much better than mechanical ventilation in the simulated case study. Simulations show that under moderate weather conditions and if the building is properly operated (ventilation, lighting and solar screening systems) the retrofitted building may perform well also without additional active cooling.

KEYWORDS

Hybrid Ventilation, Mixed Mode, Mechanical Ventilation, Energy Retrofit

1 INTRODUCTION

The Italian educational buildings stock consists of 52 000 buildings for a total covered surface of 73.3 million square meters; around 63% of them constructed more than 40 years ago (CRESME, 2014). The large majority of these buildings are not equipped with mechanical ventilation and thus rely completely on manual opening of windows to provide ventilation air change (Legambiente, 2014). Similar scenarios are shared by many other countries (Daisey et al., 2003; Santamouris et al., 2008; Wyon et al., 2010). The analysis of CRESME (2014) shows also that the Italian school building stock could achieve energy savings of about 48.3 % and shift from a current energy consumption rate of 9.6 TWh/a to a target value of 5.0 TWh/yr. These results may be obtained improving opaque and transparent envelope performance, enhancing building systems efficiency and optimizing building management and control. A typical consequence of this kind of intervention is the drastic drop of air infiltrations, substantially enhancing the building's airtightness. This is very good from an energy point of view but it may have drawbacks in terms of indoor air quality (IAQ), which is typically already very bad in existing educational buildings. The problem of insufficient ventilation in schools appears, in fact, to be quite common (Daisey et al., 2003; Santamouris et al., 2008; Wyon et al., 2010). In many school buildings, the CO_2 concentration reached very high values, quite above what is suggested in relevant standards and building codes (Brelih, 2012; Danish Building Regulations, 2010; Dimitroulopoulou, 2012; Hasegawa et al., 2012). Insufficient ventilation in schools has been linked with respiratory and general symptoms, infectious diseases and impaired learning outcomes. Poor ventilation is also associated with higher levels of chemical pollutants, and problems with mold and dampness (WHO, 2011). The present paper investigates the possibility to include a dedicated ventilation strategy in the deep energy retrofit of a kindergarten, in order to improve the IAQ of the building while controlling energy requirements for ventilation. A decentralized ventilation system is studied, included in prefabricated modules for the renovation of transparent and opaque envelope components and solar screening. Different ventilation options are compared in terms of energy and thermal comfort performance in order to define a trade-off between occupants' needs and retrofit aims.

2 BUILDING DESCRIPTION

The analyzed building is a kindergarten built in the 80s. The one-story building has a length of 44 m (south-west and north-east façades) and a depth of 23 m (south-east and north-west façades). Around 35 % of the ground floor is dedicated to children activities and the rest to the staff and service areas. The building has a gross floor area of 944 m^2, a net floor area of 855 m^2 and a gross volume of 3 422 m^3 (S/V ratio equal to 0.77 m^2/m^3).

Figure 1: (left) Picture of the southwest façade; (right) kindergarten plan view including the five monitored rooms.

Walls description: the existing building is a typical heavy-prefabricated building made of precast concrete panels including a thin polystyrene layer. The U-value of the walls before retrofit is estimated to be 1.0 W/(m^2 K).

After the retrofit the façade will be covered with prefabricated modules including mechanical ventilation and automated solar shading system. The U-value of the walls after retrofit will be 0.1 W/(m^2 K).

Roof description: the existing roof is a pitched metallic plate with no insulation, placed upon a horizontal concrete slab (Predal system). The U-value of the roof before retrofit is estimated around 0.9 W/(m^2 K). The metallic plate will be removed and a new insulation layer will be laid on the existing slab (approximately 38-40 cm of mineral wool). After the retrofit the U-value of the roof will be 0.1 W/(m^2 K).

Windows description: the existing windows are single pane windows with aluminum frames. In addition to the low thermal performance, the low airtightness of the existing windows cause a high infiltration loss. The U-value of the windows before retrofit is estimated to be 5.85 W/(m^2 K). The post retrofit triple glazing windows will be integrated in the prefabricated façade modules. The U-value of windows will be 0.73 W/(m^2 K).
Heating system: a natural gas boiler for heating is currently installed in the kindergarten in combination with metal radiators, whereas a connection to the local district heating system will be provided after the retrofit.
Ventilation system: currently, no mechanical system is available in the building and the ventilation is accomplished by manual operation of the windows. A new decentralized ventilation system will be installed inside the prefabricated façade with high-efficiency heat recovery units with a nominal sensible recovery efficiency of $\eta = 0.80$.
The indoor environmental conditions were monitored in the building from July 2014 to May 2015. Figure 2 – left, shows the measured operative temperature in room 5. It is possible to check discomfort conditions against category I boundaries according to standard EN 15251 (CEN, 2007). Constant Fanger's boundaries are plotted from 15th October to 15th April (heating season) following Table A.3, Annex A, of EN 15251 for kindergartens. According to EN 15251 specifications, the adaptive model limits are applied during the free running period only, when no mechanical system is operating. Both Fanger's and adaptive comfort models were developed for grown up persons and mostly for office spaces, however the comfort boundaries provided by EN 15251 for category I, i.e. for very sensitive persons, may be considered quite restrictive and adequate for a kindergarten, while waiting for a dedicated future comfort models for kids. It is quite evident that indoor temperature drops down during Christmas holiday, when the heating system is off, and similar, though less dramatic, patterns are clear during winter weekends and at nighttime. This is evidence of the poor thermal behaviour of the building envelope.

