ESCUELA TÉCNICA SUPERIOR DE INGENIEROS INDUSTRIALES Y DE TELECOMUNICACIÓN

Titulación:

INGENIERO INDUSTRIAL

Título del proyecto:

Design of a Test Bench for Static Calibration of Torque Transducers

Iñaki Garro Moreno

Pablo Sanchis Gúrpide (UPNA)

Jean-Marc Timmermans (VUB)

Pamplona, 16 de Junio de 2011

Abstract

Torque, speed and power are the defining mechanical variables associated with the functional performance of rotating machinery. Normally the power is determined by combination of the measures of torque and rotational speed, therefore the relevance of torque measurement.

Torque transducers are used to measure the torque of rotatory machines. They are produced with a determined accuracy, but over time they lose accuracy and it is necessary to re-calibrate them. There are different ways to calibrate torque transducers, such as use another torque transducer with a better accuracy class or use a lever deadweight bench, which is a more accurate method.

The current document collects the design of a “Test bench for static calibration of torque transducers” at the Department of Electrical Engineering and Energy Technology (ETEC) from Vrije Universiteit of Brussel (VUB). This project was made as final master thesis project to obtain the degree of Industrial Engineer at the Universidad Pública de Navarra (UPNA) during an Erasmus scholarship with Erasmus Hogeschool Brussel (EHB).

This report collects the design process of the different parts of the bench, the different analysis performed with Autodesk Inventor Stress Analysis and the calculation of the test bench accuracy. Also software was performed in Microsoft Excel to calculate automatically the metrological properties of the torque transducers from the values obtained during the calibration tests.

Resumen

El par motor, la velocidad de rotación y la potencia son las variables mecánicas asociadas con el desempeño funcional de la maquinaria de rotación. Normalmente la potencia se determina mediante la combinación de las medidas de par y velocidad, de ahí la importancia de la medición del par motor.

Los transductores de par se utilizan para medir el par motor en máquinas rotatorias. Se fabrican con una precisión determinada, pero con el tiempo van perdiéndola y es necesario volver a calibrarlos. Hay diferentes maneras de calibrar los transductores de par, como el uso de otro transductor con una clase de precisión superior o utilizar un sistema de palanca con pesos calibrados.

El presente documento recoge el diseño de un "Banco de pruebas para la calibración estática de transductores de par" en el Departamento de Ingeniería Eléctrica y Tecnología Energética (ETEC) de la Vrije Universitait Brussels (VUB). Este proyecto fue realizado como proyecto final de carrera para obtener el título de Ingeniero Industrial en la Universidad Pública de Navarra (UPNA), durante la estancia de su autor en Bruseals participando en una beca Erasmus con la universidad Erasmus Hogeschool Brussels (EHB).

Este informe recoge el proceso de diseño de las diferentes partes del banco, además de los diferentes análisis realizados con la herramienta de Análisis de Tensiones de Autodesk Inventor y el cálculo de la precisión de banco de pruebas. También se realizó un programa en Microsoft Excel para calcular automáticamente las propiedades metrológicas de los transductores de par a partir de los valores obtenidos durante las pruebas de calibración.

Acknowledgements

I would like to thank Dr. Jean-Marc, Prof. Philippe Lataire, Patrick Vanroose and Francis for the guidance and all the help provided. Without their invaluable support it would have been difficult to me to conclusion this thesis.

List of symbols

b	Reproducibility [mV/V^2]	T	Temperature [ºC]
b'	Repeatability [mV/V^2]	T_a	Applied torque [Nm]
d	Diameter [m]	T_b	Balancing error torque [Nm]
E	Buoyancy force [N]	T_f	Friction torque [Nm]
E	Elastic modulus [N/m^2]	T_h	Hanger's balancing torque [Nm]
F	Force [N]	U	Expanded standard uncertainty
f_a	Deviation of indication from the fitting curve [mV/V^2]	u	Standard uncertainty
f_o	Residual value at zero torque [mV/V^2]	v	Poisson's coefficient
f_q	Deviation of indication of the torque measuring device with defined scale [Nm]	*Vol.*	Volume [m^3]
g	Gravity [m/s^2]	W	Relative expanded uncertainty
G	Shear modulus [N/m^2]	w	Relative uncertainty
γ_m	Surface strain	X	Indicated value at torque step with increasing torque [mV/V^2]
h	Reversibility [mV/V^2]	X_a	Indicated value calculated from the interpolation equation [mV/V^2]
I	Indication of torque measuring device at torque step with increasing torque [mV/V^2]	$\bar{X}$	Mean value of the torque measuring [mV/V^2]
I	Inertia [m^4]	α	Linear expansion coefficient [ºC^{-1}]
K	Strain gauge's sensibility	α_c	Angular acceleration [rad/s^2]
L	Length [m]	ε	Unitary deformation [m/m]
m	Mass [kg]	ρ	Density [Kg/m^3]
M_A	Minimum torque value of the measuring range [Nm]	σ	Normal stress [N/m^2]
P	Load [N]	τ_{ij}	Shear stress [N/m^2]
R	Electric resistance [Ω]	τ_m	Stress [N/m^2]
r	Resolution of the indicating device [Nm]	ϕ	Twist angle [º]
S	Sensibility [(mV/V)/Nm]	θ	Level angle [º]

List of Figures

List of Tables

Table of contents

1. Project goals

The goal of this master thesis is to design a test bench for static calibration of torque transducers, capable to calibrate the different torque transducers existing in the Department of Electrical Engineering and Energy Technology (ETEC) from the Vrije Universiteit of Brussel.

We have to apply a known torque value to the transducer, measure its output response and determine its metrological properties in order to calibrate the transducer. For this purpose we have to calculate the uncertainty of the torque we are applying during the calibration test to the torque transducer. Also we have to automate as possible the analysis of the output response measures making a program to calculate the metrological properties.

We should use for this purpose as many of the available items in the laboratory as possible, trying to find new applications for the disused material existing in the laboratory. Other parts of the machine may have to be designed. We should use as possible the resources available in the university, such as material, machining machines and qualified professionals, to produce those parts of the machine which need to be designed. Also we have to check if those parts are commercially available at a reasonable price, and whether it's better to buy rather than make them.

2. Background

This is an innovative project as there is no information available at the university about how to calibrate the torque transducers or how to build the machine in question. However, this is a technology commercially available and well-studied by the torque transducer manufacturers and metrological laboratories, which usually do the calibration of these measurement devices.

2.1 Location

The "Bench for static calibration of torque transducers" will be placed in the laboratory of the Department of Electrical Engineering and Energy Technology (ETEC) from Vrije Universiteit Brussel (VUB), located in Etterbeek (Brussels).

2.2 Torque transducers

A transducer is a device that converts one type of energy to another. The conversion can be to/from electrical, electro-mechanical, electromagnetic, photonic, photovoltaic, or any other form of energy. In the case of torque transducers, they convert an input electrical signal to an output electrical signal whose frequency or voltage is proportional to the applied torque.

Torque, speed and power are the defining mechanical variables associated with the functional performance of rotating machinery. Normally the power is determined by combination of the measures of torque and rotational speed:

$$\text{Power (W)} = \text{Torque (Nm)} \cdot \text{Rotational speed (rad}\cdot\text{s}^{-1}\text{)}$$

Accuracy of power measurement is generally limited by the torque measurement (±0.05% to ±1%) since rotational speed can be measured with almost any desired accuracy. Torque errors can arise from the application of extraneous torques from hose and cable connections, from wind-age of external parts, and from miscalibration of the torque transducer.

Our purpose is to calibrate the torque transducers existing in the lab in order to have an accurate measure of the torque transmitted by the electric motors we have in the laboratory, and from there be able to calculate the power. But first of all it's important to have a good knowledge of the device we are going to calibrate. For this reason we did a study of how they work, what kind of mechanic and electric accessories they need to work and the meaning of their basic specifications.

In "Annex I: Torque transducers", we have an extended explanation of how a torque transducer operates. In "Annex II: Torque transducer characteristics", we show the definition of the typical specification terms for torque transducers, and in "Annex III - Torque transducer metrological properties calculation" we show the formulas used to calculate its metrological properties.

Torque transducers to calibrate

We have to build a versatile machine that could calibrate the different transducers existing in the VUB's lab. So firstly we need to know the different types of torque transducers we have and their characteristics. We made a study of the different types of transducers, their rated torque, dimensions and their quantity. As we saw, there are several types of transducers with different dimensions, couplings, rated torques and accuracies.

Fig. 2.1- TT Torque transducer

In Table 2.1 we can find the data collected in our study. Features like the accuracy class, length and couplings diameter were collected from the specification sheets available in the laboratory.

Torque transducer	Rated torque (Nm)	Quantity	Accuracy class	Length (mm)	Flange couplings diameter (mm)
Vibro-Meter TT 111	1.000	1	0,2	270	190
Vibro-Meter TT 110	500	2	0,2	240	170
Vibro-Meter TM 113/M1	500	2	0,1	275	170
Vibro-Meter TT 109	200	5	0,2	240	170
Vibro-Meter TT 108	100	1	0,2	220	95
Vibro-Meter TT 107	50	1	0,2	220	95

Table 2.1 - Torque transducers at ETEC lab in VUB [1]

As we see in Table 2.1 the majority of the torque transducers are for 500Nm and 200Nm (nine from twelve torque transducers). We will design the calibration test bench firstly for 500Nm and 200Nm torque transducers. We may use it later for the rest of them adding some adapters.

Mechanic accessories:

All the torque transducers are coupled with the driven and the driving device via flexible couplings in order to protect the torque transducers from any damage. In the ETEC laboratory KTR Rigiflex flexible couplings are used to connect the flange ended torque transducers with the shaft of the devices. All of them are assembled in suspended installation.

Electrical accessories:

Vibro-Meter TT torque transducers have a rated output of $\pm 5V_{DC}$ that can be measured using a DC voltage multimeter. The supply voltage is $24\pm 3V_{DC}$, so they don't need signal conditioning.

Vibro-meter TM torque transducers have a $70mV_{eff\ AC}$ (20 kHz) sinusoidal output signal, so they need a previous amplification stage before measuring. The supply is a sine wave, so they also need signal conditioning equipment. The amplification and the supply are made by the ISC 228 signal conditioner, which has an output for the rated torque of $\pm 5V_{DC}$. After that conditioning stage the output can be measured using a multimeter. TM torque transducers also can measure speed. The ISC 228 is connected with the torque transducer via EH 135 cable assembly. It needs a power supply of 115/230 V_{AC} (50 Hz).

In Fig. 2.2 we have a scheme of the TM torque transducers connexion with the ISC signal conditioner.

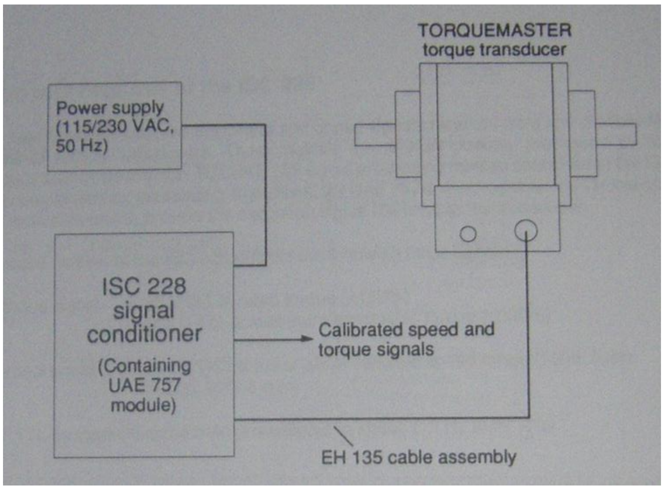


Fig. 2.2 - TM torque measuring system using ISC 228

3. Torque transducers calibration

In this chapter we are going to introduce how the torque transducers are calibrated. Firstly we can differentiate two basic ways to do the calibration: static and dynamic. We are going to design a static calibration bench, so we will explain more deeply the static calibration methods.

3.1 Static / Dynamic calibration

Standard calibration methods for torque transducers are static in a double sense. For one, torque is measured only in a state when it is constant with respect to time. For another, the torque transducer is mounted in a calibration machine in a non-rotating setup. This is a contrast to the conditions in the real application of most industrial torque transducers. Typical industrial application for torque measurement is in power test stands for engines, gearboxes and transmissions in research and development in the automotive industry, in which the operating conditions are both rotating and dynamic.

Researches of dynamic torque measurement lead to the conclusion that the dynamic performance of torque transducers under periodic or impact load may differ considerably from the result of the corresponding static calibration. The effect of a torque transducer rotating during measurement results from the centrifugal forces. It has actually two distinct components. The first one being the influence of rotation on the zero signal and the second one being the influence on the sensitivity. The influence of rotation on the zero signal can be examined by a so-called spin test. In such a test, the applied torque is zero Investigating the effect on the sensitivity is the more difficult task. In this study, a cradle-mounted dynamometer and a rotating torque transducer are implemented in the same shaft train of a test stand in order to determine the potential influence of rotation speed on torque measurement with a rotating torque transducer. [2]

3.2 Calibration methods

3.2.1 Lever deadweight

This is a static calibration method that consists of a set of calibrated masses which act on a calibrated lever arm. The system can be directly applied to the transducer in the case of an unsupported calibration beam. In the ideal case the lever arm will be connected to a bearing to eliminate bending from the weights and masses and to minimize friction. This type of system is used for static torque calibration and offers the best uncertainties. [3]

Fig. 3.1- Lever deadweight calibration [4]

3.2.2 Reference torque transducer

With this method torque can be applied using a motor or hydraulics, and the torque is controlled by a calibrated reference torque transducer in a feedback loop. This type of system can be used to measure static torque but has the additional advantage of being suitable for continuous torque calibration, whereby the applied torque is applied over a much shorter time. The disadvantage of these systems is that the uncertainty of applied torque will be much higher because the system is dependent on the prior calibration of the reference.

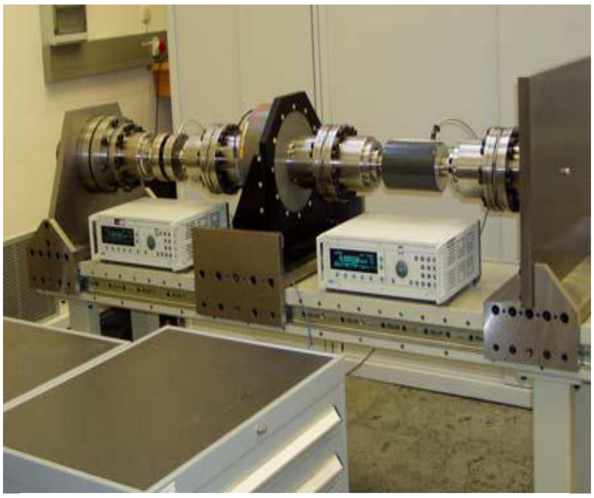


Fig. 3.2 – Reference torque transducer calibration method

Regardless of the torque facility to be used, it is important to evaluate the uncertainty of the system. This should include contributions from all influencing parameters (e.g. mass, length, alignment, and environmental factors). [3]

3.3 Supported / Unsupported lever deadweight beams

We are going to design a lever deadweight system due to its best accuracy and because there is no need of another torque transducer with a better accuracy to do the calibration. That's why we are going to study it in greater depth.

Lever deadweight calibration methods can be divided in two main groups depending if the transducer is directly coupled with the lever arm or not: unsupported beams and supported beams respectively.

3.3.1 Unsupported beams

Calibration beams coupled directly to a test transducer are used in industry for reasons of cost and simplicity, and because this is the way in which the transducer will be used in many applications. Unsupported beams are flexible and easy to use, and in many instances replicate the way a transducer is subsequently used. They are well suited for the calibration of the many forms of torque measuring device found in general use in industry up to 1.5 kNm capacity.

Any bending and sensitivity effects should be allowed for within the calibration uncertainty budget. With these factors taken into consideration, it is possible to perform fit-for-purpose calibrations, to high levels of accuracy over the whole of a transducer's range. The geometry of the transducer is an important factor to consider when using unsupported beams. In Fig. 3.3 we can see an unsupported lever deadweight beam construction. It is composed by the lever arm, the weights, the torque transducer (1) and some adapters (2).

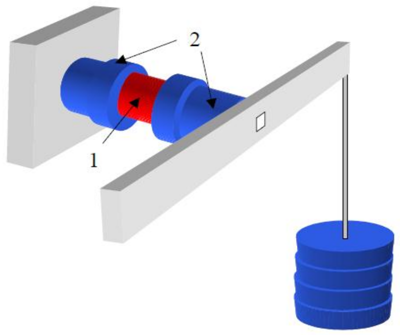


Fig. 3.3 - Unsupported lever deadweight beam [4]

In practice the transducer will be subject to additional bending and parasitic forces and torques, which may affect the transducer to varying degrees. It should always be the aim to keep these additional forces to a minimum. When using unsupported calibration beams bending forces are unavoidable. In this case it is important to be aware of the influence that bending can have on the measurement result and, where possible, to be able to quantify this. [4]

In order to determine the ***Bending effect*** the following test is carried out for a symmetrical unsupported beam:

- Calibrate the torque transducer using the basic calibration method (see chapter 4 – Calibration process).
- At each torque step apply an additional 50% of the load on each side of the beam, so that 150% of the applied load is in the direction of the torque is to be applied and 50% of the load is in the opposite direction. Record the deflection. Remove the additional load and then apply the next increasing torque.
- Calculate the maximum difference between the single loading deflections (d_{single}) and the double loading deflections (d_{double}) at each increasing calibration torque across the measurement series at different orientations, expressing this as a percentage of the single load deflection.
- The bending effect should be taken as the maximum of these differences. [4]

$$Max\left[\frac{d_{double}-d_{single}}{d_{single}}x100\right]$$

Eq. 3.1 [4]

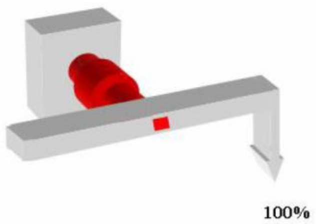


Fig. 3.5 - Single loading [4]

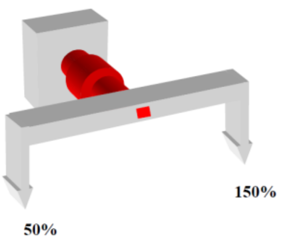


Fig. 3.4 - Double loading [4]

3.3.2 Supported beams

Supported beams refer to calibration beams that are coupled with the transducer via bearings. The bearings transmit the vertical force produced by the weights to the ground in order to minimize the bending moment induced to the transducer.

Flexible couplings are needed to protect the transducer from impact forces. Usually there will be some misalignment due to the mismatch of the two axes that will give a radial, angular or axial misalignment, or any combination of the three. They induce parasite bending moments to the torque transducer that may affect the torque measure. One way to minimize this is using flexible coupling elements. [4]

In Fig. 3.6 we see a typical supported lever deadweight beam construction. It is composed by the torque transducer (1), adapters (2), flexible couplings (3) and bearings (4).

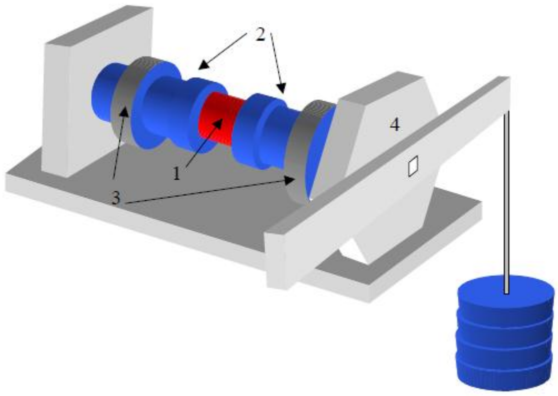


Fig. 3.6 - Supported lever deadweight beam [4]

Bearings are often used to eliminate bending effects due to the weight of the lever beam and weights. However, depending on its quality the bearing itself may introduce errors such as friction and the performance of the bearing still requires evaluation. Bearing supported beams are used for measurements of the highest accuracy. [4]

To conclude this chapter, we have studied the different methods to calibrate torque transducers with their advantages and disadvantages. We decided to build a supported lever deadweight bench because it's the most accurate method to do the calibration, and it simplifies the calibration test because there is no need to take into account the bending effect. The only disadvantage is that the assembly is more complicated and requires more parts, but we have some of these parts currently in the lab. We have the flexible couplings and the bearings, which are the principal parts required.

4. Calibration process

The calibration test can be carried out for clockwise and/or anti-clockwise torque. The calibration of torque transducers should be carried out as a static procedure by measuring discrete approximately equally spaced torque values, which is typical of calibration facilities with lever-mass systems. [5]

Now we are going to describe the different steps we have to follow in order to calibrate torque transducers. All this information is based in a document redacted by the EA (European co-operation for Accreditation), the "EA Guidelines on the Calibration of Static Torque Measuring Devices" (EA-10/14 June 2000). This document was redacted to improve the harmonisation in determining the calibration results and uncertainties in torque measurements.

4.1 Description and identification of the torque measuring device

The torque measuring device consists in the complete set of measuring instruments and other equipment assembled to carry out torque measurements. All components of the torque measuring device (including cables for electrical connection) shall be individually and uniquely identified (for example by the manufacturer's name, the type, four or six conductor circuit or similar, and the serial number).

For the torque transducer, the maximum working torque and the measuring end of the transducer should be indicated. The torque transducer and any associated mechanical coupling should be designed and assembled such that both clockwise and anti-clockwise torque can be applied without the significant influence of non-torsional forces, such as bending moments [5].

4.2 Overloading test

It is recommended that prior to the first calibration, the torque transducer, including its mechanical couplings, is subjected to two overload tests in the course of which the nominal torque is exceeded by 8% to 12% of the nominal torque and this value is maintained for 1 to 1,5 minutes. This should exclude unexpected failure of the torque transducer during application of the calibration load, for example by fracture, resulting in consequential damage to the calibration facility. [5]

4.3 Setup

Indicating device - The readout unit should be chosen carefully so that there is adequate resolution over the whole of the torque range of interest. It should be adjusted according to the manufacturer's instructions and in accordance with the customer's specifications. Prior to the calibration, it is recommended that the indicating device is subject to a check to ensure it functions correctly and will not invalidate the calibration. All adjustments and corresponding setting values should be recorded before and after the calibration.

Temperature stabilization - The equipment should be allowed to thermally stabilize for at least one hour before measurements begin. Calibration should be carried out at a temperature stable to ±1°C.

Transducer zero signal - Prior to the installation of the transducer into the calibration equipment, the zero signal of the mechanically unloaded torque transducer should be measured in a specified (vertical) position and recorded.

Mounting of transducer - Failure to apply the calibration torque at the shaft end position stated by the manufacturer, or specified by the customer, may lead to erroneous measurements. The mounting position should be identified. The transducer should be held as rigidly as possible on the mounting plate. The fit of the squares should be as good as possible to minimize any slack.

Alignment - Practically it is very difficult to perfectly align a transducer in a torque machine. Usually there will be some misalignment due to the mismatch of the two axes that will give a radial, angular or axial misalignment, or any combination of the three. One way to minimize this is through the use of flexible coupling elements. [5]

4.4 Preload

After the installation into de calibration equipment, the torque transducer should be preloaded three times in the direction to be calibrated, applying the rated torque, in order to delete the "memory" of the torque transducer. The duration of the preload application should be approximately 30 seconds and the indicator reading should be recorded.

> NOTE: The stability of the zero signal may provide an indication of the performance of the device during its calibration.

If the transducer is disturbed or moved to a new mounting position, a further preload is required. [5]

4.5 Mounting position

The torque transducer should preferably be calibrated in three different mounting positions with the transducer or its mechanical coupling part rotated each time through 120° about the measurement axis. Four relative mounting positions can be used for a square drive.

Two incremental calibration series are required at the same mounting position, normally at the start of calibration, for determination of repeatability. [5]

4.6 Range of calibration

The recommended number of calibration steps should be a minimum of 5 approximately equally spaced from 20% to 100 % of the rated torque. For the calculation of a fitting curve, a minimum of 5 steps must be taken. For more accuracy than 0,1% we should take 8 steps equally spaced, usually 10%, 20%, 30%, 40%, 50%, 60%, 80% and 100% of rated torque. For transducers with accuracy of 1% or less, calibration with 3 steps for 0º and 90° may be sufficient. When calibration points below 20% of M_E (rated torque) are required, calibration steps of 10%, 5% and 2% of M_E should be used. [5; 4]

4.7 Loading conditions

The time interval between two successive calibration steps should, if possible, be similar. Recording the measured value may take place only after the indication has stabilized. Indication drift due to creep requires that the time sequence be maintained.

Calibration should be carried out at a temperature stable to +/-1°C. This temperature should be in the range between 18°C and 28°C (preferably between 20°C and 22°C) and recorded. [5]

4.8 Indicated value

The indicated value is defined as the difference between an indication in loaded condition and an indication in unloaded condition. The indication at the beginning of each measurement series should be zeroed, or taken into account by computation during the evaluation following the measurement.

NOTE: Recording of non-zeroed values provides additional information about the zero signal behaviour.

For torque measuring devices with defined scale (indication in the unit of torque), the indication should be zeroed at the beginning of each measurement series. [5]

4.9 Test process

1. Description and identification of the torque measuring device.
2. Overloading test (recommendable).
3. Set up.
4. Record the zero torque output. The leveling gearbox may need to be removed from the transducer to get a true zero.
5. Preload the transducer and record the torque output.
6. Apply at least 5 equally spaced series of torque, increasing torque from zero torque up to the rated torque. The time interval between two successive calibration steps should, if possible, be similar.
7. At each torque step level the beam using the leveling gearbox, and record the output after a period of at least 30 seconds, when the indication has stabilized.
8. If decreasing torques are to be applied, then apply decreasing series back down to zero torque.
9. Record the final zero torque output. May be necessary to remove the leveling gearbox from the transducer.
10. The transducer should be disconnected and rotated in a clockwise direction. After each rotation the transducer should be preloaded once.
11. Repeat the steps so that the transducer is calibrated symmetrically (3 orientations for flange and shaft style transducers, 4 orientations for square drive transducers). [5]

Once all the output values are collected we have to calculate the metrological properties of the torque transducer, such as sensibility, repeatability, reproducibility, hysteresis or linearity. All these terms are defined in “Annex II: Torque transducer characteristics”, and the formulas used to calculate them are defined in “Annex III - Torque transducer metrological properties calculation". Then we can classify the torque transducers attending to some requirements that we will explain in the following chapter.

In Fig. 4.1 we have some examples of calibration sequences for different accuracy classes and torque transducers. *U* is the expanded uncertainty of measurement.

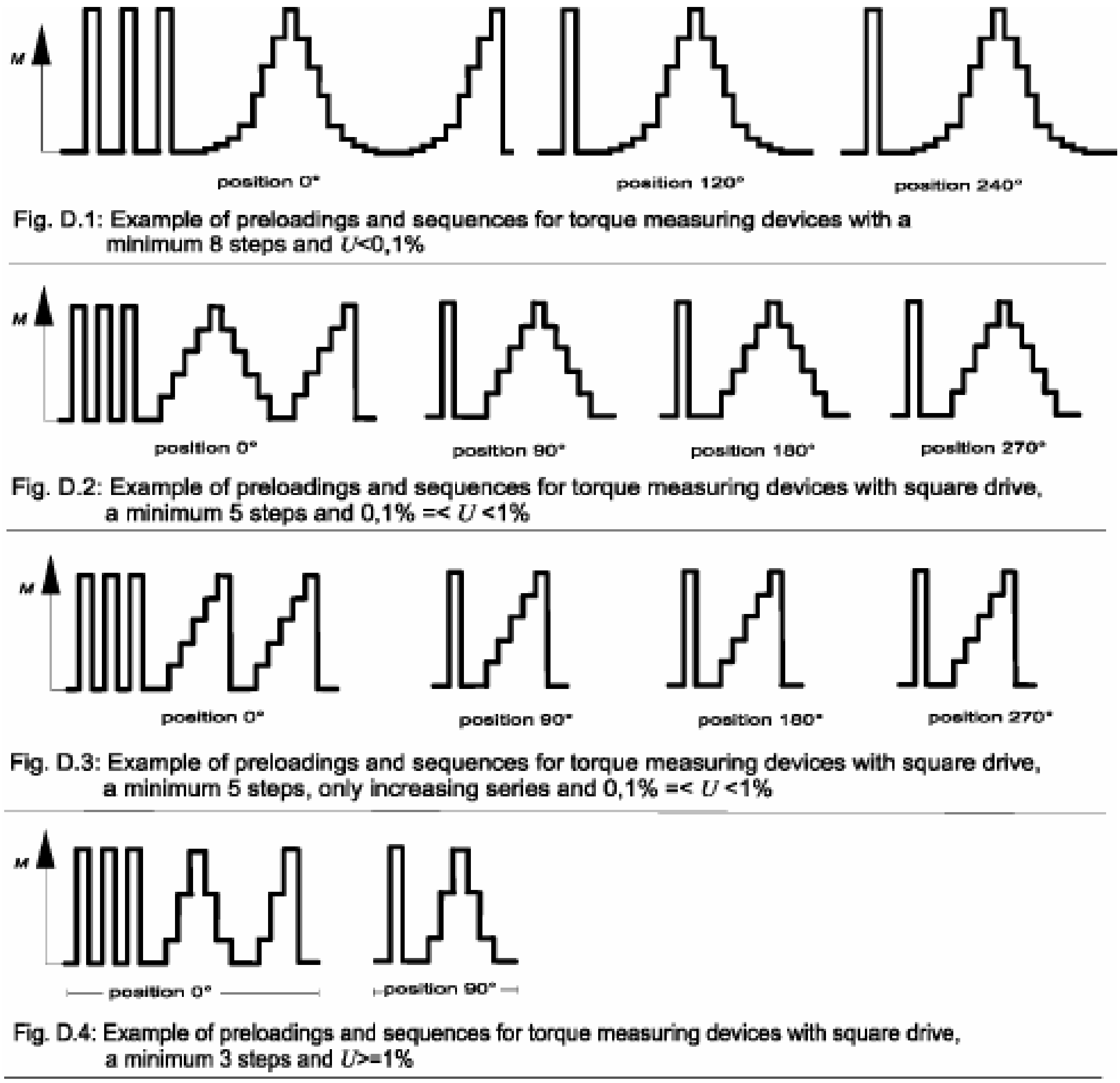


Fig. D.1: Example of preloadings and sequences for torque measuring devices with a minimum 8 steps and *U*<0,1%

Fig. D.2: Example of preloadings and sequences for torque measuring devices with square drive, a minimum 5 steps and 0,1% =< *U* <1%

Fig. D.3: Example of preloadings and sequences for torque measuring devices with square drive, a minimum 5 steps, only increasing series and 0,1% =< *U* <1%

Fig. D.4: Example of preloadings and sequences for torque measuring devices with square drive, a minimum 3 steps and *U*>=1%

Fig. 4.1 – Examples of calibration sequences. [5]

If the transducer needs adjusting we have to refer to appropriate operator's handbook or service manual.

4.10 Classification of the torque measuring device

Principle of classification

The range for which the torque measuring device is classified shall be determined by considering each calibration torque one after the other, starting with the maximum torque and decreasing from it to the minimum torque. The classification range ceases at the last torque for which the classification requirements in the following paragraphs are satisfied. [5]

Classification criteria

For the classification, the minimum value of the measuring range (M_A) shall be:

- 20 % of M_E (Rated torque), alternatively.
- 40 % of M_E for the clases 0,05 and 0,1.

For the instruments classified for interpolation, the following criteria shall be taken into consideration:

- Relative repeatability in unchanged mounting position.
- Relative reproducibility in different mounting positions.
- Relative deviation of indication or of fitting curve.
- Relative residual value at zero torque.
- Relative reversibility when increasing and decreasing torque is applied.
- Resolution of the indicating device by at the minimum value of the measuring range (M_A).

Table 4.1 states the values of these classification parameters for the different torque transducer classes. (*r*: resolution of the indicating device). [5]

class	Maximum permissible error of the torque measuring device in %					Min. value of torque M_A	Calibration torque M_k
	Relative repeat-ability $\frac{b'}{X}$	Relative reprodu-cibility $\frac{b}{X}$	Relative residual value at zero torque $\frac{f_0}{X_E}$	Relative reversibility $\frac{h}{X}$	Relative dev. of indication or of fitting curve $\frac{f_q}{X}$, $\frac{f_a}{X}$		Expanded rel. uncertainty of measurement in % $W_{tcm} = k \cdot w_{tcm}$
0,05	0,025	0,050	0,0125	0,063	±0,025	≥4000 r	0,010
0,1	0,05	0,10	0,025	0,125	±0,05	≥2000 r	0,020
0,2	0,10	0,20	0,050	0,250	±0,10	≥1000 r	0,040
0,5	0,25	0,50	0,125	0,63	±0,25	≥400 r	0,10

Table 4.1 - Classification of torque measuring devices [5]

4.11 Calibration certificate

Once the calibration of a torque measuring device has satisfied all the requirements, the calibration laboratory should draw up a certificate stating the following information:

i. Identify all the elements of the torque measuring device and its components, including mechanical coupling components to the calibration equipment.
ii. The method used, identifying whether clockwise and/or anti-clockwise torque, incremental and/or decremental.
iii. The resolution of the torque measuring device.
iv. The temperature at which the calibration was performed.
v. A statement on the expanded uncertainty of measurement and the equation of the fitting curve where applicable.
vi. Where required, a statement regarding the conformity of the calibration results to a particular classification at the criteria is used, see Table 4.1. [5]

4.12 Torque transducers re-calibration

If there is a miscalibration of the torque transducer we have to re-calibrate it following the instructions of the manufacturer. The following paragraphs contain the procedures to re-calibrate the different torque transducers we have in ETEC laboratory. They have been obtained from their specifications sheets available in the lab.

TM torque transducers re-calibration:

The procedure below requires that adjustments be made with the lid of the ISC 228 housing remove and the unit still powered up.

THE FOLLOWING SECURITY MEASURES SHOULD BE TAKEN WHEN WORKING ON THE SYSTEM WHILE IT IS POWERED UP:

-WEAR INSULATING SHOES.

-WEAR INSULATING GLOVES.

-WORK WITH TOOLS EQUIPED WITH AN INSULATED HANDLE.

-WORK ON AN INSULATING MAT HAVING A MINIMUN THICKNESS OF 20 MM.

THE OPERATOR OR THIRD PARTIES MAY BE EXPOSED TO THE RISK OF SEVERE INJURY OR EVEN DEATH BY ELECTROCUTION IF THESE MEASURES ARE IGNORED.

Proceed as follows:

- Connect the TORQUEMASTER unit to a static calibration test bench
- Check the EH 135 cable connections between the TM and the ISC 228, switch on the ISC 228 and wait approximately 30 minutes for the unit to stabilize.
- Check the specifications of your TM transducer to find the rated (nominal) torque. Place the necessary mass on the positive calibration arm in order to simulate the transducer's rated torque.
- Remove the cover of the ISC 228 so as to gain access to its UAE 757 module.
- Connect a voltmeter to the TORQUE OUTPUT line (connector P4, terminal 10). Use connector P4, terminal 8 as the 0V reference.
- Adjust potentiometer P5 on the UAE 757 until you measure exactly 5 V_{DC} ±0,1% on the voltmeter.
- Remove the mass from the calibration arm and check that 0 V_{DC} is measured on the voltmeter. If necessary, correct the zero using the ZERO ADJUST trimmer (P10) on the front of the panel of the UAE 757 module.
- Place the mass on the other calibration arm and check that you measure -5 V_{DC} ±0,1%.
- Remove the mass and recheck that 0 V_{DC} is measured. *[1]*

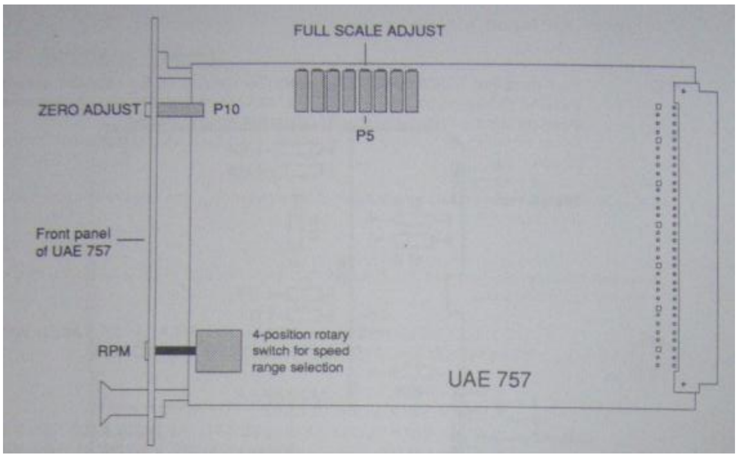


Fig. 4.2 - Positions of trimmers on the UAE 757 module [1]

Note: To obtain a real zero torque output may be necessary to remove the levelling gearbox. Also we can't apply exactly the rated torque. We have to calculate the transmitted torque and its associated output. This is taken into account in the calibration test program.

TT torque transducers re-calibration

Proceed as follows:

- Connect the torque transducer unit to a static calibration test bench.
- Set the balance without load (zero).
- Connect a precision resistor decade ($2\ k\Omega \leq R_x \leq 3{,}5\ k\Omega$). $R_x = (R_{31} + R_{32})$
- Connect an oscilloscope and an AC-Voltmeter to TP 2.
- Connect 2 precision resistor decades to R_{17} and R_{26} or R_{27}, values corresponding to existing resistors.
- Adjust the signal on the oscilloscope (TP2) to a minimum using resistance decades R_{17}, $R_{26/27}$.
- Solder fixed resistors in place of the decades (Note: only recheck when the resistors are cold).
- Connect a DC-voltmeter to the output (pin 9).
- With a resistor decade on R_{60} (offset) set the output voltage at zero, solder a fixed resistor for R_{60} and re-check.
- Load the balance with nominal torque (Clockwise).
- Find Rx value so that DC-voltmeter shows +5V_{DC} ±0,2% for class 0,2.
- Solder in R31 and R32.
- Load the balance with nominal value (Counter-clockwise).
- Check that the DC-voltmeter shows -5V_{DC}±0,2%.
- Take off the load and check the zero.
- Earth the pin "G" so that the test relay is activated.
- The value of the test resistors is found with a resistor bridge on $R_y = (R_{20} + R_{21})$.
- The DC-voltmeter must show 5 V_{DC} on the output.
- Check the zero and the sensitivity once again. [1]

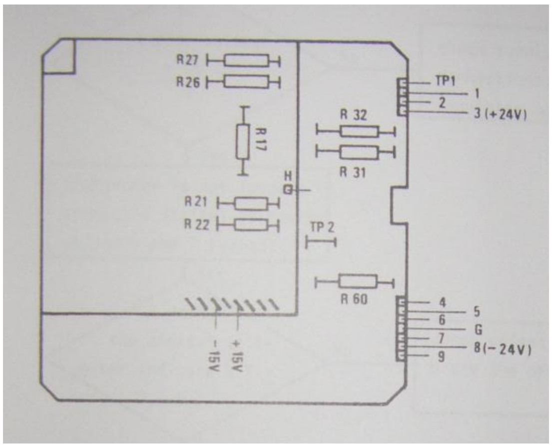


Fig. 4.3 - TT torque transducer without the housing [1]

5. State of art

Once we know what we are going to build and how to carry out the torque transducer calibration test, we are going to realize a study of the commercially available calibration devices in order to know how this technology is nowadays.