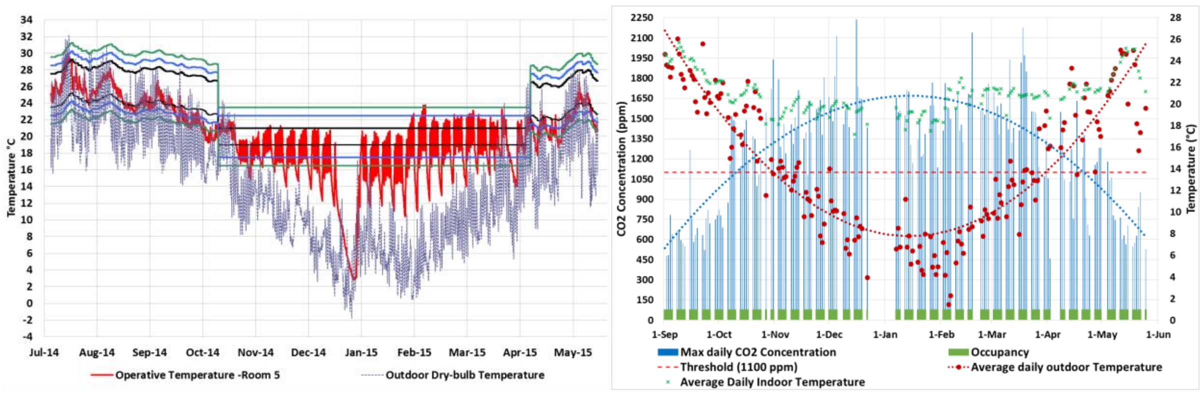


Figure 2: (left) Operative temperature in room 5, and outdoor dry-bulb temperature (DBT) between 8/7/2014 and 25/05/2015 compared to EN 15251 comfort limits (black, blue and green lines corresponding to the limits for categories I, II and III, respectively – (right) Maximum daily CO_2 concentration in room 4 versus average indoor and outdoor air temprerature over occupided period

CO_2 concentration was monitored in room 4 (Figure 2 - right), which is the common space where children play and spend a large part of their time. During August the building is unoccupied therefore the recorded average value of 400 ppm may be considered as the average background outdoor level. After September, noticeable picks are recorded in the room, with values that substantially exceed the reference value of 700 ppm above the background level (ANSI/ASHRAE, 2013), i.e. beyond 1100 ppm. This evaluation should be made under steady-state conditions, however, the recorder picks go far beyond the threshold, showing that the building needs a better ventilation strategy. During September and May, when outdoor temperature is quite mild, indoor CO_2 concentration is relatively low, as consequence of windows opening, whereas during all the winter months CO_2 levels leap above the recommended threshold, showing that manual windows opening is driven by outdoor temperature and not by indoor air quality perception (Wargocki and Da Silva. 2015). Figure 2 shows that windows start to be effectively operated, and thus CO_2 concentration decreases, when outdoor average daily temperature is in the range between 14 and 18 °C, and they keep being operated for higher temperatures, whereas they are closed for lower ones (Causone, 2015).

Energy bills for year 2011 to 2013 report an average value of 142 kWh/m^2 for heating and domestic hot water and 35 kWh/m^2 for electricity. Space heating data, normalized according to heating degree-days (HDD), i.e. winter severity correspond to 202 kWh/m^2 using the weather data used for the following energy simulations.

The building measurements show clear evidence of poor indoor air quality, low performance of the existing envelope and inefficient heating and lighting systems.

The energy retrofit strategy was therefore defined targeting the goals reported in Table 1.

Table 1: Building retrofit strategies

Retrofit target	Strategies
Energy savings	• reducing energy needs for space heating; • reducing all the final energy uses by improving the efficiency of building systems; • adopting passive strategies whenever possible, while avoiding the installation of active cooling systems; • installing new generation systems using renewable energy sources; • reducing both construction time (to limit the disturbance or interruption of the educational service) and cost (to make the intervention feasible).
Indoor climate quality	• improving IAQ, by developing a ventilation strategy; • guaranteeing adequate thermal comfort condition all year long;

3 METHODOLOGY

The objective of this study is to evaluate the performance of different ventilation strategies in the process of energy retrofit of the existing kindergarten. Evaluations are made on the basis of primary energy, delivered energy and thermal comfort after retrofit. The analysis is developed following three steps. As a first step energy savings due to various ventilation scenarios are calculated. In this regard, two energy criteria were considered for each scenario. As a second step, a long-term evaluation of the thermal comfort conditions is performed using method A of Annex F of EN 15251. Based on this method, the percentage of time (occupied hours) outside the various comfort ranges for each scenario is calculated. In the end, the post-retrofit best scenario is selected and compared against the pre-retrofit situation. All the analyses are based on energy simulations.

3.1 Energy simulation

A numerical model of the building was developed to:

- optimize the selection of opaque and transparent envelope thermal insulation;
- optimize the ventilation strategy;
- define a solar control strategy;
- check energy needs and uses to implement a zero-energy approach;
- check indoor environmental conditions.

In this paper we will focus on the optimization of the ventilation strategy only. The energy simulation of the building was performed using the building performance simulation tool EnergyPlus (Crawley, 2001), version 8.1.0. The physical models and algorithms for calculating heat exchanges have been selected with a trade-off between precision and computation time. The heat conduction through the opaque envelope was calculated via the conduction transfer function method with four time steps per hour. Natural ventilation in the classrooms and corridors through dedicated window openings was simulated using the airflow network model. The minimum outdoor ventilation rate was set according to the national standard (UNI, 1995). School working schedule, number of occupants, equipment and lighting were based on interviews with teachers and building managers. The metabolic activity rates were calculated according to the definition of a "standard kid" (Fabbri, 2013).
Seven different scenarios, including a purely mechanical condition, were modeled and compared on the basis of energy and thermal comfort results (Table 2).