5.1 Unsupported beams

- ***Norbar Radius Beam***

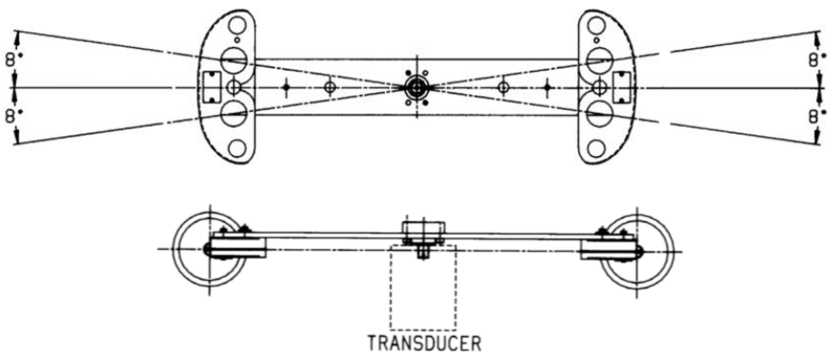


Fig. 5.1 – Norbar Radius Beam. [4]

Features:

- Beam length machined to +/- 0.01% (100 microns per metre) allowing for wire thickness.
- Clockwise and counter-clockwise operation.
- Radiused ends maintain length over +/- 8 degrees of rotation from horizontal.
- No bearings to cause energy loss during loading.
- Beams balanced to maximise energy transfer to transducer during loading.
- Radiused ends offset to bring plane of loading within transducer and so reducing bending moments.
- High beam accuracy allows use of cast iron weights rather than stainless steel. Weight accuracy is required to be equal to or better than 0.01% which approximates to class M1.
- Manufactured from aerospace alloys. Should be used in a temperature controlled environment of 20 degrees C +/- 2 degrees. Outside these limits temperature compensation is needed.
- Additionally the beams are designed to apply load on vertical plane which cuts through the square drive inside the transducer. This minimises bending movements on the transducer and also for safe operation ensures that the beam will not fall out. [4]

5.2 Supported beams

- ***Norbar supported beam 500-5.000 Nm / 500-5.000 ftm***

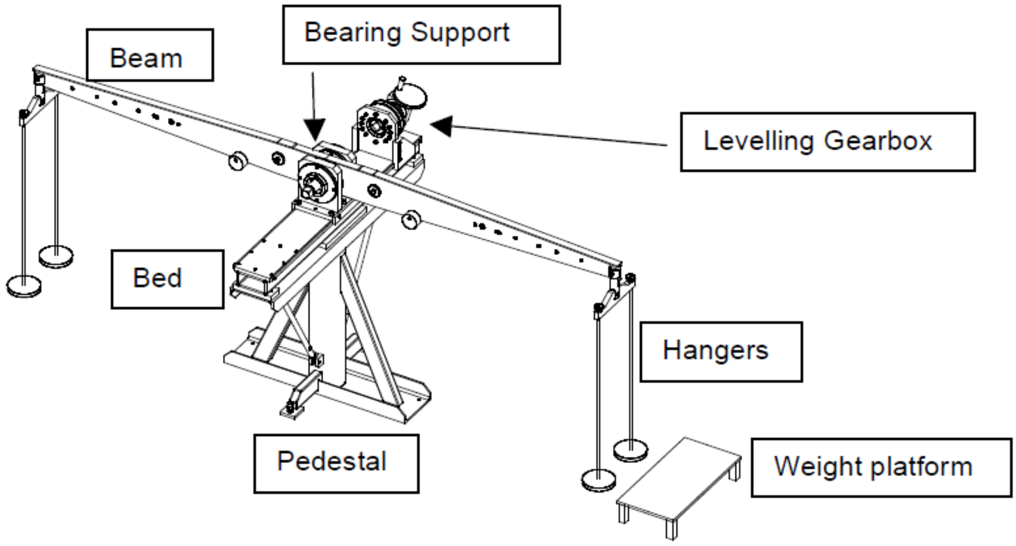


Fig. 5.2 - Norbar supported beam. [4]

Features:

- A self-locking gearbox provides an easy method of keeping the beam horizontal once loaded.
- The beam and its associate stand have been robustly constructed to ensure only torque is applied to the transducer under test. Any torque losses (through bending or bearing loss) seen by the transducer have been minimised.
- Weight accuracy is required to be equal to or better than 0.01% which approximates to class M1
- Location holes accuracy: +/- 0.01% of nominal.
- The symmetrical beams allow for both clockwise and counter-clockwise calibration.
- The calibration beam should be used in a temperature controlled environment of 20°C +/- 2°C. Outside these limits, we must apply the temperature compensation.
- The beam has many location holes for the hangers. The required location depends on the torque required and the weights used. [4]

Sushma torque calibration bench

Fig. 5.3 - Sushma supported beam [6]

Features:

- Supplied with stainless steel Newton weights of F2 class of accuracy. Weights are adjusted either to standard ‘g’ value or customer’s local ‘g’ value (if the value is provided by the customer). Minimum force that can be measured is 1,2 or 5% of rated capacity depending on the range.
- Newton weights of different denominations and quantity to cover the entire range.
- Arm fixed with plates to take care of cosine error during calibration. (optional)
- Spirit level for levelling the lever arm.
- Gear box with rotating handle to mount and align the torque sensors with the arm assembly.
- Levelling screws are provided for the machine.
- Ranges available: 0-2 Nm, 0-20 Nm, 0-200 Nm, 0-2 kNm, 0-20 kNm.
- Measurement uncertainty: 0,05%. [6]

6. Key points for the design

The main point for the design of the calibration test bench is to be able to know the transmitted torque accuracy. In this chapter we are going to analyse the different variables that affect the applied torque.

Torque consists of one force acting at a distance from the rotation centre:

$$Torque = Force \cdot Length.$$

In our case, the force is the effective load made by the weights, which is affected by the Gravitational effect and the Buoyancy effect. The effective length is affected by the temperature due to the linear expansion, and by the level angle of the lever arm.

Also we have to take into consideration the torque losses due to friction because we are going to use a supported beam with bearings. Bending effects are minimized using supported beam method and the misalignment effect can be minimized using flexible couplings.

6.1 Gravitational effect

Weights force depends directly from the gravity value, as known Load = Mass · Gravity. The gravity value contrary to common belief is not constant; it varies depending on the location of the considered point. Because of that it's important that the local gravity value for the laboratory is established. The effect of not doing this could be a variation in the force produced by the weight of perhaps 0,5% of reading.

It is therefore strongly recommended to establish the local value of gravity (g) for your laboratory and use weights that have been calibrated at that gravitational constant.

- The ideal solution is to have the gravity measured on site by the national geological survey agency.
- The second best solution is to ask the national agency for a figure calculated from gravitational contour maps.
- The third approach is to calculate the value from knowledge of the latitude, and height above sea level in meters. This will give an uncertainty of approximately +/- 0,005% but will not reflect local differences due to rock structure etc. Based [4]

In "Annex IV - Local gravity" we explain deeply the gravity effect and we calculate the local gravity value we are going to use for our calculations: $g = 9{,}81130\ {}^{m}/_{s^2}$ with an expanded uncertainty $Ug_{95\%} = \pm 0{,}000002\ {}^{m}/_{s^2}$.

6.2 Buoyancy effect

Lever deadweight calibration uses calibrated weights to generate a downwards force. This means that Archimedes principle applies, ie. Air pressure under the weights causes an upwards force. This effect reduces the effective force generated by the weights and therefore the mass must be increased to allow for this. Under standard conditions (Air density 1.2kg/m^3 at 20 degrees centigrade and working in conventional mass terms) the increase required is by a factor of around 0.015%. [4]

We are going to calculate the contribution of this effect to the applied load in order to take it into account for the calibration test calculations. First, we calculate the force *P* made by the weights:

$$P = g \cdot \rho_{weights} \cdot Vol$$

Eq. 6.1

The buoyancy force *E*, which has opposite direction from the weights force, is calculated as follows:

$$E = g \cdot \rho_{air} \cdot Vol$$

Eq. 6.2

Then we can calculate the effective force *F* and the buoyancy effect in percentage $\frac{\Delta P}{P}\%$:

$$F = P - E = g \cdot Vol \cdot (\rho_{weights} - \rho_{air})$$

Eq. 6.3

$$\frac{\Delta P}{P}\% = \frac{P - F}{P} \cdot 100 = \frac{\rho_{air}}{\rho_{weights}} \cdot 100$$

Eq. 6.4

The following equations show the buoyancy effect for weights made by steel and cast iron, which are the most common materials used.

$$\frac{\Delta P}{P}\%\ \text{steel} = \frac{\rho_{air}}{\rho_{steel}} \cdot 100 = 0{,}015\% \qquad \rho_{steel} = 7.850\ Kg/m^3$$

Eq. 6.5

$$\frac{\Delta P}{P}\%\ \text{cast iron} = \frac{\rho_{air}}{\rho_{cast\ iron}} \cdot 100 = 0{,}017\% \qquad \rho_{cast\ iron} = 7.250\ Kg/m^3$$

Eq. 6.6

Weights that are calibrated to standard procedures do not have this factor taken into account because the air buoyancy affects both sides of the mass balance and can be ignored. It is important that weights used for torque transducer calibration are adjusted for air buoyancy.

It should also be noted that the double ended beam design means that each half of the beam is balanced with regard to buoyancy of the beam. This is a significant advantage over single-arm counterbalanced systems. Based [4]

6.3 Temperature effect

The dimensions of most substances increase when they are heated at constant pressure. This phenomenon is properly called thermal expansion. When the rise in the temperature *ΔT* of a solid body is not too large, it is found experimentally that the fractional increase of the length $\frac{\Delta L}{L}$ is directly proportional to the rise in temperature.

$$\frac{\Delta L}{L} = constant \cdot \Delta T$$

Eq. 6.7

The constant of proportionality is characteristic for each material, and it's called *linear expansion coefficient, α*. [7]

$$\frac{\Delta L}{L} = \alpha \cdot \Delta T$$

Eq. 6.8 - Linear expansion equation [7]

As we have seen, the effective length from which the force is applied will vary with the temperature. Most of the manufacturers, such as Norbar, recommend using the Calibration Bench in a temperature controlled environment of 20 ºC +/- 2ºC. Keeping these limits, there is no need for temperature compensation. If the beam must be used outside these limits, the temperature must be stable (with 1ºC change per hour) and the effective length of the beam has to be calculated according Eq. 6.9. [4]

$$L = L_o \cdot [1 + \ \alpha \cdot (T - 20)]$$

Eq. 6.9 - Temperature compensation [4]

Note: A thermometer will be needed for the calibration test.

6.4 Level angle effect

The level angle is the angle between the lever arm and the horizontal axis. The effective length of the lever arm varies with the cosine of the level angle as we see in Eq. 6.10. We have to study how this effect varies the applied torque. The level angle effect can be minimized using the level gearbox in supported beams or using lever arm radiused ends for unsupported beams.

$$L = L_o \cdot \cos\theta$$

Eq. 6.10 – Level angle effect.

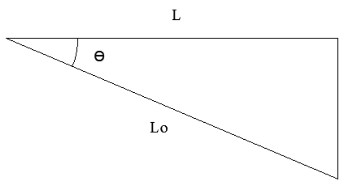


Fig. 6.1 - Level angle effect graphic.

We will use the level gearbox to rotate the lever arm and minimize the level angle effect. To measure the level angle we will use a level spirit located in the middle of the lever arm.

Note: a level spirit will be needed for the calibration test

6.5 Friction torque

As we will use bearings to support the beam, we have to consider the friction torque losses. The calibration test is done in static conditions, so the shaft doesn't have rotatory speed. Due to the applied torque the shaft twists (otherwise the torque transducer couldn't measure the torque), so there is a relative movement between the parts of the roller bearings. A relative movement means that the starting torque of the bearings has been exceeded by the applied torque, which means torque losses.

Therefore we have to consider the starting torque of the roller bearings for our calculations, which has the opposite direction of the applied torque.

All these effects will be studied deeply in chapter 10.2 – "Uncertainty of the transmitted torque calculation".

7. Design

We are going to design a supported lever deadweight bench, which is composed by the transducer to calibrate (1), flexible couplings to protect the torque transducer from damage and to reduce the misalignment effects (3), adapters that may be necessary to connect the different types of torque transducers to the test bench (2), bearings to reduce the friction torque losses (4), the lever arm and several weights to produce the different input torque steps.

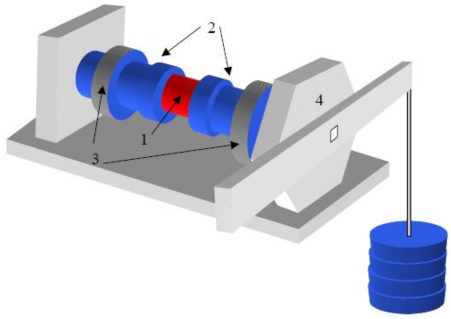


Fig. 7.1 - Unsupported lever deadweight beam [4]

We decided to build firstly the calibration test bench for 500 Nm and 200Nm torque transducers, which are the majority. We may use it later to calibrate the rest of them adding some adapters. We will use for the design the program *Autodesk Inventor Professional* in order to simulate the different parts assembly and to perform the stress analysis. See chapter 9.2 – “Autodesk Inventor Professional”.

7.1 Available material

Some of the parts we need to build the bench currently exist in the lab, so we have to adapt the designed parts to the existing ones in order to minimize the number of parts to make. In the following table we show the different parts we can use for the design.

Flexible couplings

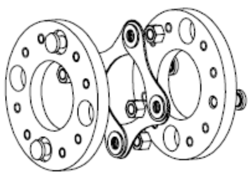


Fig. 7.2 – KTR Rigiflex type 02 [8]

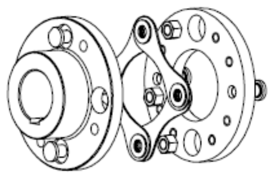


Fig. 7.3 - KTR Rigiflex type 01 [8]

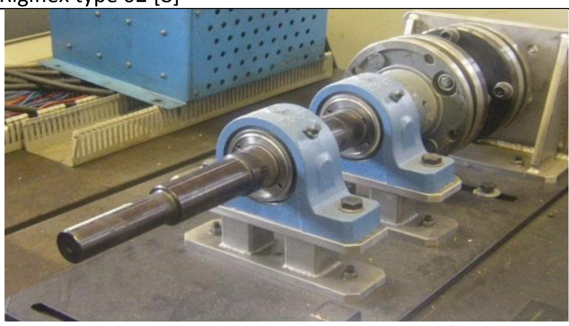
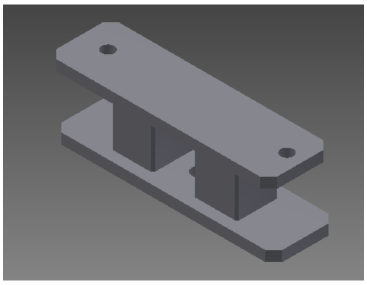
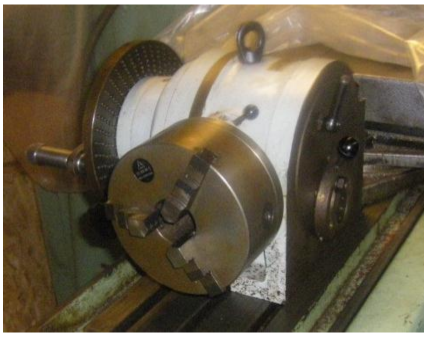


Fig. 7.4 - Available material at ETEC laboratory

Bearings  Fig. 7.5 – SKF SYT 45 LTS with 22209 rolling bearings [9]	**Shaft**  Fig. 7.6 - Shaft from ETEC lab
Bearing bases  Fig. 7.7 - Bearing bases	**Weights**  Fig. 7.8 - Weights
Rotatory table  Fig. 7.9 - Rotatory table	**Universal divisor**  Fig. 7.10 - Universal divisor
Clamping table of ETEC laboratory  Fig. 7.11 - Clamping table	**Clampling table of a milling machine**  Fig. 7.12 - Clamping table from a milling machine

In Table 7.1 we show the different flexible couplings we have for each type of torque transducer and their dimensions.

Flexible coupling	Torque transducers (Nm)	Shaft end diameter (mm)	Flange end diameter (mm)	Type 01 length (mm)	Type 02 length (mm)
KTR Rigiflex 28	50, 100	28	95	78,5	29,5
KTR Rigiflex 55	200, 500	55	170	87	40,5
KTR Rigiflex 65	1000	65	210	96	42,5

Table 7.1 - Flexible couplings [8]

We measured the different components in order to use them for the 3-D design program and the stress analyis simulations.

7.2 Lever arm design

The lever arm is the most important part for the design. It has to resist the torque produced by the masses hanged at a certain distance from the centre of the arm and the masses' weight. As there are different torque transducers to calibrate, the lever arm will be subjected to different solicitations.

The lever arm will be symmetrically double ended in order to minimize the buoyancy effect and to be able to make clockwise and anti-clockwise calibrations. Considering the space available in the lab and trying to reduce the quantity of masses needed we are going to make a 2m length lever arm, 1m length each side. We will hang the masses from different distances from the centre: 1m for 1000 Nm, 500 Nm and 100Nm torque transducers, 0,5m for 50 Nm torque transducers, and 0,4m for 200 Nm torque transducers.

Using this system we need three sets of weights instead of five:

1. 1.000N weight set for 1.000Nm (1.000N · 1m = 1.000Nm).
2. 500N weight set for 500Nm (500N · 1m = 500Nm) and 200Nm (500N · 0,4m = 200 Nm).
3. 100N weight set for 100Nm (100N · 1m = 100Nm) and 50Nm (100Nm · 0,5m = 50Nm).

For the design we are going to consider the worst case, which is a 1.000Nm rated torque. This torque is achieved by a total mass of 1.000N hanged at 1m distance from the centre of the lever arm. One side of the lever arm will be subjected to that solicitations and the opposite side will be free. The arm can be loaded in clockwise or anti-clockwise direction for the calibration, depending if we load the right or the left end of the arm.

First design of the lever arm

Considering the solicitations of the lever arm, which are greater in the centre than in the ends, and trying to reduce the lever arm's weight, which determines the friction torque of the bearings, we will try to design the lever arm with a greater profile section in the centre than in the ends. We made a first design in steel and another in alluminium in order to determine which will be the better material to use taking into account the material cost, the machining cost and the functional performance.

Firt design - Steel lever arm:

The lever arm is composed by four parts, the spirit and the three wings, made from a steel sheet of 5mm thick. The different parts are welded to form a single T-profile beam. It has a total weight of 10,7 Kg. The arm is thought to be assembled with the shaft via flange coupling. For this purpose six holes have been performed in the centre of the lever arm. Also it has holes at 1m, 0,5m and 0,4m from the centre for the weight hangers.

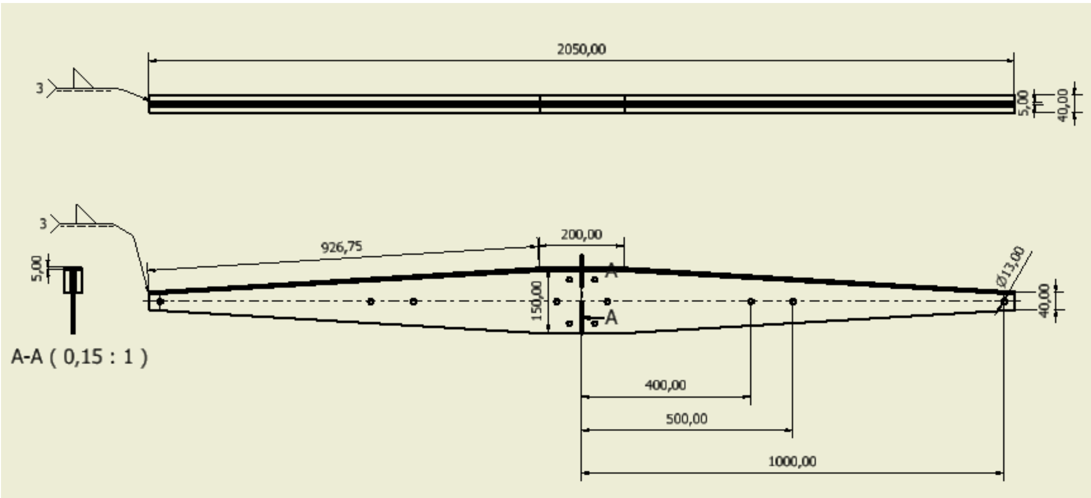


Fig. 7.13 - 1st design. Steel lever arm design

First design – Aluminium lever arm:

The aluminium lever arm is single piece from aluminium 7075-O sheet of 15mm thickness and it has several machined holes in order to reduce the weight. It has a total weight of 7 Kg. The arm is thought to be assembled with the shaft via flange coupling. For this purpose holes have been performed in the centre of the lever arm. Also it has holes at 1m, 0,5m and 0,4m from the centre for the weight hangers.