Table 2: Ventilation scenarios considered for simulation

Scenario Code			Winter			Summer		
			Heating System	Ventilation		Cooling System	Ventilation	
				Mechanical	Natural		Mechanical	Natural
A		Day	✓	✓	-	✓	✓	-
		Night	-	-	-	-	-	-
B	1	Day	✓	✓	-	✓	✓	-
		Night	-	-	-	-	-	✓
	2	Day	✓	✓	-	✓	✓	✓
		Night	-	-	-	-	-	✓
	3	Day	✓	✓	-	✓	✓	✓
		Night	-	-	-	-	✓	-
C	1	Day	✓	✓	-	-	✓	-
		Night	-	-	-	-	-	✓
	2	Day	✓	✓	-	-	✓	✓
		Night	-	-	-	-	-	✓
	3	Day	✓	✓	-	-	✓	✓
		Night	-	-	-	-	✓	-

The target of ventilation strategies is to provide adequate indoor climate conditions while reducing energy use directly (electricity) or indirectly (heating, cooling) connected to the ventilation.

Scenario A:

In this scenario an active system is applied for both the heating and the cooling period. The set-point temperature is 24 °C and 20 °C for cooling and for heating respectively. Minimum renovation air change rates, as defined by the national standard (UNI, 1995), are provided by a decentralized mechanical ventilation system with a high-efficiency sensible heat recovery unit. Ventilation is provided according to occupation schedules. This system is working all year long and it should guarantee stable environmental conditions during the entire occupation time.

Scenario B:

The active heating and cooling systems are set as for scenario A, whereas three different ventilation strategies are studied, adopting various mixed mode (or hybrid) settings. This system follows the logic presented in Figure 3. When the defined conditions are met the mechanical ventilation is switched off and the outdoor fresh is introduced to the zone through automated windows opening on the external façade and core light-wells to fulfill the indoor air quality.

Scenario C:
The active heating system follows the same schedule as scenario A and B, whereas no active cooling is foreseen in summer. During this period thermal comfort conditions are provided only by daytime and nighttime ventilation, following the logic reported in Figure 3 (Breesch, 2005). If the conditions are favorable for natural ventilation, the mechanical ventilation system is disabled and the automatic window openings are operated.

4 RESULTS

4.1 Energy

Two indicators were chosen to compare the energy performance of different scenarios: the delivered energy and the primary energy. An energy breakdown is provided for each ventilation scenario in term of delivered energy in Figure 4 – left and of primary energy in Figure 4 – right.
From the energy point of view, scenario C1 shows the best performance, whereas B2 is the best scenario among the ones including active cooling. In particular, scenario B2 shows a reduction of 68.5 % in term of delivered energy for cooling, compared to scenario A.

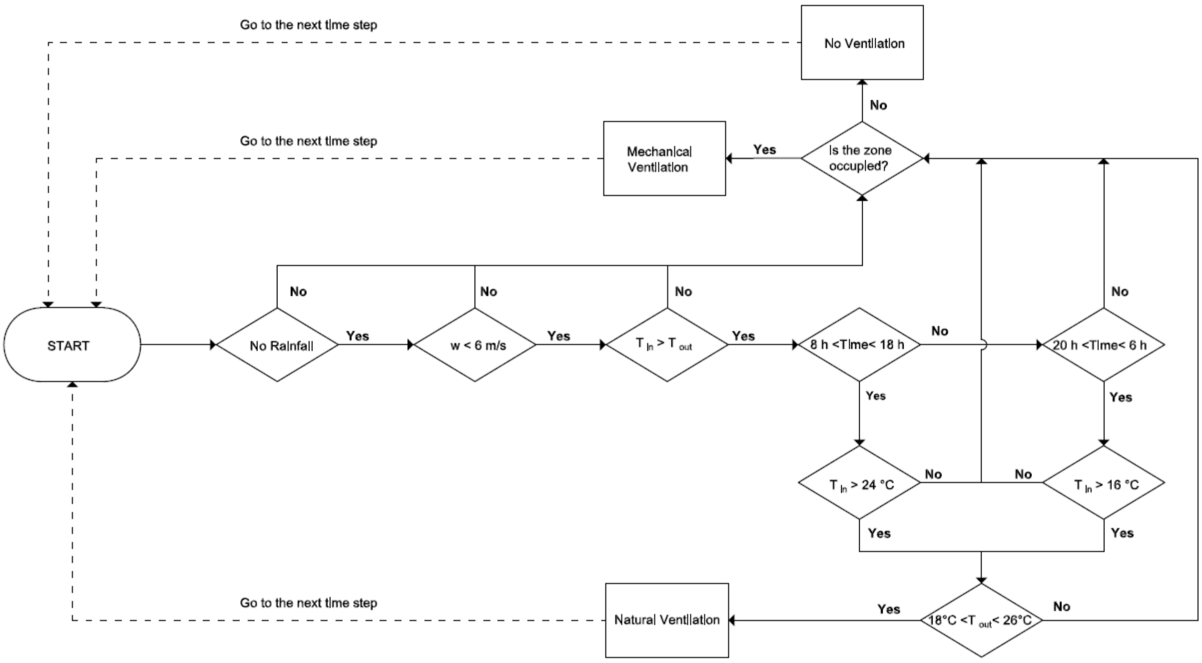
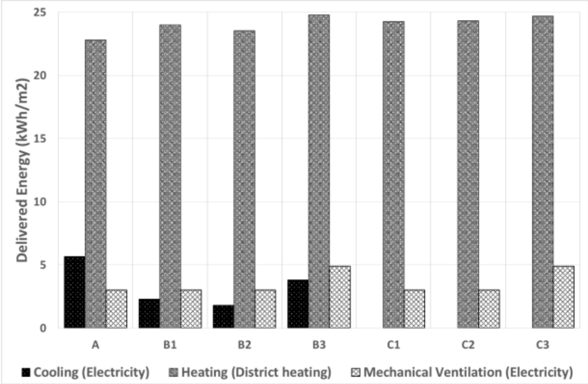