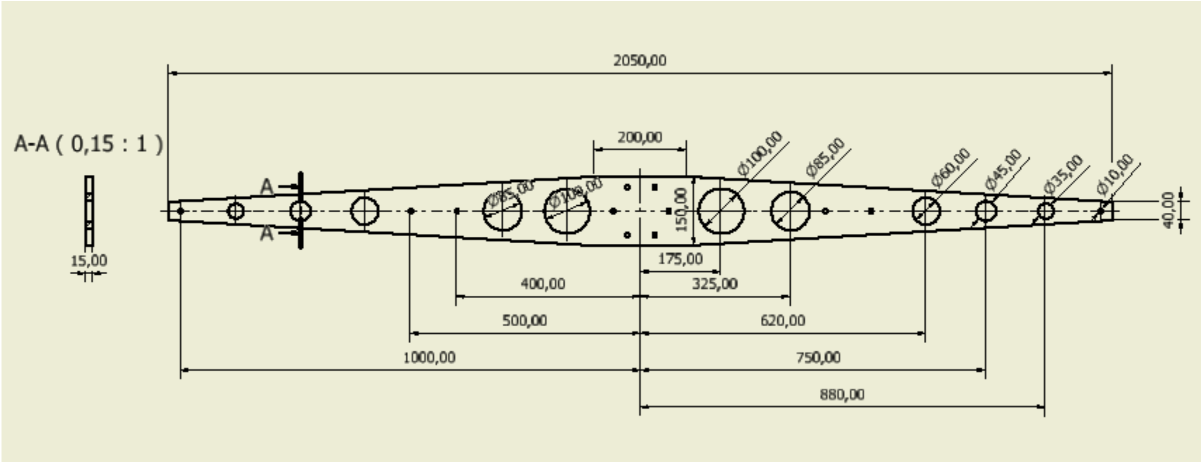


Fig. 7.14 - 1st design. Aluminium lever arm design

First design analysis

We could see from the stress analysis results that the aluminium has a better performance with less weight than the steel. The aluminium lever arm doesn't require welding, so the machining costs are smaller than in steel, but there is a big difference between both material costs, aluminium is more expensive than steel.

We have also to take into account the corrosion effect if we want a prolonged live of the lever arm. Aluminium generates its own protection against corrosion, but for steel we need to use protection means against corrosion, such as painting, surface treatment or stainless steel. We need a high precision for the location of the holes, therefore we can't use painting. A good solution is to use stainless steel; the inconvenient is that it's more expensive than aluminium.

Once the design of the lever arm was done we got in communication with the mechanic department of the V.U.B. in order to know if it could be made with our equipment and resources. They informed us that a piece of that length couldn't be made, the maximum length that could be machined is 60cm and our design has 2m length.

After that first design we thought we could design the arm with a standard rectangular profile, which only requires the machining of some holes. We have rectangular steel profiles of different dimensions available in the laboratory. Then maybe we could adapt any of the drilling machines to make those holes. Also the method to attach the lever arm to the shaft has to be improved because the torque transducer has to be rotated each 120º or 90º from the lever arm during the calibration test. We need an easier way to rotate it without using screws, which represent a great waste of time for the assembly and disassembly.

Second design of the lever arm

Taking into account these considerations we made a second design using stainless steel and aluminium standard profiles. The arm will be centred with the shaft by a central hole and the torque will be transmitted to the shaft via clevis pins through both holes each side of the central hole. It has also several lightening holes near the ends and several balancing canals near the center. It's equipped with holes to accommodate the hangers at different distances from the centre: 0,4m, 0,5m and 1m.

Fig. 7.15 – Lever arm second design

Second design – Stainless steel lever arm:

The lever arm is single piece made from a standard rectangular stainless steel profile 20mm x 60mm of 2.060mm length. It has a total weight of 13 Kg.

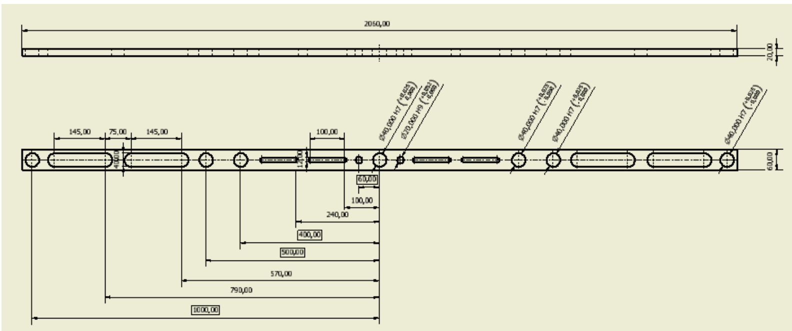


Fig. 7.16 - 2nd design. Stainless steel lever arm design

Second design – Aluminium lever arm:

The lever arm is single piece made from a standard rectangular aluminium profile 20mm x 80mm of 2.060mm length. It has a total weight of 6,3 Kg. The design is similar to that of stainless steel, only differs in the lightening holes dimensions.

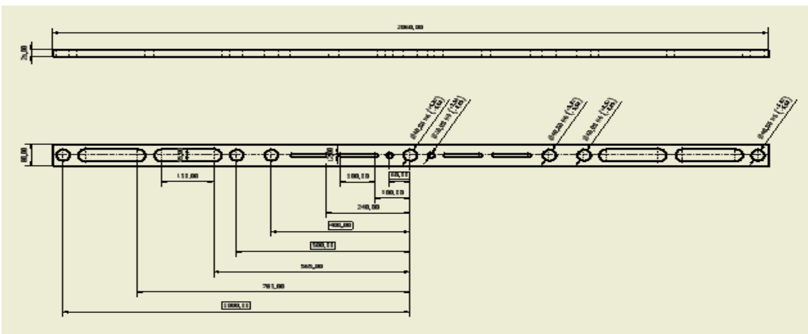


Fig. 7.17 - 2nd design - Aluminium lever arm design

Second design analysis:

Both designs (stainless steel and aluminium) satisfy the 1.000Nm calibration solicitations. An important fact is that the aluminium lever arm has less weight than the stainless steel lever arm but it’s little more bulky and has a higher linear expansion coefficient. Also aluminium is cheaper than stainless steel.

7.3 Weights and Hangers design

First design of the hangers

First idea was to use a calibrated steel hanger. It can be made or it can be bought in any metrological equipment manufacturer. Both hangers should be calibrated with the same accuracy as the masses to hang in order to avoid unbalancing torque. There are commercial hangers available with accuracy class M1, which is the approximated accuracy the masses will have. To hang them we need to make some holes in the lever arm.

Fig. 7.18 – Hanger’s first design

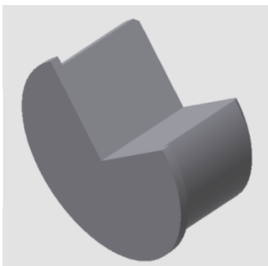
Hanger’s second design

We thought we could improve the hanger’s system accuracy with a new design. This new hanger has a sharp support that combined with the hanger support, which will be fitted in the lever arm hanger holes, allows an accurate location for the masses force actuating point. The force is transmitted from the hanger to the lever arm via one single line. We can measure accurately the distance from this point to the centre of the lever arm.

This hanger is made by different aluminium parts that have to be assembled. Some other parts such as nuts, washers or clevis pins are commercial. It has an approximate weight of 0,8 kg.

Fig. 7.19 – Hanger’s second design

Fig. 7.20 – Support for the hanger



Weight's design

Weights are an important part of the design because a great portion of the applied torque uncertainty depends directly from the weights uncertainty. Table 7.2 shows the combinations of weights and lever arm lengths used to produce the rated torque of the different torque transducers.

Rated torque (Nm)		Length (m)					
		-1	-0,5	-0,4	0,4	0,5	1
Set of weights (N)	1.000	-1000					1000
	500	-500		-200	200		500
	100	-100	-50			50	100

Table 7.2 – Combinations of weights and lengths to produce the rated torque

As we can see in chapter 4 – "Calibration process", to calibrate each torque transducer we need to apply eight torque steps for an accuracy class 0,1 of the torque transducer (the best accuracy class of our torque transducers). These steps are 10%, 20%, 30%, 40%, 50%, 60%, 80% and 100% of the rated torque. Considering this fact we will have the following weight's sets:

- 1.000N Weight's set:
  - 6 x 100N weights. or 10x 100 N weights.
  - 2 x 200N weights.
- 500N Weight's set:
  - 6 x 50N weights.
  - 2 x 100N weights.
- 100N Weight's set:
  - 6 x 10N weights.
  - 2 x 20N weights.

These weights are going to be hanged from the lever arm, so they will be slotted weights in order to load and unload them easily from the hanger. As we saw in chapter 5 – "State of art", the calibration equipment manufactures usually employ M1 class weights made of stainless steel or cast iron.

We will use the weights available in the ETEC laboratory. The weights we still don't have in the lab we can make and then calibrate them with the precision scales available in the V.U.B. As we will see in chapter 10.2 – "Uncertainty of the transmitted torque calculation" we will use weights with an accuracy of 0,1%. Depending of the type of weight scale we will have to take into account the Buoyancy effect or not for the calculations. This is taken into account in the calibration test software.

7.4 Adapters and supports

Once we had an idea of how the lever arm would be, we started the design of the rest parts. In this section we are going to show all the parts that have to be designed to support the torque calibration set and to be able to mount the different torque transducers with several lengths and couplings in the calibration bench. In the following paragraphs we explain the different designs we made before reaching the optimum one.

7.4.1 First design

Firstly we fixed the bench on a clamping table from the ETEC laboratory. A rotatory table is used vertically to be able to rotate the calibration set in order to level the lever arm in horizontal position during the calibration.

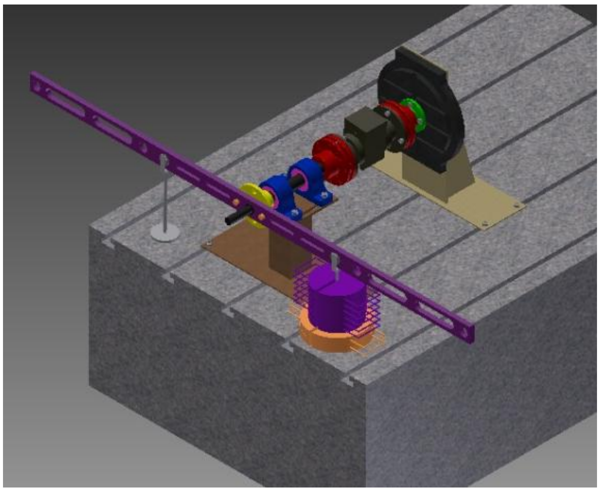


Fig. 7.21 - Calibration bench. First design

In Fig. 7.22 and Fig. 7.23 we have a scheme of the parts needed for the assembly. The parts in white are the ones that need to be designed. The new designed parts are: the Adapter lever arm-shaft, the Bearings base, the Adapter to rotatory table and the Foot for the rotatory table. The rest of the parts currently exist in the laboratory: the bearings, the shaft, the flexible couplings and the rotatory table. We measured their dimensions using the appropriate measuring systems (micrometre, calliper, metre, ...) in order to translate them to the 3-D design program.

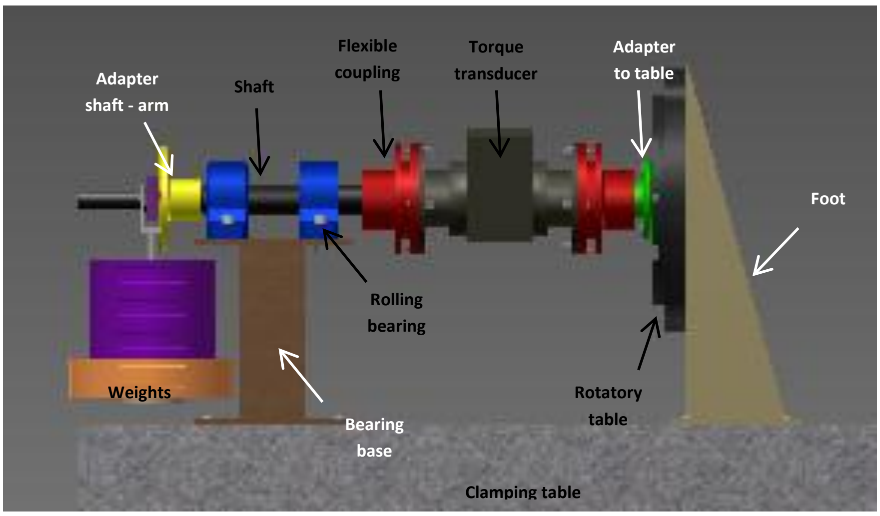


Fig. 7.22 - Calibration bench side view. First design

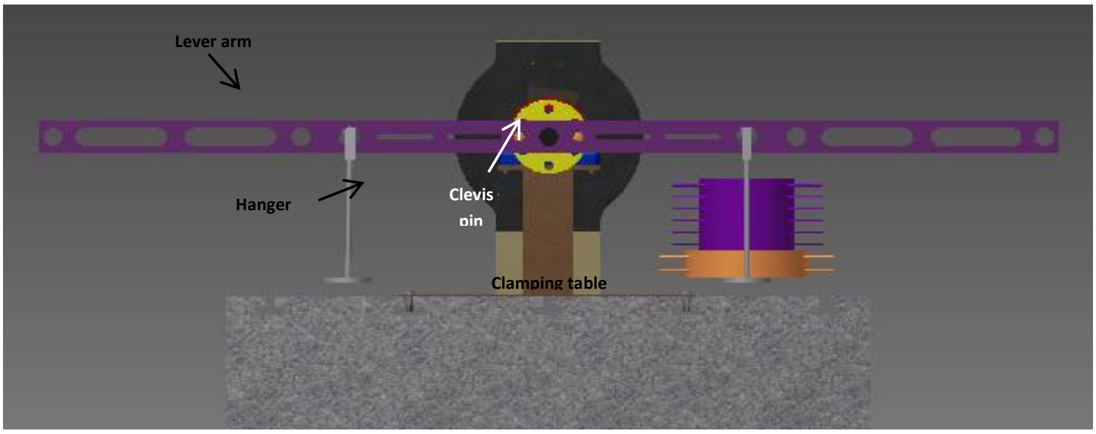


Fig. 7.23 - Calibration bench front view. First design

In the following paragraphs we are going to show the different designed parts.

Adapter to rotatory table

This adapter couples the flexible coupling with the rotatory table. In one end it has a key way that fits with the flexible coupling key end. In the opposite end it has a flange end to connect it with the rotatory table with clamping screws. Also it has a centring shaft end in order to locate it accurately on the rotatory table without waste of time doing the centring. We can make it for several types of flexible couplings in order to be able to mount the different torque transducers in the bench.

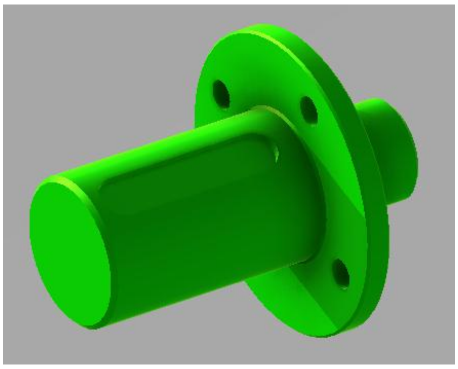


Fig. 7.24 - Adapter to rotatory table

Bearing base and Foot

The lever arm is larger than the clamping table but we need to hang weights from different positions of the lever arm in order to reduce the number of weights needed. For that reason we need a minimum height distance from the clamping table to the lever arm to be able to position the hangers with the weights. Also the calibration set can't overhang the front of the clamping table because the laboratory corridors must be free of obstacles and the arm has to be protected against possible impacts that may damage it or change its specifications. That's why we need to design a support for the bearings and another for the rotatory table which fix the calibration set at a certain height. These two parts are the "Bearing base" and the "Foot". They are made from welded 5mm thick steel sheets and they are fixed to the clamping table with clamping screws.

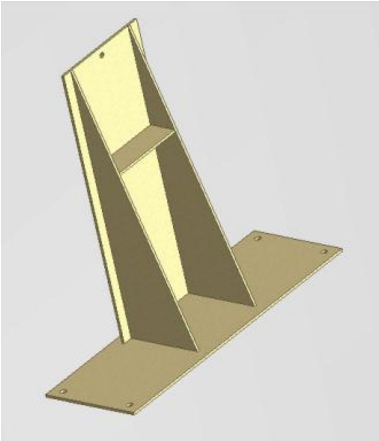


Fig. 7.26 - Foot

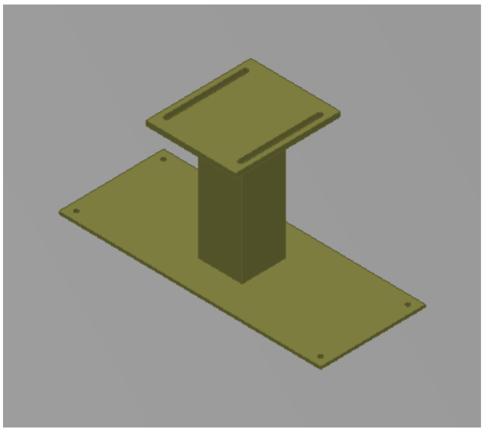


Fig. 7.25 - Bearing base for first design

Adapter lever arm - shaft

The lever arm is centered with the shaft by its central hole but it needs a way to transmit the applied torque to the shaft. This adapter transmits the torque applied by the weights hanged in the lever arm to the shaft. It is connected with the shaft through key end. In the opposite side it has holes at different angles, four each 90º and six each 120º, for the different positions of the lever arm during the calibration. It allows to calibrate torque transducers each 90º or 120º. The coupling between the adapter and the lever arm is done via clevis pins (see Fig. 7.27), which allow an easy assembly and disassembly of the lever arm as well as be able to rotate the lever arm without removing it from the shaft.

Fig. 7.28 – Adapter lever arm - shaft

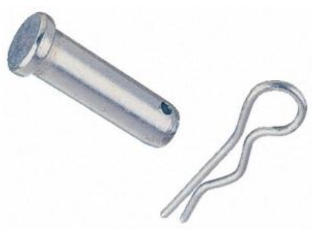


Fig. 7.27 - Clevis pin [30]

Comments of the first design

We thought we could use the clamping table from a milling machine that was located in the laboratory and was currently disused. This table is narrower that the one we were using for the design so we can reduce the supports dimensions. We made a second design.

7.4.2 Second design

In this design the calibration set is supported on a clamping table from a milling machine. We use a universal divisor as gearbox. It allows us to rotate the calibration set in order to level the lever arm horizontally and lock its position once we have achieved the appropriate position. Due to the short width of the clamping table we can use the whole length of the lever arm to hang the weights without having to lift the calibration set. So we need smaller supports than in the first design.

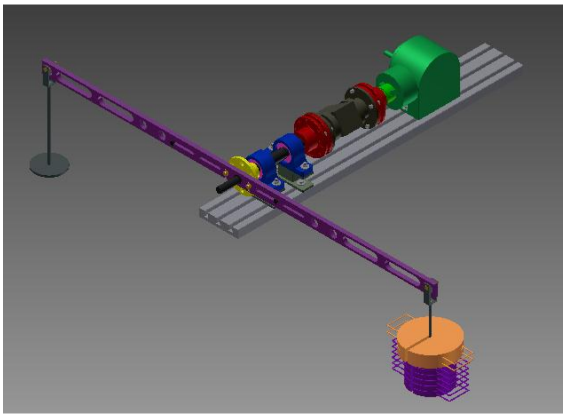


Fig. 7.29 - Calibration bench. Second design

In Fig. 7.30 and Fig. 7.31 we have a scheme of the parts we need for the calibration bench. Red named parts are those which need to be designed.

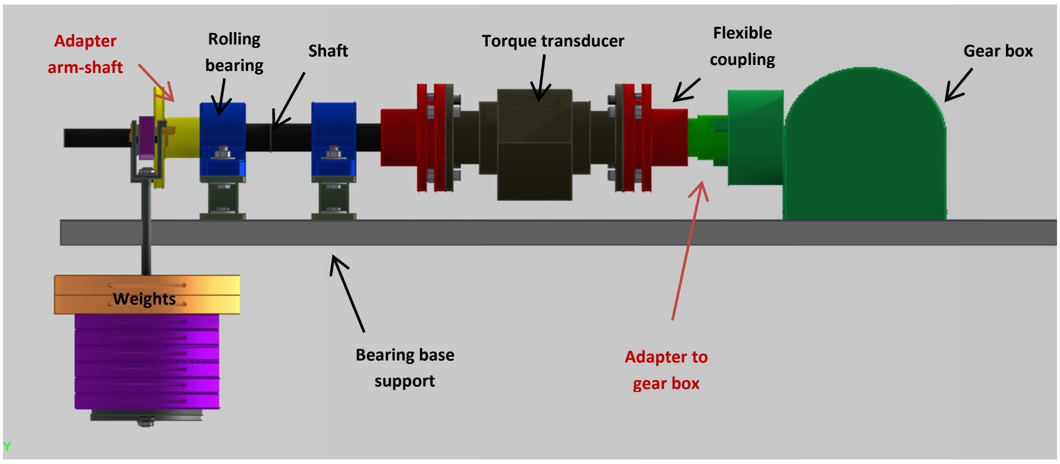


Fig. 7.30 - Calibration bench side view. Second design

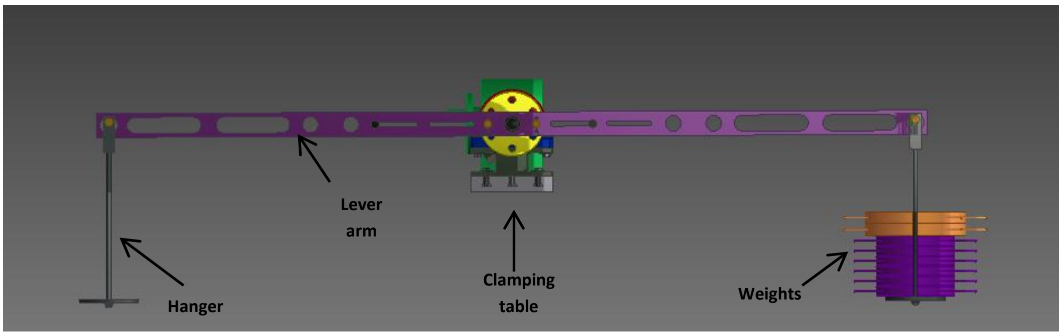


Fig. 7.31 - Calibration bench frontal view. Second design

The new parts we have to make are: the “Adapter lever arm-shaft” and the “Adapter to the gear box”. The rest of the parts currently exist in the ETEC laboratory.

Adapter to gear box

This part couples the flexible coupling with the gear box. It's connected with the flexible coupling via key end. The diameter of the shaft depends of the diameter of the flexible couplings, so three different adapters will be needed for the calibration of the different torque transducers. The opposite side has a shaft end.

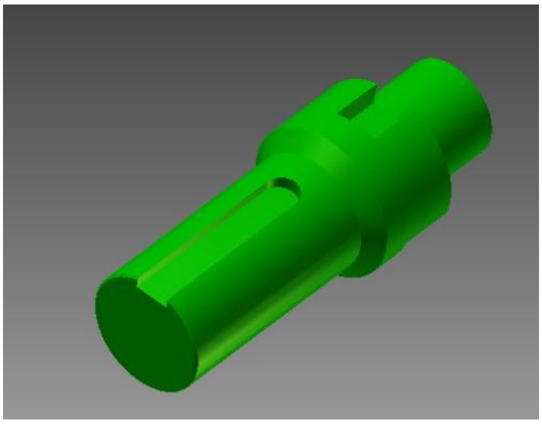


Fig. 7.32 - Adapter to gearbox

The universal divisor has three jaw chucks to fix and centre the adapter's shaft. The shaft end has machined safety stops to prevent from accidental rotation in case of failure of the jaw chucks grip.

Adapter lever arm-shaft

This adapter is the same as explained for the *First design*. It couples the lever arm with the shaft and transmits the applied torque.

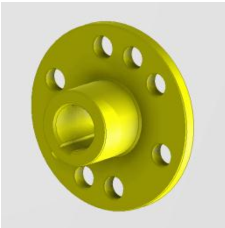


Fig. 7.34 - Adapter lever arm - shaft

Fig. 7.33 - Clevis pin

Comments of the second design

With this design we reduced the number of parts to make and we simplified its assembly. We simulate its behaviour with the Autodesk Invertor Stress Analysis and we checked that one of the bearings stand the majority of the load produced by the weights. For 500N weight one bearing stand 1000N and the other -500N (both vertically) in order to balance the bending moment induced by the weights to the shaft.

We thought these reactions could be reduced significantly locating the lever arm between the two bearings, so we have for 500N weight that each bearing stand 250N vertical load. Adding the weight of the rest components we checked that we can reduce the friction torque to a half using this new system.

7.4.3 Third design

In this design the lever arm is located between the two roller bearings in order to distribute equally the load between them and to minimize the bending effects. Also the better distribution of the load means less friction loses from the bearings.

The parts are the same from the Second design with slightly different dimensions. It requires the design of a new shaft of 45mm diameter.

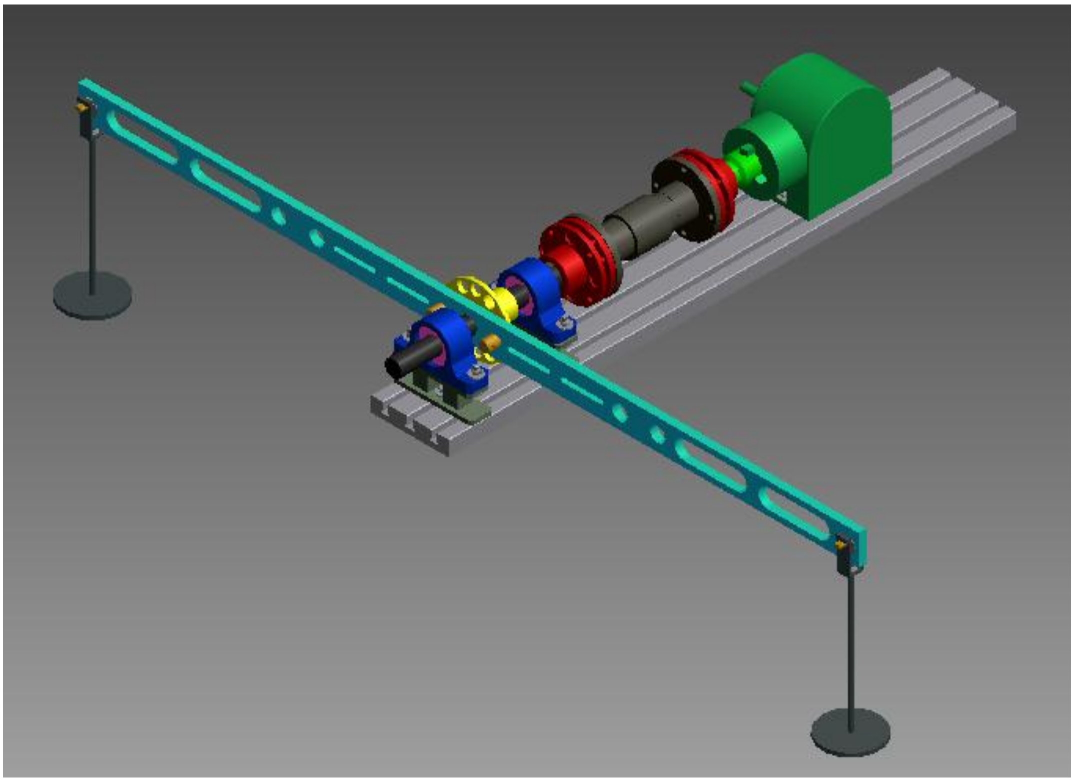


Fig. 7.35 – Calibration bench. Third design

Shaft

A new shaft has to be designed regarding the rolling bearings specifications: in normal cases shafts machined to h9 tolerance can be used with cylindricity tolerance IT5/2, for less demanding applications h10 and IT7/2 may be satisfactory. Surface roughness Ra should not exceed 3,2µm. [9] Also it has to couple the 55mm diameter flexible couplings. In the middle of the shaft there is a key way to couple the Adapter arm-shaft and be able to centre it between the rolling bearings.

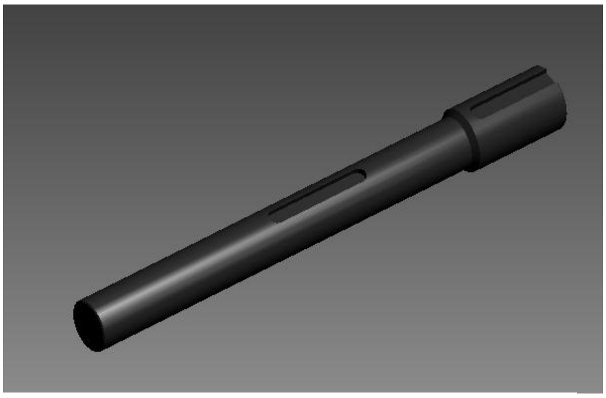


Fig. 7.36 - Designed shaft

7.5 Indicating device

The indicating device is the electronic device that visually shows the voltage output of the torque transducer with a certain resolution. The resolution *r* is considered to be one increment of the least significant active digit of the numerical indicating device, provided that the indication does not fluctuate by more than one increment when the instrument is unloaded. If the reading (with the instrument unloaded) fluctuates by more than the value previously determined for the resolution, the resolution should be deemed to be equal to half the range of fluctuation. The resolution *r* shall be converted to units of torque using the sensitivity factor *S* at M_E , the maximum torque value of the measuring range.

Taking into consideration the resolution *r* with which the indicator can be read, the minimum torque M_A applied to a torque measuring device should be not less than 0,02 M_E (2% of the maximum torque value of the measuring range); see also Table 4.1. From that table we obtain the following expression. [5]

$$M_E \geq 2000r \; for \; accuracy \; class \; 0.1$$

Eq. 7.1

Our torque transducers have an output of ±5 V_{DC} for rated toque. The output for 2% of the rated torque is ±0,1 V_{DC}.

$$r < \frac{0{,}1\,V}{2000} = 0{,}05\,mV$$

Eq. 7.2 – Indicating device resolution

We deduce that the resolution our indicating device should have for the calibration is at least 0,05 mV.

8. Resolution adopted

In the previous paragraphs we have exposed the different proposed designs to mount the Static calibration bench, see chapter 7 – "Design". In this chapter we are going to discuss the resolution adopted and the reasons which have led us to select that design. The design was made for 500Nm and 200Nm torque transducers calibration.

8.1 Components

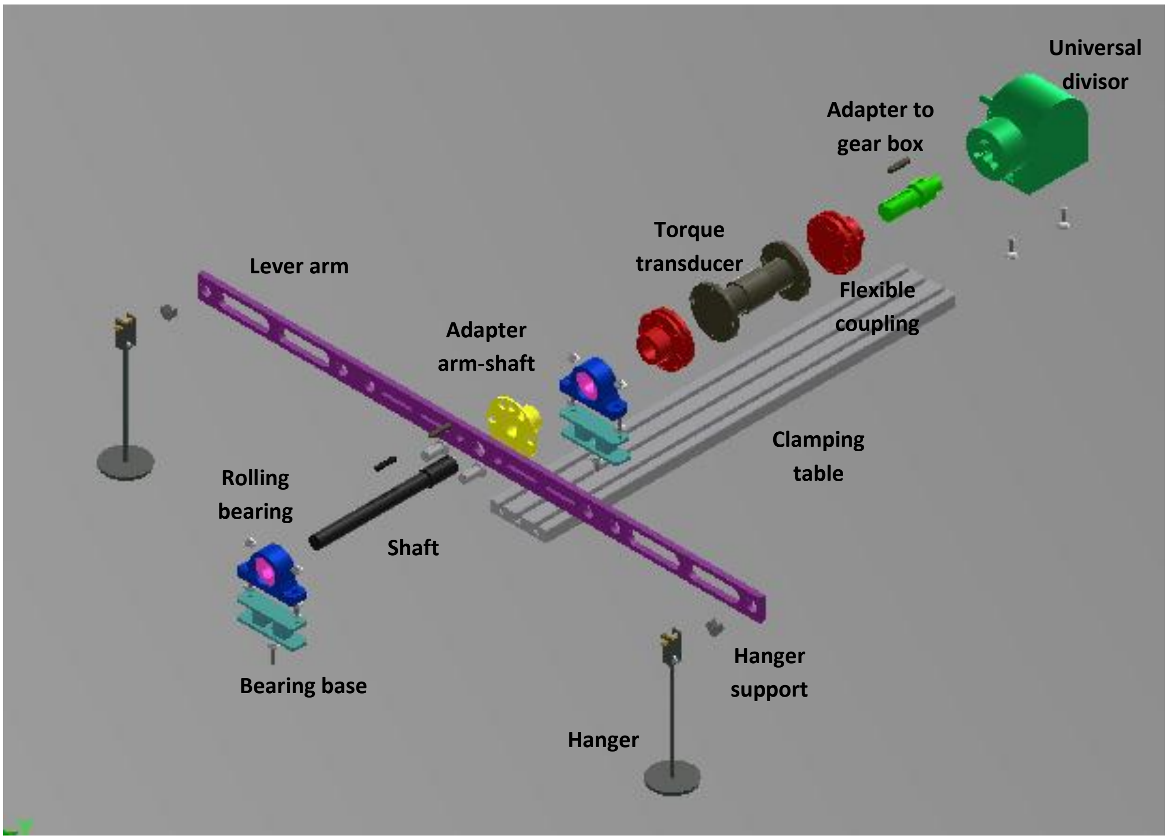


Fig. 8.1 - Resolution adopted assembly

Lever arm

We decided to use the aluminium lever arm made from a rectangular profile (Aluminium 7075-O lever arm, second design). It is symmetrical double ended in order to be able to do clockwise and anti-clockwise calibration tests. The aluminium protects the arm against corrosion effects and ensures a long life operation.

The Lever arm is centred with the shaft via a precision sliding fit 45 H7/h6, which permits movement by hand with a lubricant applied and can be assembled an disassembled easily without damaging the components. The Hanger supports are also accurately located in the lever arm via sliding fits 40 H7/h6.

The lever arm is balanced with several balancing weights that can be located at different distances from the centre. The balancing canals are machined near the centre in both sides of the lever arm in order to have a high precision balance. The arm has lightening holes to reduce the weight in its ends, where it's less solicited. A reduction of the arm's weight means less friction torque loses and easier handling.

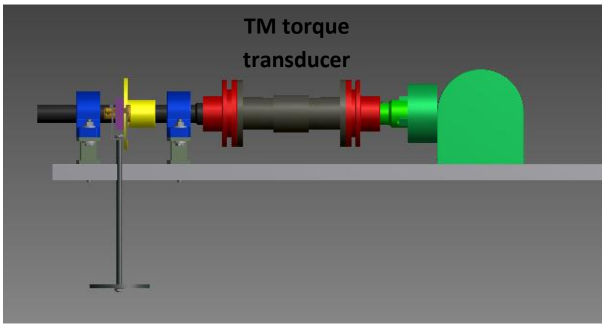
The Lever arm has to be equipped with a level spirit in its centre in order to measure its levelling angle.

The torque is transmitted to the shaft through the clevis pins and the Adapter lever arm-shaft. It has 30mm diameter machined holes with tolerance H9. Here we don't need great location accuracy so we will use commercial 30mm diameter clevis pins h11.

Adapters

The adapters were made to connect the Vibro-meter TT 500Nm, TM 500Nm and TT 200Nm torque transducers. For the rest of the transducers new adapters are needed. There is a 35mm length difference between the TT and TM 500 Nm torque transducers. This is not a problem because the shaft and the universal divisor can be displaced longitudinally.

In Fig 8.2 we see a scheme of the mounting positions for TT and TM torque transducers keeping fixed the distance between the bearings and the universal divisor.



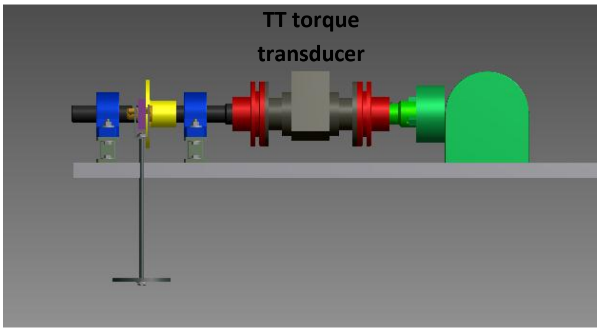


Fig. 8.2 - Comparison assembly for TM and TT torque transducers

The Adapter arm-shaft and the Adapter to rotatory table can be made in stainless steel for 1.000Nm rated torque or in aluminium for 500Nm torque. We decided to make them in aluminium because firstly we want to calibrate transducers up to 500Nm and they are cheaper that the ones in stainless steel. Other advantage is the less weight of aluminium which is important for the calibration of small rated torque transducers (TT 50Nm and TT 100Nm).

Supports

To support the calibration set we are going to use the existing stainless steel Bearing bases and the universal divisor. Both are clamped to the milling table with clamping screws. The universal divisor's plate can be rotated manually when is loaded and it's self-locking.

The new shaft is made of stainless steel and one of its ends has a key end for the 500/200Nm flexible coupling.

8.2 Indicating device

As shown in chapter 7.5 we need an indicating device with a resolution of at least 0,05mV. In the ETEC laboratory we have precision multimeters that are normally used for the calibration of other electronic devices.

We will use as indicating device for the calibration bench the precision multimeter HP 34401A. It has a resolution of 4, 5, 6 or $6^1/_2$ digits selectable. The more digits we choose the better accuracy and the less speed of the measures.

Fig. 8.3 - Multimeter HP 34401A [10]

It has a DC measuring range up to 1.000V. We can choose different measuring ranges: 100mV, 1V, 10V, 100V or 1.000V. For the calibration we will use the 10V range because the rated output for our torque transducers is 5V. In this range we have a resolution of 0,01mV, which meets our requirements (0,05mV). Values based on the specification sheets [10]

9. Stress analysis of the resolution adopted

In this chapter we are going to analyse the stresses of the different designed parts to check that they support the loads to which they will be subjected.

9.1 Stress analysis theory

We are going to do a brief explanation of the stress analysis theories we are going to apply for the design of the different parts of the machine. For these calculations we are going to use the Autodesk Inventor Stress Analysis application, which is based in Von Mises failure theory. To understand the failure theories we have to know first what the principal stresses are, which is the particular notation system that these theories use, and why the necessity to create this notation system.