Figure 3: Logic control for Hybrid Ventilation (at each Time Step)



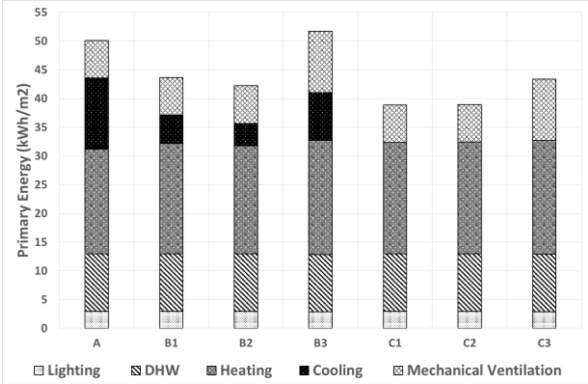


Figure 4: (left) Delivered energy for each scenario (electricity for mechanical ventilation and cooling; thermal energy from district heating for space heating); (right) Primary energy for each scenario.

The primary energy reported for scenario C1 results 22.4 % lower of scenario A and 7.8 % lower of scenario B2 (Figure 4). The delivered energy for mechanical ventilation is, nevertheless, almost constant for scenarios A, B1, B2, C1, and C2. It means that natural ventilation is not really effective during the occupied hours but mostly in the early morning before occupation or in the late evening and at nighttime. Its role is therefore not explained in terms of reductions of delivered energy for mechanical ventilation but in terms of thermal comfort. This is confirmed by the fact that the delivered energy for mechanical ventilation rises for scenarios B3 and C3, when it is also used at nighttime. In case C1, C2 and C3 it should nevertheless be noticed that no delivered energy for cooling is considered, and that only natural ventilation controls thermal comfort conditions.

4.2 Indoor climate

Minimum ventilation air change rates according to the national standard (UNI, 1995) are provided in all scenarios, though actual ventilation rates may be higher under natural ventilation conditions and thermal comfort results also different. Scenario A and B has fully active heating, cooling, and mechanical ventilation systems, which provides indoor thermal comfort during the winter and summer. In scenario C, with an active heating system for winter and no active cooling system for summer, the indoor thermal comfort in cooling season is evaluated using method A of Annex F of EN 15251 (CEN, 2007). The percentage of time (occupied hours) outside the comfort ranges has been calculated as a long term indicator of discomfort (Figure 5 – left), adopting four thresholds of operative temperature, i.e. 24, 25, 26 and 27 °C. The best scenario is C2, for all of three analysed rooms. Adopting the 24 °C threshold, that is the set-point temperature for active cooling in scenario A and B, and focusing on room 5, the percentage out of range results 29.4 % in scenario C3 whereas 15.1 % in scenario C2.

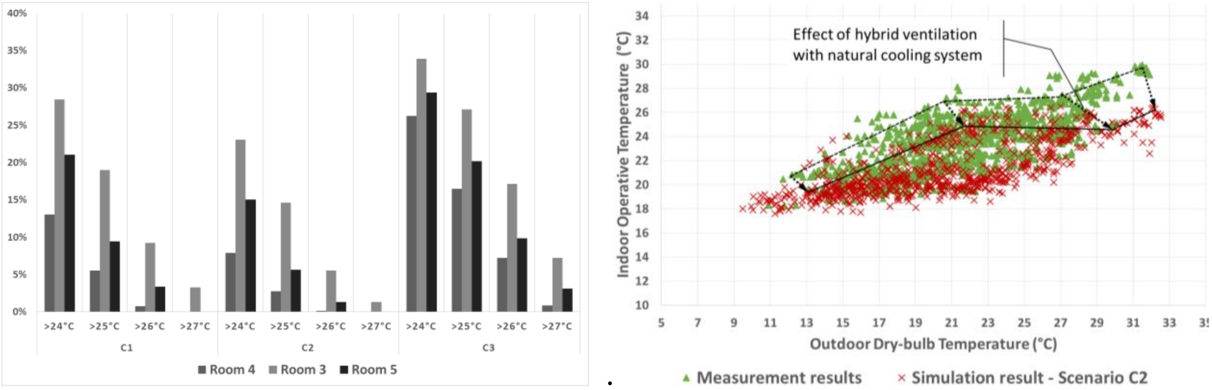


Figure 5: (left) Percentage of time out of range; (right) Indoor operative temperature as a function of outdoor dry-bulb temperature during the occupied hours from 09/07/14 to 25/05/15 (real data against C2 scenario simulations at reference room 5)

Figure 5 reports indoor operative temperature in room 5, as function of outdoor dry-bulb temperature for the existing building (based on measured data) and for the energy simulation (scenario C2). Measured data show indoor temperatures up to 30 °C when the outdoor temperature is above 30 °C. In simulated data, instead, under similar outdoor conditions the indoor temperature reaches the maximum value of 26 °C. A clear decrease of maximum indoor operative temperature is identifiable in the diagram of Figure 5.