9.1.1 Principal stresses

In practice there are many mechanic components which are subjected to several types of loads simultaneously. A transmission shaft is subjected to bending as well as torsional moment at the same time. In design, it is necessary to determine the state of stresses under these conditions. The stresses are classified for the analysis into two groups: normal stresses and shear stresses. The normal stress is perpendicular to the area under consideration, while the shear stress acts over the area.

There is a particular notation system for these stresses. The normal stresses are denoted by σ_x, σ_y and σ_z in the *x*, *y* and *z* directions. Tensile stresses are considered to be positive, while compressive stresses as negative. The shear stresses are denoted by subscripts viz. τ_{xy} or τ_{yx}. The first subscript denotes the area over which it acts and the second indicates the direction of shear force. The shear stresses are positive if they act in the positive direction of the reference axis. The normal to the plane makes an angle *θ* with the *x*-axis, *σ* and *τ* are normal and shear stresses respectively, associated with this plane. [11]

Eq. 9.1 and Eq. 9.2 are the equations to change the reference system for normal and shear stresses.

$$\sigma_{x1} = \frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2} cos2\theta + \tau_{xy} sin2\theta$$

Eq. 9.1 [12]

$$\tau_{xy1} = -\frac{\sigma_x - \sigma_y}{2} sin2\theta + \tau_{xy} cos2\theta$$

Eq. 9.2 [12]

These equations are known as the transformation equations for plane stress because they transform the stress components from one set of axes to another. Setting τ_{xy1} to zero we obtain:

$$\tan 2\theta = \frac{2\tau_{xy}}{\sigma_x - \sigma_y}$$

Eq. 9.3 [12]

If σ_1 and σ_2 are the maximum and minimum values of normal stress, replacing θ value in Eq. 9.1 we obtain the following equations.

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + {\tau_{xy}}^2}$$

Eq. 9.4 [12]

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + {\tau_{xy}}^2}$$

Eq. 9.5 [12]

σ_1 and σ_2 are called the Principal Normal Stresses.

$$\tau_{max} = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + {\tau_{xy}}^2}$$

Eq. 9.6 [12]

τ_{max} is called the Principal Shear Stress. [11]

9.1.2 Failure theories

The design of machine parts subjected to combined loads should be related to experimentally determined properties of material under 'similar' conditions. However, it is not possible to conduct such test for different possible properties. In practice, the mechanical properties are obtained from simple tension test. They include yield strength, ultimate tensile strength and percentage elongation. In tension test, the specimen is axially loaded in tension. It is not subjected to either bending moment, torsional moment or a combination of loads.

Theories of elastic failure provide a relationship between the strength of machine component subjected to complex state of stresses with the mechanical properties obtained in tension test. With the help on these theories, the data obtained in tension test can be used to determine the dimensions of the component, irrespective of the nature of stresses induced in the component due to complex loads. Several theories have been proposed, each assuming a different hypothesis of failure. The principal theories of elastic failure are as follows:

I. Maximum principal stress theory (Rankine's theory).
II. Maximum shear stress theory (Coulumb, Tresca and Guest's theory).
III. Distortion energy theory (Huber von Mises and Hencky's theory).
IV. Maximum strain theory (St. Venant's theory).
V. Maximum total strain energy theory (Haigh's theory).

Let us assume that σ_1, σ_2 and σ_3 are the principal stresses induced at a point on the machine part as a result of several types of loads. We will apply the theories of failure to obtain relationship between that σ_1, σ_2 and σ_3 on one hand and the properties of material such as S_{yt} (yield strength) or S_{ut} (ultimate tensile strength) on the other. [12]

Maximum shear stress theory and distortion energy theory are used for ductile materials, such as aluminium and steel. Both are theories to be applied to our design. The other theories are used for brittle materials, which hardly undergo plastic deformation before the rupture failure. We will study in greater depth Von Mises failure theory because it is the criterion used by the simulation program we will use.

Distortion-energy theory (Von Mises)

Distortion-Energy theory was advanced by M.T. Huber in Poland (1904) and independently by R. von Mises in Germany (1913) and H. Henchy (1925). It is known as the Huber von Mises and Henchy's theory. The theory states that the failure of a mechanical component subjected to bi-axial or tri-axial stresses occurs, when the strain energy of distortion per unit volume at any point in the component, becomes equal to the strain energy of distortion per unit volume in a standard specimen of tension-test, when yielding starts. For this criterion we use the Von Misses equivalent stress, so we can calculate the safety factor for the mechanic component comparing it with the yield strength from the simple tension test:

$$\sigma_{VM} = \sqrt{\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_1-\sigma_3)^2}{2}} = \frac{S_Y}{fs}$$

fs: Factor of safety

$$\sigma_1 \geq \sigma_2 \geq \sigma_3$$

Eq. 9.7 - Von Mises yield criterion. [12]

For bi-axial stresses ($\sigma_3 = 0$):

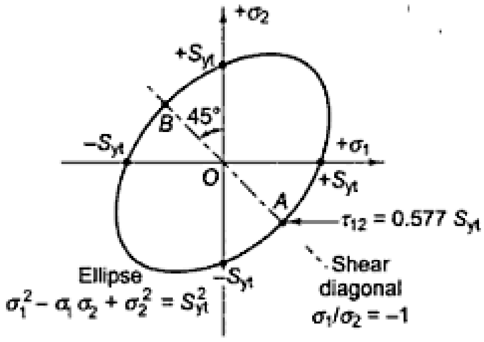


Fig. 9.1 - Bi-axial stress Von-Mises diagram. [12]

$$\sigma_{VM} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + {\sigma_2}^2 + {\sigma_1}^2}{2}} = \frac{S_Y}{fs}$$

Eq. 9.8 – Von Mises yield criterion for bi-axial stresses. [12]

Distortion energy theory predicts yielding for ductile materials with precise accuracy in all four quadrants. The design calculations involved in this theory are slightly complicated as compared with those of maximum shear stress theory. Based [12]

9.2 Autodesk Inventor Professional

We will use Autodesk Inventor Professional, which allows 3-D design of the different components and their posterior stress analysis simulation. It is a really appropriated tool for this project and it was available at the VUB with license. Firstly we will do a brief explanation of how it works.

Stress Analysis in Autodesk Inventor Simulation is an add-on to the Autodesk Inventor assembly, part, and sheet metal environments. Autodesk Inventor Simulation Stress Analysis provides tools to determine structural design performance, to place loads and constraints on a part or assembly and calculate the resulting stress, deformation, safety factor, and resonant frequency modes. You can visualize the affects in 3D volume plots, create reports for any results, and perform parametric studies to refine your design. Stress analysis is done using a mathematical representation of a physical system composed of:

- A part or assembly (model).
- Material properties.
- Applicable boundary conditions (loads, supports), contact conditions, and mesh, referred to as pre-processing.
- The solution of that mathematical representation (solving).To find a result, the part is divided into smaller elements. The solver adds up the individual behaviour of each element to predict the behaviour of the entire physical system by resolving a set of simultaneous algebraic equations.
- The study of the results of that solution is referred to as post-processing. [13]

Analysis Assumptions

The stress analysis is appropriate only for linear material properties, which means the material working in the elastic region. The total deformation is assumed to be small in comparison to the part thickness. The results are temperature-independent. The temperature is assumed not to affect the material properties. [13]

The analysis is based in the Finite Element Method, which consists of solving a system of differential equations evaluated for each element in which is divided the mechanic component. These equations are based in Von Mises failure theory and deformation theory.

Typical Stress Analysis workflow

With the Stress Analysis you can analyse a model under different conditions using various materials, loads and constraints (also called boundary conditions), and then view the results. You have a choice of performing a static analysis or a frequency (also called modal) analysis with associated mode shapes. After you view and evaluate the results, you can change your model and rerun the analysis to see the effect your changes produce. [13]

The typical Stress Analysis workflow is as follows:

1. Create Simulations and specify their properties.
2. Exclude components not required for simulation.
3. Assign materials.
4. Add Constraints (Fixed, pin or frictionless constraint).
5. Add Loads (Force, pressure, bearing load, moment or gravity)
6. Specify contact conditions, an optional step.
7. Specify and preview the mesh, an optional step.
8. Run the simulation.
9. View and interpret the results.

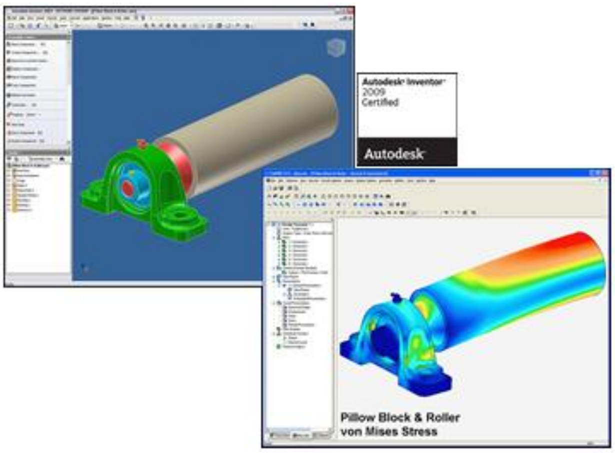


Fig. 9.2 - Autodesk Inventor Stress Analysis [13]

Results

Once the analysis has concluded, you can see the results of your solution in 3-D coloured plots. If you did a stress analysis, the Von Mises Stress results set displays, which contains the following information:

- ✓ Von Mises Stress.
- ✓ 1st Principal Stress.
- ✓ 3rd Principal Stress.
- ✓ Displacement.
- ✓ Safety Factor.

9.3 Designed parts stress analysis

We are going to analyse the different designed parts using Autodesk Inventor Stress Analysis. During the design process we have made several designs and then we have checked them with the stress analysis. Considering the results obtained we have modified some parts of the design to achieve a valid security factor. The calibration test bench will ideally work in static conditions, there won't be fatigue effects due to cyclic loading, so the safety factor should be around 1,5.

In this chapter we are going to show the stress and deformation analysis performed for the adopted resolution parts.

9.3.1 Structural analysis for 500 Nm

We simulated the Calibration test bench for 500Nm torque transducer calibration conditions (500N vertical load at 1m from the lever arm centre) using the required flexible couplings and torque transducer. We used for the simulation a 14x9x74mm key for the Adapter to shaft and 16x10x76mm keys for the flexible couplings, all of them made of low carbon steel.

As we can see in Fig. 9.3 we obtained a safety factor of 1,56 for the whole structure, so we can conclude the current design support the solicitations for 500Nm calibration.

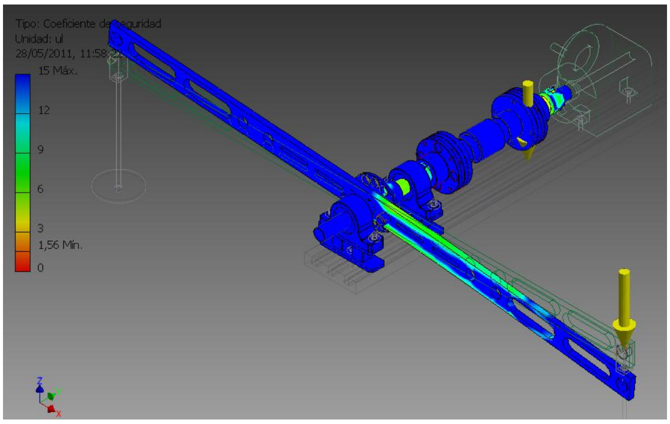


Fig. 9.3 - Safety factor graphic for 500 Nm

Also we obtained the values for the horizontal and vertical displacement of the point where the load is applied, see Fig. 9.4. The horizontal displacement is 0,0018mm, which is negligible compared with the length uncertainty of 0,1mm we will use. The vertical displacement is 8,7mm; this is corrected using the levelling gearbox. As we can see the deformation effect for the applied torque is negligible.

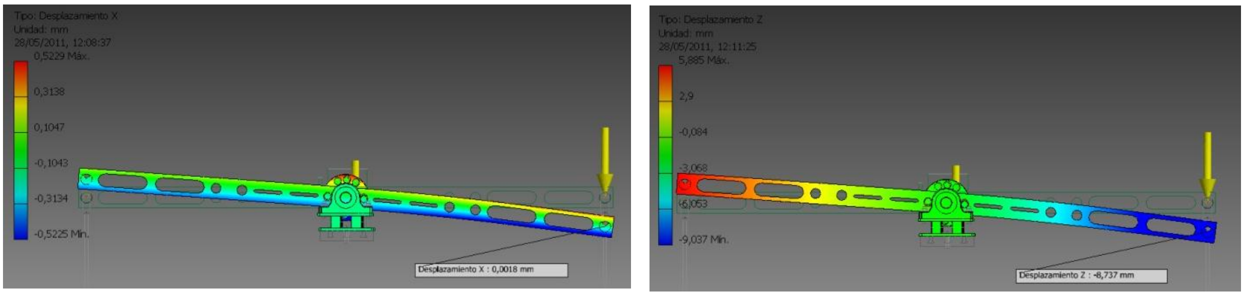


Fig. 9.4 – X-axis and Z-axis displacement for 500Nm

9.3.2 Structural analysis for 200Nm

We did a simulation for 200Nm torque transducer calibration conditions (500N load at 0,4m from the centre of the lever arm). The couplings and the torque transducer dimensions are the same to those of 500Nm calibration. As we see in Fig. 9.5 we obtained a safety factor of 3,88.

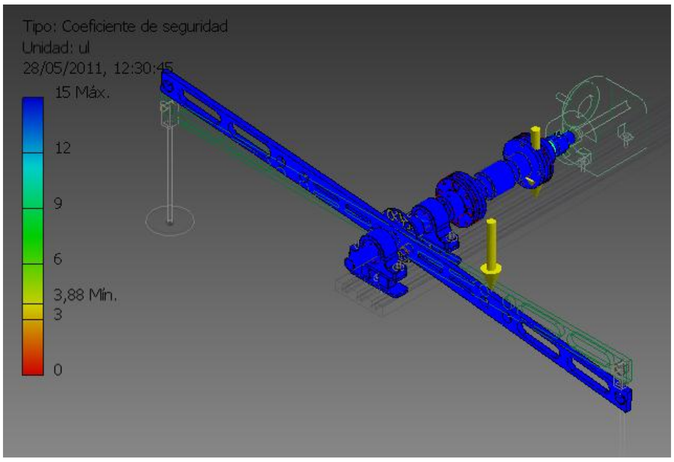


Fig. 9.5 - Structural stress analysis for 200Nm

9.3.3 Structural analysis for 1.000Nm

The lever arm, the hangers, the hanger's supports and the clevis pins will be designed for 1.000Nm calibration test conditions (1.000N load at 1m from the centre of the lever arm) in case the calibration bench is used in the future to calibrate the 1.000N torque transducer.

Also we checked the displacements for the point where the load is applied. We obtained a horizontal displacement of 0,005mm and a vertical displacement of 17mm (see Fig. 9.6). We can consider that the deformation is negligible for the length's uncertainty (0,1mm).

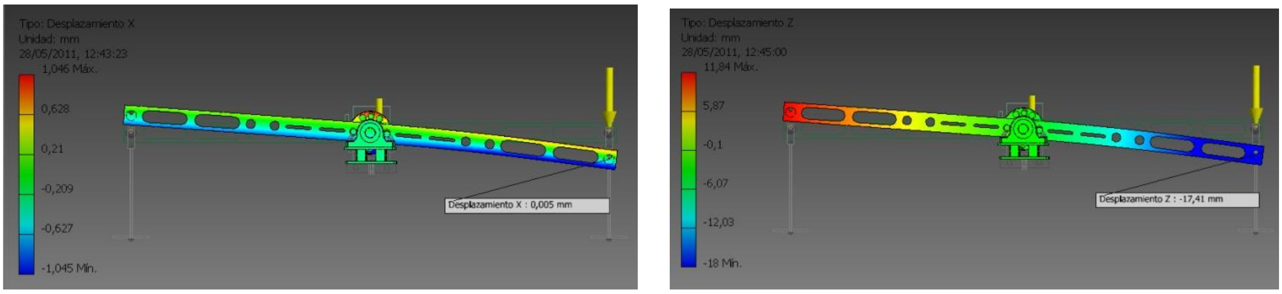


Fig. 9.6 - X-axis and Z-axis displacement for 1.000Nm

9.3.4 Stress analysis summary

From the global stress analysis we obtained the safety factor for each component. In Table 9.1 we have a summary of the stress analysis performed for the different designed parts. Further on we have the screenshots of the stress analysis we did for the different components.

Part	Material	Torque transducers calibration (Nm)	Security factor (for the worst conditions)	Weight (kg)
Lever arm	Aluminium 7075-0	1.000, 500, 200, 100, 50	2,2	6,3
Hanger support	Aluminium 7075-O	1.000, 500, 200, 100, 50	8,9	0,06
Adapter lever arm - shaft	Aluminium 7075-O	500, 200,100,50	1,85	0,9
Shaft	Stainless steel	500, 200, 100, 50	2,48	6,9
Adapter to gearbox	Aluminium 7075-O	500, 200,100,50	1,56	1,2
Hanger	Aluminium 7075-O	1.000, 500, 200, 100, 50	4,76	0,8
Clevis pin	Stainless steel	1.000, 500, 200, 100, 50	2,9	0,25

Table 9.1 - Stress analysis summary

Lever arm

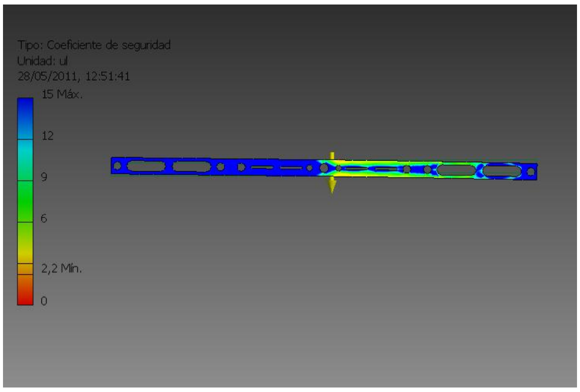


Fig. 9.7 – Lever arm stress analysis

S.F.=2,2 (1.000Nm)

Clevis pin

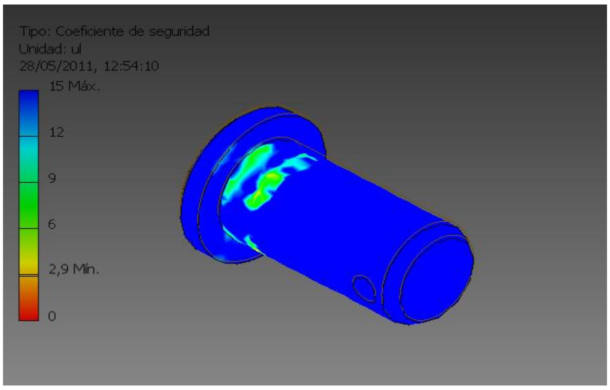


Fig. 9.8 – Clevis pin stress analysis

S.F.=2,9 (1.000Nm)

Adapter lever arm-shaft

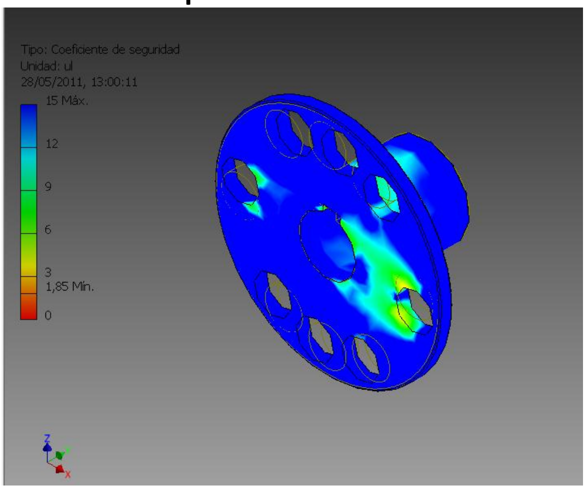


Fig. 9.9 - Adapter lever arm-shaft stress analysis

S.F.=1,85 (500Nm)