4.3 Pre- and post-retrofit energy performance

The energy required for space heating and the production of domestic hot water of the existing building, relying both on the same boiler, were normalized by using a regression model ($R^2 = 0.87$) made on the basis of the metered data and heating degree days (HDD) from 2010 to 2013. Space heating value was substantially higher than the one of domestic hot water, therefore the correlation with HDD is motivated. The energy consumption of post-retrofit are calculated and compared to the existing situation in order to evaluate the potential energy savings of the energy retrofit (Table 3).
Energy simulations shows, for scenario C1 and C2, thermal comfort conditions during all the occupied hours in summer, i.e. 100 % of occupation time. The calculations are made according method B of annex F, EN 15251, adopting category I limits of the adaptive comfort model. This value decreases to 93 % for scenario C3. In Table 3, scenario C2, the best in terms of thermal comfort and the one with the second best energy performance (after C1), is compared to existing building performance. Energy reduction up to 86 % in terms of primary energy from pre- to post-retrofit are evaluated.

Table 3: energy consumption per net floor area of the pre and post-retrofit building

Building condition	Energy carrier	Delivered energy kWh/(m²a)	Primary energy conversion factor	Primary energy kWh/(m²a)
Existing building (pre-retrofit)	Fuel (Natural gas)[1]	202.07	1.00	202.07
	Electricity[2]	35.32	2.18	77.00
Post retrofit (scenario C2)	District heating[1]	36.82	0.8	29.46
	Electricity[2]	4.35	2.18	9.48

(1) Heating and production of domestic hot water
(2) Lighting, laundry, kitchen, equipment, mechanical ventilation

5 DISCUSSION

The scenario that shows the lowest energy consumption is C1, whereas the one that shows the best comfort conditions is C2. Since C2 also shows a quite considerable energy saving, it was chosen as the best scenario according to design aims. However, each scenario has pros and cons, which may be summarised as follows.

A: the envelope retrofit based on prefabricated modules including mechanical ventilation and solar shading, substantially reduces the primary energy up to 82 % compared to the existing building. Yet this scenario is not in line with the main goal of the retrofit that is to achieve indoor thermal comfort with no active cooling system.

B1: this scenario shows nighttime natural ventilation coupled with daytime active cooling. The results show 60 % reduction of delivered energy for cooling compared to the reference scenario A. This is a clear evidence of nighttime ventilation cooling effectiveness.

B2: in this scenario the hybrid mechanical logic is applied (Figure 3). The natural ventilation system is used as the primary means of ventilation and if this is not applicable, the mechanical system activates. The results show 22 % reduction of delivered energy for cooling compared to scenario B1.

B3: as a further step, in this scenario mechanical night ventilation is simulated, i.e. the mechanical system that is designed to provide minimum ventilation rates during the occupied time, works also at nighttime on full capacity. This results in 63 % higher electrical energy use compared to scenario B1. It is moreover less effectiveness then scenario B2, when only nighttime natural ventilation is simulated.

C1: the scenario includes daytime mechanical ventilation, nighttime natural ventilation and no active cooling system. This solution provides the lowest primary energy requirements and 100 % of occupied hours within category I limits according to the adaptive thermal comfort model of standard EN 15251. Adopting another indicator of thermal discomfort, in room 5 only 21 % of occupied hours show indoor temperatures higher than 24 °C, that is the set-point temperature for active cooling.

C2: this scenario adds the possibility of daytime natural ventilation to C1, and this reduces the percentage of occupied hours with indoor temperature higher than 24 °C to 15.1 %.

C3: this scenario shows that mechanical ventilation at nighttime does not provides the same thermal comfort level as natural ventilation, mostly because the airflow rate is fixed and dimensioned according to the minimum airflow requirements for daytime. Moreover, scenario C3 shows higher primary energy requirements compared to C1 and C2.

6 CONCLUSIONS

Seven different ventilation scenarios were simulated in order to evaluate which one could better fit the comprehensive deep energy retrofit of a kindergarten. The existing building showed very poor ventilation conditions during the occupied time whenever the average daily outdoor temperature dropped below 14 °C. It means that occupants (teachers) operates windows as function of outdoor temperature more than as function of perceived air quality.

All the ventilation strategies thus include mechanical ventilation during the heating season, whereas attention was made on different options for mechanical cooling and mechanical/hybrid ventilation.

Scenario C2, with no active cooling, mechanical ventilation at daytime and hybrid ventilation at nighttime (either natural or mechanical), showed to be the best scenario according to thermal comfort conditions and the second best in term of primary energy.

Natural ventilation at nighttime results extremely effective in improving daytime thermal comfort conditions during the cooling season, showing that active cooling could be avoided if the weather conditions are not too much severe, and if the building is adequately operated (ventilation, lighting, solar screen). Nighttime natural ventilation performs better than mechanical ventilation, in the simulated building, reducing furthermore the final energy use.

7 ACKNOWLEDGEMENTS

The authors would like to thank the Department of School Construction and the Office for EU Affairs of the Municipality of Milano, ROCKWOOL Italia and Askeen for the collaboration on this project. The study was developed within the EU-GUGLE project funded by Seventh Framework Programme under grant agreement n. 314632.

8 REFERENCES

ANSI/ASHRAE. (2013). *Ventilation for Acceptable Indoor Air Quality.* ASHRAE (ASHARE Standard 62.1).

Breesch, H., Bossaer,A., Janssens, A. (2005). Passive cooling in low-energy office building. *Solar Energy,* 79 (6), 682-696.

Brelih, N. (2012). Ventilation rates and IAQ in national regulations. *REHVA. Eur. HVAC. J.,* 1, 24–28.

Causone, F. (2015). Climatic potential for natural ventilation. *Architectural Science Review,* in press.

CEN. (2007). Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics. Bruxelles: Comité Européen de Normalisation (Standard EN 15251).

Crawley, DB., Lawrie LK., Winkelmann, FC., Buhl, WF., Huang, YJ., Pedersen, CO., Strand, RK., Liesen, RJ., Fisher, DE., Witte, MJ., Glazer, J. (2001). EnergyPlus: Creating a new-generation building energy simulation program. *Energy Buildings*, 33, 319-331.

CRESME. (2014). *RIUSO03: Ristrutturazione edilizia riqualificazione energetica rigenerazione urbana – estratto della ricerca CRESME.*

Daisey, J., Angell, W.J. and Apte, M.G. (2003). Indoor air quality, ventilation and health symptoms in schools: an analysis of existing information. *Indoor Air,* 13, 53–64.

Danish Building Regulations. (2010). *The Danish Ministry of Economic and Business Affairs.* Copenhagen: Danish Building Regulations.

Dimitroulopoulou, C. (2012). Ventilation in European dwellings: A review. *Build. Environ.,* 47, 109–125.

Fabbri K. (2013). Thermal comfort evaluation in kindergarten: PMV and PPD measurement through datalogger and questionnaire. *Building and Environment,* 68, 202-214.

Hasegawa, A., Goto, T., Yoshino, H., Iino, Y. and Muramatsu, S. (2012). Regulations and standards for indoor environments (IAQ) in schools – an international review. *In: Proceedings of Healthy Buildings 2012.* 8-12 July 2012, Brisbane, Queensland.

Legambiente. (2014). *Ecosisistema scuola: XV rapporto di Legambiente sulla qualità dell'edilizia scolastica, delle strutture e dei servizi.*

Santamouris, M., Synnefa, A., Asssimakopoulos, M., Livada, I., Pavlou, K., Papaglastra, M., Gaitani, N., Kolokotsa, D. and Assimakopoulos, V. (2008). Experimental investigation of the air flow and indoor carbon dioxide concentration in classrooms with intermittent natural ventilation. *Energy Build.*, 40, 1833–1843.

UNI. (1995). *Air-conditioning systems for thermal comfort in buildings- General, classification and requirements.*UNI (UNI 10339).

Wargocki, P., Da Silva, N.A.F. (2015). Use of visual CO2 feedback as a retrofit solution for improving classroom air quality. *Indoor Air,* 25, 105–114.

World Health Organization (regional office for Europe). (2011). *Methods for monitoring indoor air quality in schools, Report of a meeting.* Bonn: Germany 4-5 April 2011.

Wyon, D., Wargocki, P., Toftum, J. and Clausen, G. (2010). Classroom ventilation must be improved for better health and learning. *REHVA. Eur. HVAC. J.,* 3, 12–16.

PCMS AS A TOOL FOR INCREASING THERMAL INERTIA IN BUILDINGS

Lidia Navarro[1], Alvaro de Gracia[2], Luisa F. Cabeza[1*]

[1] *GREA Innovació Concurrent, Universitat de Lleida, Edifici CREA, Pere de Cabrera s/n, 25001, Lleida, Spain. Tel: +34.973.00.35.77. Email: lcabeza@diei.udl.cat*

[2] *CELiMIN, Universidad de Antofagasta, Campus Coloso, Av. Universidad de Antofagasta, 02800 Antofagasta, Chile*

1 INTRODUCTION

It is well known, that energy consumed by the HVAC systems in buildings represents an important part of the global energy consumed in Europe (Directive 2010/31/EU). Latent heat storage has been widely studied (Cabeza et al. 2011, Zhou et al. 2012) for its potential in many applications for building energy management (Lim et al. 2014). Passive implementation of phase change materials (PCM) in buildings has demonstrated significant energy reduction of HVAC systems, but with some limitations (Castell et al. 2010). For this reason, active implementation of PCM in buildings has high potential. In this paper two innovative active systems are presented, consisting of thermal energy storage units embedded inside the two different parts of building components. A double skin facade and an internal slab were filled with PCM in order to act as a storage unit and a heating and cooling supply.

2 EXPERIMENTAL SET-UP

In the experimental set-up located in Puigverd de Lleida (Spain) several house-like cubicles were built to study different constructive systems and materials. Three of these cubicles are used to test the PCM active systems; one of them has a double skin facade with PCM, another one has an active slab as internal separation with PCM, and the third one has conventional constructive system acting as a reference. Both technologies presented in this paper are designed to cover the cooling and heating demand of a building. A structural component of the building is used as a storage unit with an active charge and discharge process for covering the energy demand of the building. The novelty of the system is the inclusion of phase change materials (PCM) inside the storage unit in order to increase the heat storage capacity.

3 OPERATING PRINCIPLE

3.1 Double skin facade with PCM

The ventilated facade acts as a solar collector during daytime in winter season (Figure 1, left). Once the PCM is melted and the solar energy is needed by the heating demand, the heat discharge period starts. The openings drive the air flowing from indoor to the facade cavity, where it is heated up by the PCM panels and sent it back into the cubicle. On the summer mode (Figure 1, right) the PCM is solidified by the outside air which is pumped into the channel during night time. The air is cooled down by the PCM and is pumped to the inner environment providing cooling supply. The air flow from outdoors to outdoors prevents the overheating effect in the air channel when there is no more cooling supply available. Moreover, night ventilation mode could be also performed to achieve a free cooling effect.