Adapter to gearbox

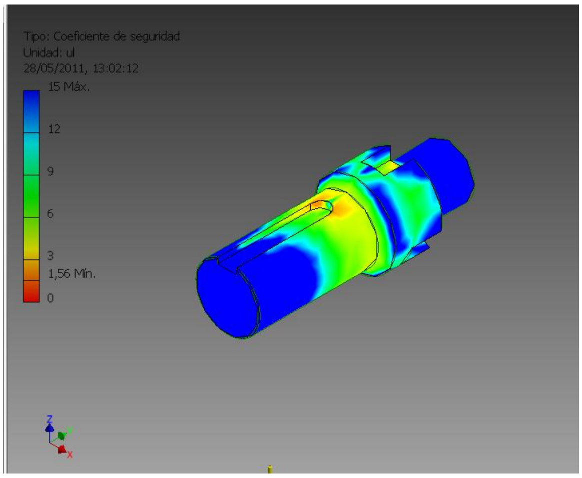


Fig. 9.10 - Adapter to gearbox stress analysis

S.F.=1,56 (500Nm)

Hanger support

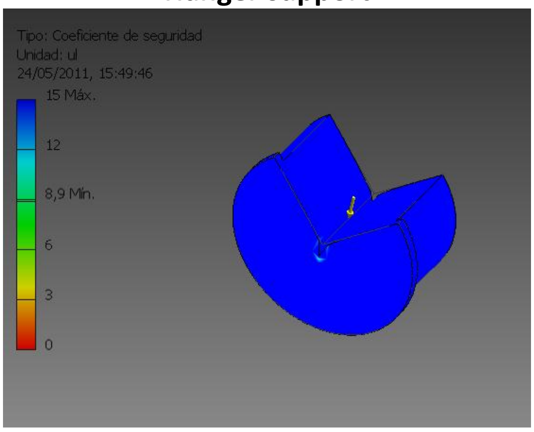


Fig. 9.11 – Hanger support stress analysis

SF=8,9 (1.000Nm)

Hanger

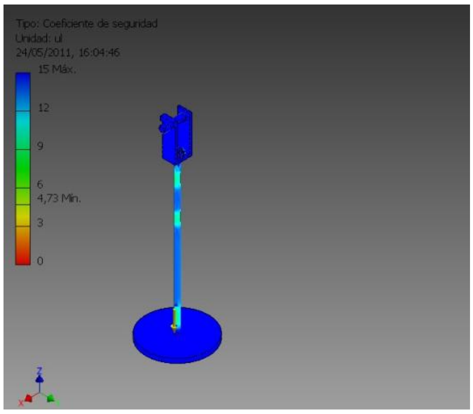
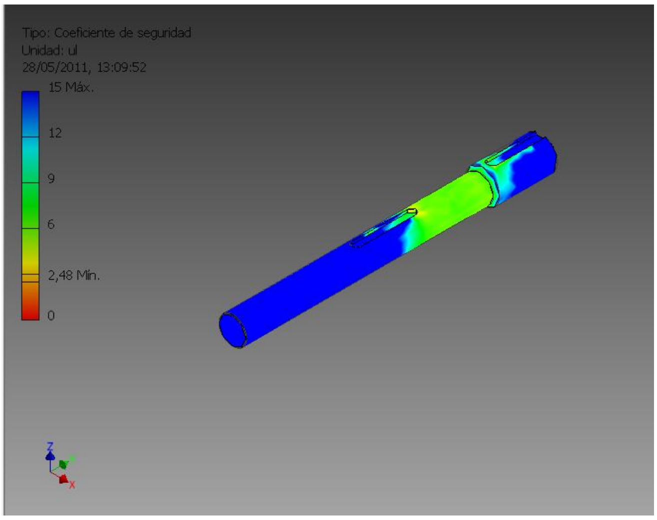


Fig. 9.12 – Hanger stress analysis

S.F.=4,7 (1.000Nm)

Shaft



S.F.=2,48 (500Nm)

10. Accuracy of the calibration test bench

In this chapter we are going to calculate the accuracy of the transmitted torque to the torque transducer by the Calibration test bench and the uncertainty of the torque measures during the calibration test. In "Annex V - Uncertainty theory" we have an explanation of the terms we are going to use.

10.1 Determination of the standard uncertainty of measurement for increasing torque

The calibration of torque measuring devices is carried out by comparison, using a torque calibration machine with known torque steps, or calibration equipment with a torque reference transducer. The calibration result is the output signal of the torque measuring device and is obtained from the approximate model:

$$\overline{X} = (S + \delta S_{b'} + \delta S_b + \delta S_{fa}) M_{\mathrm{k}} + \delta X_r$$

Eq. 10.1 [5]

Where:

M_{k}: *Torque generated by the torque calibration machine with an associated uncertainty* $u(M_{\mathrm{k}}) = u_{\mathrm{tcm}}$.

S: Sensitivity (mV/V)/Nm.

$\delta S_{b'}$ *: Repeatability with an associated uncertainty* $u(\delta S_{b'}) = \frac{S}{M_{\mathrm{k}}} u_{b'}$.

δS_b *: Reproducibility with an associated uncertainty* $u(\delta S_b) = \frac{S}{M_{\mathrm{k}}} u_b$.

δS_{fa} *: Deviation resulting from the fitting curve with an associated uncertainty* $u(\delta S_{fa}) = \frac{S}{M_{\mathrm{k}}} u_{fa}$.

δX_r *: Observed influence due to instrument resolution with an associated uncertainty* $u(\delta X_r) = S \cdot u_r \sqrt{2}$ *(two readings for one value indicated)*

The standard uncertainty $u(\overline{X})$ expressed in units of indication and the relative standard uncertainty $w(\overline{X})$ are obtained by the law of propagation of uncertainty in the approximation of non-correlated variables:

$$u(\overline{X}) = \sqrt{\sum_{i=1}^{5}\left(\frac{\partial \overline{X}}{\partial x_i}\right)^2 u^2(x_i)} \qquad w(\overline{X}) = \frac{u(\overline{X})}{\overline{X}} \cdot 100$$

With:

$$u^2(\overline{X}) = S^2\left(u_{\text{tcm}}^2 + u_{b'}^2 + u_b^2 + 2u_r^2 + u_{fa}^2\right)$$
$$w^2(\overline{X}) = \left(w_{\text{tcm}}^2 + w_{b'}^2 + w_b^2 + 2w_r^2 + w_{fa}^2\right)$$

Eq. 10.2 [5]

The example furnishes information on the uncertainty of measurement at the time of calibration. It does not allow for the uncertainty of components' long-term stability, or the influence of angular velocity and/or the effects of mechanical couplings used in practice, for example. [5] In Table 10.1 we show the different uncertainty distribution types we have and how to calculate them.

Quantity	**Evaluation of standard uncertainty**	**Standard uncertainty in Nm**	**Relative standard uncertainty in %**
Repeatability in unchanged mounting position *b'*	Type A	$u_{b'} = \frac{b'}{S\sqrt{2}}$	$w_{b'} = \frac{b'}{\sqrt{2}} \cdot \frac{100}{\overline{X}}$
Reproducibility in different mounting positions *b*	Type A	$u_b = \frac{b}{S\sqrt{n}}$	$w_b = \frac{b}{\sqrt{n}} \cdot \frac{100}{\overline{X}}$
Deviation resulting from fitting curve f_a	Type B with triangular distribution	$u_{fa} = \frac{\lvert f_a \rvert}{S\sqrt{6}}$	$w_{fa} = \frac{\lvert f_a \rvert}{\sqrt{6}} \cdot \frac{100}{X_a}$
Resolution *r*	Type B with rectangular distribution	$u_r = \frac{r}{\sqrt{12}}$	$w_r = \frac{r}{\sqrt{12}} \cdot \frac{100}{M_k}$
Reference torque	Type B	u_{tcm}	w_{tcm}

Table 10.1 - Uncertainty budget (increasing torque only). [5]

The *Type A evaluation of standard uncertainty* is the method of evaluating the uncertainty by the statistical analysis of a series of observations. In this case the standard uncertainty is the experimental standard deviation of the mean that follows from an averaging procedure or an appropriate regression analysis.

The *Type B evaluation of standard uncertainty* is the method of evaluating the uncertainty by means other than the statistical analysis of a series of observations. In this case the evaluation of the standard uncertainty is based on some other scientific knowledge. [5]

Calibration of devices with undefined scale

The expanded uncertainty of measurement *U* for each calibration step is calculated from the uncertainty of measurement. The expanded relative uncertainty of measurement *W* for each calibration step is calculated from the uncertainty of measurement. The coverage factor *k*=2 applies in both cases. [5]

$$U = k \cdot u(\overline{X}) \qquad W = k \cdot w(\overline{X})$$

Eq. 10.3 [5]

Calibration of devices with a non-adjustable defined scale or where a straight line fit only can be applied.

An exceptional case is where the indicator of the torque measuring device is non-adjustable, or has the capability only of fitting a straight line to the data. The values determined for f_q and f_a are treated as systematic errors whose moduli represent a non-dominant part of the uncertainty. In these cases the expanded uncertainty statement at the desired coverage probability of 95% can only be obtained by the following equations.

The standard uncertainty $u(\overline{X})$ expressed in units of indication and the relative standard uncertainty $w(\overline{X})$ of the random variables is calculated for each calibration step:

$$u(\overline{X}) = S\sqrt{u_{\text{tcm}}^2 + u_{b'}^2 + u_b^2 + 2u_r^2}$$
$$w(\overline{X}) = \sqrt{w_{\text{tcm}}^2 + w_{b'}^2 + w_b^2 + 2w_r^2}$$

Eq. 10.4 [5]

The appropriate formulas for the straight line fit are:

$$u_c(\overline{X}) = \sqrt{\left(\frac{f_a}{S}\right)^2 + u^2(\overline{X})} \qquad w_c(\overline{X}) = \sqrt{\left(\frac{f_a}{\overline{X}}\right)^2 + w^2(\overline{X})}$$

Eq. 10.5 [5]

f_a - is the deviation from the straight line fit

The appropriate formulae for the defined scale are:

$$u_c(\overline{X}) = \sqrt{\left(\frac{f_q}{S}\right)^2 + u^2(\overline{X})}$$

$$w_c(\overline{X}) = \sqrt{\left(\frac{f_q}{\overline{X}}\right)^2 + w^2(\overline{X})}$$

Eq. 10.6 [5]

f_q: deviation of indication of the torque measuring device with defined scale

The expanded uncertainty of measurement *U* or the expanded relative uncertainty of measurement *W* for each calibration step is calculated from the combined uncertainty of measurement with coverage factor k=2. [5]

$$U = k \cdot u_c(\overline{X})$$

$$W = k \cdot w_c(\overline{X})$$

Eq. 10.7 [5]

10.2 Uncertainty of the transmitted torque calculation

The torque transducers we have to calibrate have accuracy class 0,1 (for 500 Nm TM Vibro-meter torque transducers) and 0,2 (for 1.000 Nm, 500 Nm, 200 Nm, 100 Nm and 50 Nm TT Vibro-meter transducers). As seen previously in Table 4.1, we should achieve an expanded relative uncertainty of the torque transmitted by the Calibration test bench, W_{tcm}, less than 0,02% for TM 500Nm transducers and 0,04% for the rest of the transducers. We are going to check in this chapter that our design satisfies these requirements.

First, we have to know the expression of the torque transmitted to the transducer. Ideally it would be the torque generated by the weights at a certain lever arm distance; hereinafter we will refer to it as the applied torque T_a. But there are parasite torques due to the friction torque produced by the bearings T_f, the hanger's balancing error torque T_h and the lever arm's balancing error torque, T_b,.

Considering these effects we obtain the following equation to calculate the transmitted torque *T*:

$$T = T_a - T_f - T_b - T_h$$

Eq. 10.8

T: Transmitted torque
T_a: Applied torque.
T_f: Friction torque.
T_b: Arm's balancing error torque.
T_h: Hanger's balance error torque.

Once we have defined the expression for the transmitted torque we can calculate its combined standard uncertainty:

$$u_T = \sqrt{{u_{T_a}}^2 + {u_{T_f}}^2 + {u_{T_b}}^2 + {u_{Th}}^2}$$

Eq. 10.9

10.2.1 Applied torque uncertainty (Ta)

To calculate the applied torque *Ta* we have to apply the following expression:

$$T_a = P \cdot L = m \cdot g_L \cdot L$$

Eq. 10.10

P: Load of the weights [N].
g_L: Local gravity [m/s^2].
L: Lever arm's effective length [m].
m: Effective mass of the weights[kg].

The applied toque uncertainty can be calculated from Eq. 10.10 applying the combined relative standard uncertainty and the existing relation between the standard uncertainty and the relative standard uncertainty.

$$u_{T_a} = w_{T_a} \cdot T_a$$

$$w_{T_a} = \sqrt{{w_m}^2 + {w_{g_L}}^2 + {w_L}^2}$$

Eq. 10.11

To evaluate the applied torque relative uncertainty we have to study the uncertainty of its components: the arm's length, the local gravity and the mass.

Mass uncertainty

We are going to use masses with a relative expanded uncertainty of 0,01%, which approximates to M1 class. For the calculations we will take into account the Buoyancy effect.

$$m = m_{weight} \cdot \left(1 + \frac{\rho_a}{\rho_{weight}}\right)$$

Eq. 10.12

The effect of the Buoyancy in the mass uncertainty is negligible; it only affects the mean value.

$$W_{m\,(95\%)} = 0{,}01\%$$

Local gravity uncertainty

As seen in chapter 6.1, we are going to use a gravity value with an expanded uncertainty of 0,00002 $\mathrm{m/s^2}$, which implies an relative expanded uncertainty of 0,0002%.

$$W_{g\,(95\%)} = 0{,}0002\%$$

Length uncertainty

The lever arm's length depends on the ambient temperature due to the linear thermal expansion and the levelling angle as we can see in the following equation:

$$L = L_o \cdot \big(1 + \alpha(T - To)\big) \cdot cos\theta$$

Eq. 10.13

L: Arm's effective length [m].
Lo: Arm's initial length [m].
α: Thermal expansion coefficient.
θ: Levelling angle [º].
T: Ambient temperature during the calibration test [ºC].
To: Reference temperature [ºC].

To simplify the uncertainty calculation we will study both effects separately because, as we will see, we are going to minimize them with the levelling gearbox and the temperature compensation, so the combined effect will be negligible. We will try to limit the length's expanded relative uncertainty below 0,01% in order to achieve the relative expanded uncertainty of the transmitted torque (0,02% for 500 Nm and 0,04% for the rest).

$$W_{l\,(95\%)} = 0{,}01\%$$

-Temperature effect in arm's length:

We are going to study the effect of temperature in beam's length in order to fix an operating temperature range for the Calibration test bench. The reference temperature will be 20ºC as in most industrial activities. We should measure the different hang point's distances from the centre of the lever arm at this temperature. Otherwise we should take as reference temperature the one used for the length measures.

Linear coefficient, α, at 20 °C (10^{-6}/°C):

Steel: 11 ~ 13 *(Depends on composition)*
Stainless steel: 17,3
Aluminium: 23

Beam's length variation due to temperature effect:

$$\frac{\Delta L}{L} = \alpha \cdot (T - 20)$$

Eq. 10.14

We will try to maintain this value below 0,005% in order to have a length expanded uncertainty below 0,01%.

$$\frac{\Delta L}{L_0} \cdot 100 = \alpha \cdot (T - 20) \cdot 100 < 0.005\%$$

Eq. 10.15

We are going to make the lever arm in aluminium, which has a linear expansion coefficient of $23 \cdot 10^{-6}$/ºC. For this material we obtain an operating temperature range of 20ºC ±2ºC. Keeping these limits there is no need for temperature compensation.

$W_{L,T} = 0{,}005\%$ within the operating temperature range.

If the beam must be used outside these limits, the temperature must be stable (with 1ºC change per hour) and the effective length of the beam has to be calculated according the following formula:

$$L = L_o \cdot [1 + \alpha \cdot (T - 20)]$$

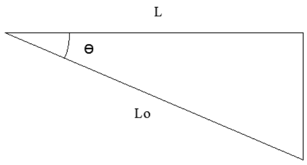
Eq. 10.16 – Temperature compensation formula

L: Effective length [m].
Lo: Length at 20ºC [m].
α: Linear expansion coefficient [m/ºC].
T: Ambient temperature during calibration [ºC].

Since we are going to use a computer program to do the calibration test we can do automatically the compensation introducing the ambient temperature. Using the temperature compensation with 1ºC uncertainty we will have a length's uncertainty of 0,002%. For the following calculations we will assume a length's relative expanded uncertainty due to temperature: $W_{L,T} = 0{,}002\%$

- ***Levelling angle effect:***

We are going to analyse the contribution of the levelling angle to the length's uncertainty. We will try to maintain the relative expanded uncertainty due to the levelling angle below 0,005% in order to achieve an expanded uncertainty of 0,01% for the arm's length. A level gearbox will be used to minimize this effect, but we have to know the accuracy of the level spirit we will use to measure the levelling angle and the range in which we can do the calibration.



$$L = L_o \cdot cos\theta$$

Eq. 10.17

$$\frac{\Delta L}{L_o} = (1 - cos\theta)$$

Eq. 10.18

$$W_{L,\theta}(\%) = (1 - cos\theta) \cdot 100 < 0{,}005\%$$

Eq. 10.19

$$\theta < 0{,}57°$$

In conclusion, the beam has to be levelled to a horizontal angle less than 0.57º (equivalent to 34'). The spirit level should have an accuracy of at least 10'.

Fig. 10.1 – Spirit level

- ***Fits effect to the length uncertainty:***

The initial length of the lever arm has also an uncertainty due to the clearance of the holes fits. In the adopted resolution we have two fits hole-shaft: the lever arm/shaft fit and the lever arm/hanger support fit. The standards used for the design of the fits are shown in "Annex VI – Fits standards".

Both fits has the same tolerance.

Lever arm/shaft: $45H7^{+0,025}_{-0}\ /h6^{+0}_{-0,016}$

Lever arm/hanger support: $40\ H7^{+0,025}_{-0}/h6^{+0}_{-0,016}$

Maximum clearance = 0,025 + 0,016 = 0,041 mm.

Minimum clearance = 0 mm.

$Average\ clearance = \frac{0,041+0}{2} = 0,0205\ mm.$

Considering that the shaft and the hole have a normal distribution, the fit uncertainty will be:

$$u_{fit\ 1,2} = \sqrt{\left(\frac{0,025-0}{2}\right)^2 + \left(\frac{0,016-0}{2}\right)^2} = \pm 0,0148\ mm.$$

Eq. 10.20

The combined uncertainty for the fits will be:

$$u_{L,fit} = \sqrt{u_1^2 + u_2^2} = \sqrt{0,0148^2 + 0,0148^2} = \pm 0,021\ mm$$

Eq. 10.21

$$U_{L,fit\ 95\%} = \pm 2 \cdot 0,021 = \pm 0,042mm$$

Eq. 10.22

Then, the expanded relative uncertainty due to fits can be calculated as follows.

$$W_{Lo=1.000mm}(\%) = \frac{U_{L,fit}}{Lo} \cdot 100 = \frac{0,042}{1.000} \cdot 100 = 0,0042\%$$

Eq. 10.23

$$W_{Lo=500mm}(\%) = \frac{U_{L,fit}}{Lo} \cdot 100 = \frac{0,042}{500} \cdot 100 = 0,0084\%$$

Eq. 10.24

$$W_{Lo=400mm}(\%) = \frac{U_{L,fit}}{Lo} \cdot 100 = \frac{0,042}{400} \cdot 100 = 0,0105\%$$

Eq. 10.25

- ***Initial length uncertainty***

We will measure the length using a measuring system with an uncertainty of 0,01%, for example a Co-ordinates measuring machine. We will measure the distance between the centre of the lever arm and the centre of the hanger holder.