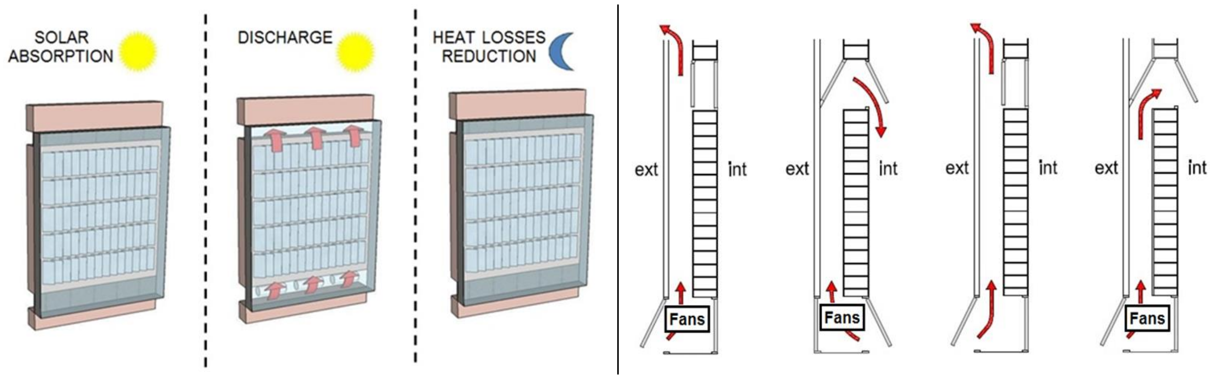


Figure 1: Operating principle double skin facade; left winter, right summer.

3.2 Active slab with PCM

In winter season, the active slab (Figure 2a) is charged during daytime through the injection of hot air from the solar air collector. The thermal energy is stored inside the slab until a heating demand is needed. The air of the internal ambient is pumped through the hollows of the slab and the heat exchange with the PCM provides the heat needed to cover the demand totally or partially. On the other hand, operational mode during the summer period (Figure 2b) outside cool temperature at night is used to cool down the concrete slab and to solidify the PCM by the circulation of outdoors air through the hollows. During daytime, the inner air of the cubicle is pumped through the slab getting a cooling supply and covering part or the whole cooling load. The operational schedule of both winter and summer periods are driven by a control system.

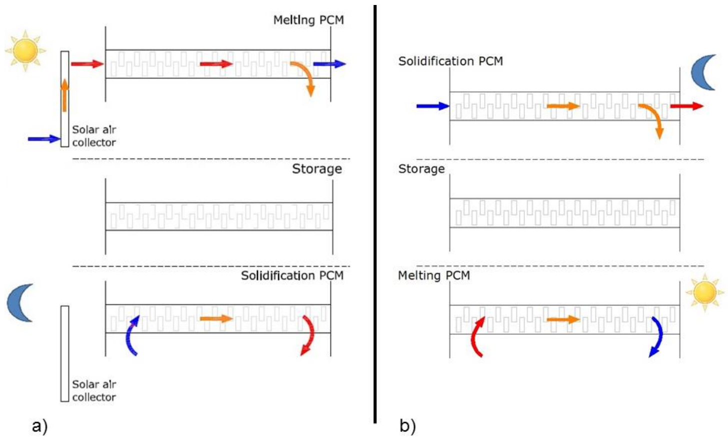


Figure 2: Operating principle active slab (a) winter and (b) summer.

4 RESULTS AND CONCLUSIONS

The use of the ventilated facade with PCM reduces the electrical energy consumption of the installed HVAC systems even without using mechanical ventilation in the air cavity (De Gracia et al 2013). The active slab system registered also significant energy savings that highlight its potential.

During the summer tests, the high potential of the night free cooling effect was demonstrated in both systems for reducing the cooling loads. In both cases, double skin facade (De Gracia et al 2013) and active slab, the use of fans must be optimized to reduce the overall electrical energy consumption. The cold storage capacity of the systems is very sensitive to the outer night temperature, being limited under severe summer conditions. In the case of the double skin facade the system prevents successfully the overheating effect that could be found in the air channel.

5 ACKNOWLEDGEMENTS

The work partially funded by the Spanish government (ENE2011-28269-C03-01 and ULLE10-4E-1305). The authors would like to thank the Catalan Government for the quality accreditation given to their research group (2014 SGR 123) and the city hall of Puigverd de Lleida. The research leading to these results has received funding from the European Union's Seventh Framework Programme (FP7/2007-2013) under grant agreement n° PIRSES-GA-2013-610692 (INNOSTORAGE). Alvaro de Gracia would like to thank Education Ministry of Chile for Grant PMI ANT1201.

6 REFERENCES

Cabeza, LF, Castell, A, Barreneche, C, de Gracia, A, Fernández, AI (2011). Materials used as PCM in thermal energy storage in buildings: A review. *Renewable and Sustainable Energy Reviews*, 15, 1675-95.

Castell, A, Martorell, I, Medrano, M, Perez, G, Cabeza, LF (2010). Experimental study of using PCM in brick constructive solutions for passive cooling. *Energy and Buildings*, 42, 534-540.

De Gracia, A, Navarro, L, Castell, A, Ruiz-Pardo, A, Alvárez, S, Cabeza, LF (2013). Experimental study of a ventilated facade with PCM during winter period. *Energy and Buildings*, 58, 324-332.

De Gracia, A, Navarro, L, Castell, A, Ruiz-Pardo, A, Alvárez, S, Cabeza, LF (2013). Thermal analysis of a ventilated facade with PCM for cooling applications. *Energy and Buildings*, 65, 508-515.

Directive 2010/31/EU of the European parliament and of the council of 19 May 2010 on the energy performance of buildings. Available from: http://www.epbd-ca.eu.

Lim, J-H, Song, JH, Song, SY (2014). Development of operational guidelines for thermally activated building system according to heating and cooling load characteristics. *Applied Energy*, 126, 123-135.

Zhou, D, Zhao, CY, Tian, Y (2012). Review on thermal energy storage with phase change materials (PCMs) in building applications. *Applied Energy*, 92, 593-605.

SHIFTING THE IEQ PARADIGM FROM COMFORT SILOS TO HOLISTIC HEALTH AND PERFORMANCE

William P. Bahnfleth, PhD, PE

The Pennsylvania State University
104 Engineering Unit A
University Park, PA 16801 USA
wbahnfleth@psu.edu

ABSTRACT

Indoor environmental quality (IEQ) is generally taken to encompass four main factors: indoor air quality (IAQ), thermal conditions, visual quality, and acoustical quality. Although there is an implicit concern for safety, the predominant metrics all four in standards for design of buildings are based on perceived quality or comfort. For example, ASHRAE Standard 62.1-2013 *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 2013a) strives to provide indoor air "in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction." ASHRAE Standard 55-2013 *Thermal Environmental Conditions for Human Occupancy* (ASHRAE 2013b) defines thermal comfort as "that condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation." Additionally, both the scientific and standards writing communities treat the main IEQ factors independently and lack metrics for overall quality. Within these areas, a large body of research has established criteria for ventilation, thermal conditions, light, and sound that are widely accepted and utilized.

An alternative view of IEQ is that it should be defined in a way that more directly addresses health and productivity rather than the subject perception of comfort. In other words, measures like differences in healthcare costs, absenteeism from work, quality of learning in school, and rate of production take the place, or are added to considerations of satisfaction. Research to establish connections between performance and IEQ have, to date, focused primarily on the effects of thermal conditions and air quality while little has been done to extend this approach to lighting and acoustics. Air quality studies predominantly consider the effects of varying ventilation rate or of interventions such as particulate filters or other air cleaners. Studies have examined data under laboratory and field conditions and have used simulation to predict various outcomes. Monetized benefit: cost ratios of improved IEQ

predicted by these studies are as large as 10:1 and in some cases can be achieved with no additional energy cost or with a net reduction in energy use, for example, through the use of outside air economizer controls where feasible. These benefits are not purely economic, but also human, resulting from predicted reductions in morbidity and mortality. A more limited body of work has considered interactions among IEQ factors, again primarily IAQ and thermal environment, but this has not resulted in a widely accepted approach to integrating them.

While widely discussed and debated the impact of this body of work on policy, standards, and practice, has been relatively small and generally limited to credits in high performance building rating programs such as the US Green Building Council's *Leadership in Energy and Environmental Design* program (USGBC 2014. A promising recent development is the establishment by the US General Services Administration (GSA), which establishes design standards and criteria for government buildings, of a three-tiered system in its PBS-P100 *Facilities Standards for the Public Buildings Service* (GSA 2015) for high performance buildings that includes specifications for enhanced particulate filtration of outside and supply air and, at the highest level, ultraviolet germicidal irradiation of cooling coils. However, in the private sector, minimum standards based on acceptability criteria continue to predominate and IEQ is specified piecemeal without consideration for the relative importance of the various factors. The most significant advances in indoor air quality in recent times have been source removal and control actions directed at asbestos, radon, indoor smoking, and carbon monoxide. Much remains to be done.

The slow, essentially organic, penetration of research findings on the costs and benefits of IEQ into building standards and policies stands in sharp contrast to aggressive programs for energy conservation, energy supply, and atmospheric protection that have been implemented worldwide by nations independently and as a result of international agreements. The ability to mobilize significant resources to address energy and environmental issues can be credited to an effective effort to explain the significance of research findings to professionals and the public, the ability to express the impacts of energy consumption and environmental damage in concrete and fiscal terms, and the consequent mobilization of political forces. Where IEQ is concerned, this effort has been less coordinated and less effective – but the opportunity exists for this community to take a positive example from the successes of others.

To this end, an alliance of international organizations, the Indoor Environmental Quality Global Alliance (IEQ-GA 2015) has been initiated with the goal of improving the actual, delivered indoor environmental quality in buildings through coordination, education, outreach, and advocacy. Founding members include organizations that broadly deal with indoor environments as well as those specialized primarily in IAQ. Discussions with potential member organizations representing lighting and acoustics are in progress. Although in its infancy, IEQ-GA has already begun work to collect and critique IEQ standards and is organizing and presenting programs at the conferences of member organizations and others. Such an organization can help to create the public and professional awareness and provide the advocacy needed to influence research funding and public policy by speaking with one voice across national boundaries. History clearly suggests that such an organization is needed and the key question at this point is whether the IEQ-GA will succeed in meeting that need. Regardless, organizations that believe in the need for a better approach to IEQ that must recognize the need for a more concerted effort on multiple fronts – research, education, standards, policy – to achieve their goals.

KEYWORDS

Indoor Environmental Quality, Standards, Health, Productivity

REFERENCES

ASHRAE (2013a). ANSI/ASHRAE Standard 62.1-2013. *Ventilation for Acceptable Indoor Air Quality*. Atlanta:ASHRAE.

ASHRAE (2013b). ANSI/ASHRAE Standard 62.1-2013. *Ventilation for Acceptable Indoor Air Quality*. Atlanta:ASHRAE.

USGBC (2015) *LEED* web site. http://www.usgbc.org/leed accessed July 10 2015.

GSA (2015). PBS-P100 *Facilities Standards for the Public Building Service*. Washington DC:US General Services Administration.

IEQ-GA (2015). *IEQ Global Alliance* web site. http://ieq-ga.net/ accessed on 10 July 2015.

Event sponsors

The conference organizers are grateful to the following organizations and networks for their support to this event:

ISBN : 2-930471-45-X
EAN: 9782930471457