- ***Length combined uncertainty:***

In Table 10.2 we have a summary of the different effects which contribute to the length uncertainty for the different hang points of the lever arm.

We can calculate the combined expanded uncertainty for the length.

$$U_L = \sqrt{{U_{L,T}}^2 + {U_{L,\theta}}^2 + {U_{L,fit}}^2 + {U_{Lo}}^2} = \sqrt{\left(L \cdot W_{L,T}\right)^2 + \left(L \cdot W_{L,\theta}\right)^2 + \left(L \cdot W_{L,fit}\right)^2 + (L \cdot W_{Lo})^2}$$

$$W_L = \frac{U_L}{L} = \sqrt{{W_{L,T}}^2 + {W_{L,\theta}}^2 + {W_{L,fit}}^2 + {W_{L0}}^2}$$

Eq. 10.26

Length L (mm)	Temperature effect $W_{L,T}$ (%)	Level angle effect $W_{L,\theta}$ (%)	Fits effect $W_{L,fit}$ (%)	Initial length uncertainty W_{Lo} (%)	Length combined uncertainty W_L (%)
1.000	0,002	0,005	0,0042	0,01	0,012
500	0,002	0,005	0,0084	0,01	0,014
400	0,002	0,005	0,0105	0,01	0,015

Table 10.2 - Length uncertainty

Applied torque expanded relative uncertainty

Applying the equation of the expanded relative uncertainty for the applied torque, $W_{T_a} = \sqrt{{W_m}^2 + {W_{g_L}}^2 + {W_L}^2}$, and its relation with the expanded standard uncertainty $U_{T_a} = W_{T_a} \cdot T_a$ we obtain the values shown in Table 10.3.

Length L (mm)	Length rel. uncertainty W_L (%)	Mass rel. uncertainty W_m (%)	Gravity rel. uncertainty W_{gl} (%)	Applied torque rel. uncertainty W_{Ta} (%)
1.000	0,012	0,01	0,0002	**0,0156**
500	0,014	0,01	0,0002	**0,0172**
400	0,015	0,01	0,0002	**0,0180**

Table 10.3 - Applied torque expanded relative uncertainty

10.2.2 Friction torque T_f

As the calibration test bench will be operating stationary (the shaft doesn't rotate) we have to consider the starting torque as the friction torque of the bearings. The starting torque of a rolling bearing is defined as the moment that must be overcome in order for the bearing to start rotating from the stationary condition. Under normal ambient temperature, +20 to +30 °C, starting at zero speed, the starting torque can be calculated using only the sliding frictional moment and the frictional moment of seals, if present. [14] Therefore:

$$M_{Start} = M_{Sl} + M_{Seal}$$

Where:

M_{Start}: Starting frictional moment [Nmm].
M_{Sl}: Sliding frictional moment [Nmm].
M_{Seal}: Frictional moment of the seals [Nmm]. [14]

Eq. 10.27

Our bearing bases are SKF SYT 45 LTS with 22209 spherical rolling bearings. We tried to find their starting torque specifications from the manufacturer. SKF has an interface available in internet to calculate these values. We used it to calculate the starting torque for several load values, then we collected all these values in Table 10.4 and we made a graphic, see Fig. 10.2, in order to have an approximate value for the intermediate load values.

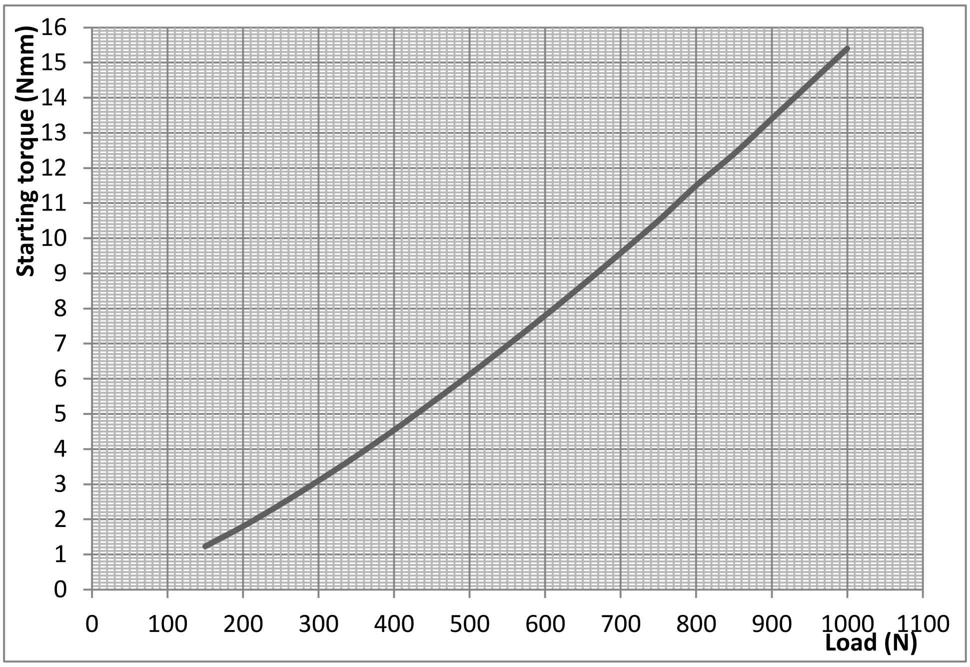


Fig. 10.2- Starting torque graphic for 22209 rolling bearing

F_a (N)	T_f (Nmm)
1.000	15.4
950	14.4
900	13.4
850	12.4
800	11.5
750	10.5
700	9.58
650	8.68
600	7.8
550	6.95
500	6.12
450	5.32
400	4.54
350	3.8
300	3.1
250	2.43
200	1.8
150	1.23
100	0,72

Table 10.4 - Starting torque for 22209 rolling bearings [14]

We are going to use two bearing bases, so the load will be equally distributed between the two. Part of the load will be supported by the universal divisor. We did a simulation in Autodesk Inventor Stress Analysis for the calibration set shaft in order to obtain the reaction forces in the bearings for different rated torques. In order to know the friction torque range for each torque transducer we analysed also the first torque step (10% of the rated torque). The information obtained is collected in Table 10.5 and Table 10.6.

Rated torque (Nm)	Weight's load (N)	Calibration test set weight(N)	Bearings radial load (N)	Bearing's starting torque (Nmm)	Friction torque (Nmm)
1.000	1.000	700	700	9,58	19,16
500 /200	500	500	450	5,32	10,64
100 / 50	100	350	150	1,23	2,46

Table 10.5 - Starting torque for different rated torques

1st step torque (Nm)	Weight's load (N)	Calibration test set weight (N)	Bearings radial load (N)	Bearing's starting torque (Nmm)	Friction torque (Nmm)
100	100	700	300	3,1	6,2
50 /20	50	500	200	1,8	3,6
10 / 5	10	350	100	0,72	1.44

Table 10.6 –Starting torque for different 1st steep torque

Once we know the limits of the friction torque range for the different torque transducers we can estimate an approximated normal distribution for the friction torque T_f with its expanded uncertainty U_{Tf}.

$$\bar{T}_{f,1.000Nm} = \frac{19{,}16 + 6{,}2}{2} = 12{,}68\ Nmm$$

$$u_{Tf,1.000Nm} = \frac{19{,}16 - 6{,}2}{2} = 6{,}48\ Nmm \qquad U_{Tf,1.000Nm} = 2 \cdot 6{,}48 = 12{,}96\ Nmm$$

Eq. 10.28

$$\bar{T}_{f,1.000Nm} = \frac{10{,}64 + 3{,}6}{2} = 7{,}1\ Nmm$$

$$u_{Tf,500/200Nm} = \frac{10{,}64 - 3{,}6}{2} = 3{,}52\ Nmm \qquad U_{Tf,500/200Nm} = 2 \cdot 3{,}52 = 7{,}04\ Nmm$$

Eq. 10.29

$$\bar{T}_{f,100/50Nm} = \frac{2{,}46 + 1{,}44}{2} = 1{,}95\ Nmm$$

$$u_{Tf,100/50Nm} = \frac{2{,}46 - 1{,}44}{2} = 0{,}51\ Nmm \qquad U_{Tf\ 100/50Nm} = 2 \cdot 0{,}54 = 1{,}02\ Nmm$$

Eq. 10.30

10.2.3 Unbalancing torque T_b

The lever arm is not perfectly symmetrical, so there will be an unbalancing torque that will affect the torque measures. To minimize this effect the lever arm has machined canals to locate balancing loads. The lengths available to position the balancing loads are 110mm to 210mm and 250mm to 350mm from the centre in both sides of the lever arm. We can use one or more balancing loads and move them through the canals till the lever arm is balanced.

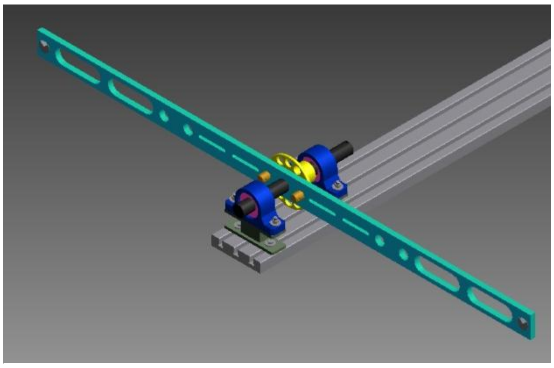


Fig. 10.3 - Lever arm balancing test

To balance the lever arm we have to remove everything from the calibration set except the lever arm, the adapter arm-shaft and the shaft, as we can see in Fig. 10.3. We are going to analyse the case in which we use balancing loads of weight P_b. They produce the maximum balancing torque at 350mm distance from the centre. If we use one weight in each side the minimum balancing torque depends of the location precision we can achieve with the balancing weights. We are going to suppose we can position them in the canals with an accuracy of 1mm. Therefore:

$$T_{b,max} = P_b \cdot 0{,}35m$$

$$T_{b,min} = P_b \cdot 0{,}001m$$

The lever arm can have a maximum unbalance $T_{b,max}$ of $0{,}34 \cdot P_b$. The balancing torque accuracy $T_{b,min}$ we can achieve is $0{,}001 \cdot P_b$. During the balancing test the minimum unbalancing torque we can notice with the level spirit is the starting torque of the bearings for that weight. We can suppose a weight of 200N that will be equally distributed between the two bearings, so the minimum unbalancing torque we can notice is 2x0,72 Nmm = 1,44 Nmm (0,72Nmm friction torque for 100N radial load). We are going to round it to 2 Nmm.

$$P_b < \frac{T_{b\ min}}{1\ mm} = \frac{2\ Nmm}{1\ mm} = 2N$$

We will use 1N balancing loads (≈100 gr). The maximum unbalancing torque using two balancing loads of 1N would be $1N \cdot 0{,}34m = 0{,}36Nm$. If more balancing torque is needed we can add another balancing load with the weight required in one side.

We will suppose for the uncertainty calculations that the unbalancing torque T_b has a normal distribution with mean 0Nm and an expanded uncertainty $U_{Tb\ 95\%}$ = 2 Nmm.

10.2.4 Hanger's unbalance torque

The hangers have a weight of 0,8 kg (7,85 N). We are going to measure them with an uncertainty of 0,1gr, so we obtain an expanded uncertainty for the different torque transducers of: $U_{Th\ 1000Nm\ /\ 500Nm/100Nm} = 0{,}001N \cdot 1m = 0{,}001N$, $U_{Th\ 200Nm} = 0{,}001N \cdot 0{,}5m = 0{,}0005N$ and $U_{Th\ 50Nm} = 0{,}001N \cdot 0{,}4m = 0{,}0004N$.

10.2.5 Transmitted torque expanded relative uncertainty

The transmitted torque to the torque transducer can be calculated using $T = T_a - T_f - T_b - T_h$, with its related expanded uncertainty $U_T = \sqrt{{U_{T_a}}^2 + {U_{T_f}}^2 + {U_{T_b}}^2 + {U_{T_h}}^2}$. We can calculate the expanded relative uncertainty of the transmitted torque for the rated torque of the different torque transducers $W_{T\ (95\%)} = \frac{U_T}{T} \cdot 100\%$. Also we have to check that the first torque steep meets the required specifications.

Torque (Nm)	U_{Ta} (Nm)	U_{Tf} (Nm)	U_{Tb} (Nm)	U_{Th} (Nm)	**U_T (Nm)**	**W_T (%)**
1.000	0,156	0,01296	0,002	0,001	0,157	**0,016**
500	0,0780	0,00704	0,002	0,001	0,078	**0,016**
200	0,0360	0,00704	0,002	0,0004	0,037	**0,018**
100	0,0156	0,00102	0,002	0,001	0,016	**0,016**
50	0,0086	0,00102	0,002	0,0005	0,009	**0,018**

Table 10.7 - Transmitted torque expanded relative uncertainty for rated torque

Torque (Nm)	U_{Ta} (Nm)	U_{Tf} (Nm)	U_{Tb} (Nm)	U_{Th} (Nm)	**U_T (Nm)**	**W_T (%)**
100	0,0156	0,01296	0,0020	0,001	0,020	**0,020**
50	0,00780	0,00704	0,0020	0,001	0,011	**0,021**
20	0,00360	0,00704	0,0020	0,0004	0,008	**0,041**
10	0,00156	0,00102	0,0020	0,001	0,003	**0,029**
5	0,00086	0,00102	0,0020	0,0005	0,002	**0,049**

Table 10.8 - Transmitted torque expanded relative uncertainty for 1st torque steep

We can conclude that the Calibration test bench meets the specifications because the relative expanded uncertainty is better than 0,02% for all the steps of 500Nm calibration and better than 0,04% for all the steps of the rest torque transducers calibration. Note that for 50Nm the first step relative uncertainty exceeds minimally the specifications (0,04%) due to the unbalancing torque error and the friction torque error. As we can see in Table 10.8 the less weight load we hang, the more important is the friction and the unbalancing effect contribution to the expanded uncertainty.

11. Calibration test software

The software to calculate metrological properties of the torque transducers from the calibration test measures was made using Microsoft Excel. This program allows the operator to directly calculate all the important values for the torque transducer calibration having only to enter the different measures and some information about the test conditions. Also this program elaborates a standard calibration report ready to print. The program is divided in five sheets: input, results, calibration report, specifications and calibration process.

Calibration input

In this sheet we have to introduce some initial values, such as the rated torque of the torque transducer to calibrate and the calibration conditions (clockwise/ anti-clockwise calibration and ambient temperature). Also we have to introduce the measured values from each step of the calibration test in the appropriated position.

For each rated torque the program shows the lever arm length and the calibrated weights we have to use for the calibration. The weights have to be applied in the same order as shown in the sheet to have valid results. This program was performed for eight steps calibration test rotating the torque transducer each 120º.

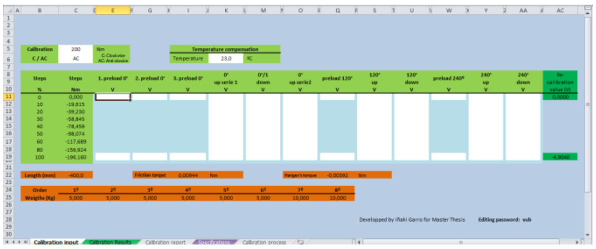


Fig. 11.1 - Calibration input

Calibration results

This sheet shows the results of the calibration test. First we have to introduce the resolution of the indicating device used during the calibration and the regression equation degree (1st, 2nd or 3th degree).

It shows the metrological properties calculated for each torque steep:

- ✓ Mean output.
- ✓ Repeatability.
- ✓ Reproducibility.
- ✓ Residual value at zero torque.
- ✓ Reversibility/ Hysteresis.
- ✓ Deviation from the fitting curve.
- ✓ Resolution.
- ✓ Expanded relative uncertainty.

Fig. 11.2 - Calibration results

Also it shows a graphic of the measured values and the fitting curve, the calculated sensibility of the torque transducer and the regression coefficients for the fitting curve. All the calculations are based on the formulas shown in chapter 14.3 – "Torque transducer metrological properties calculation".

Calibration report

This is a sheet ready to print that collects and order all the information of the calibration test: the input data and the results. It has some gaps to fill with information identifying components used for the calibration in order to generate a standard calibration certificate. The report has made taking into account all we explained in chapter 4.11 – "Calibration certificate".

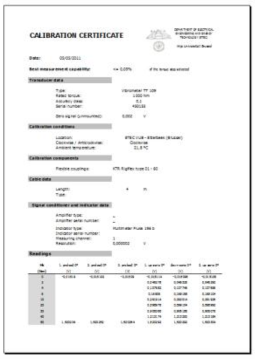


Fig. 11.3 - Calibration report

Specifications

In this sheet we have the specifications of the Calibration Bench: the length measures of the different hang points, the calibrated weights values, the local gravity value, the linear expansion coefficient of the lever arm, the buoyancy effect, the friction torque and the torque applied uncertainty. This data is used by the program for the calculations. It shouldn't be modified unless change of any of the characteristics of the Calibration Bench.

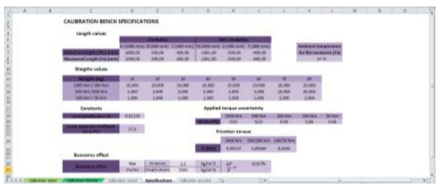


Fig. 11.4 - Calibration program. Specifications

Calibration process

In this sheet all the steps of the calibration process are explained as in chapter 4 – "Calibration process", in order to help the operator during the calibration test.

The program was tested with an example obtained from the EA guide for a 50Nm torque transducer [5]. We checked that we obtain the same results using the input values provided by the example.

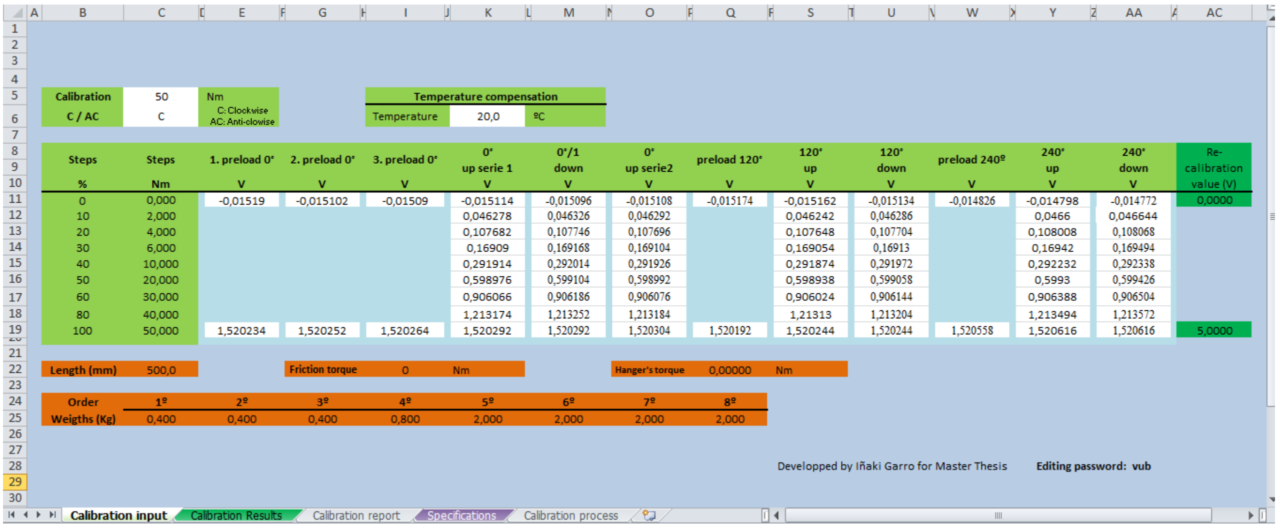


Fig. 11.6 - Calibration program example for 50Nm. Input

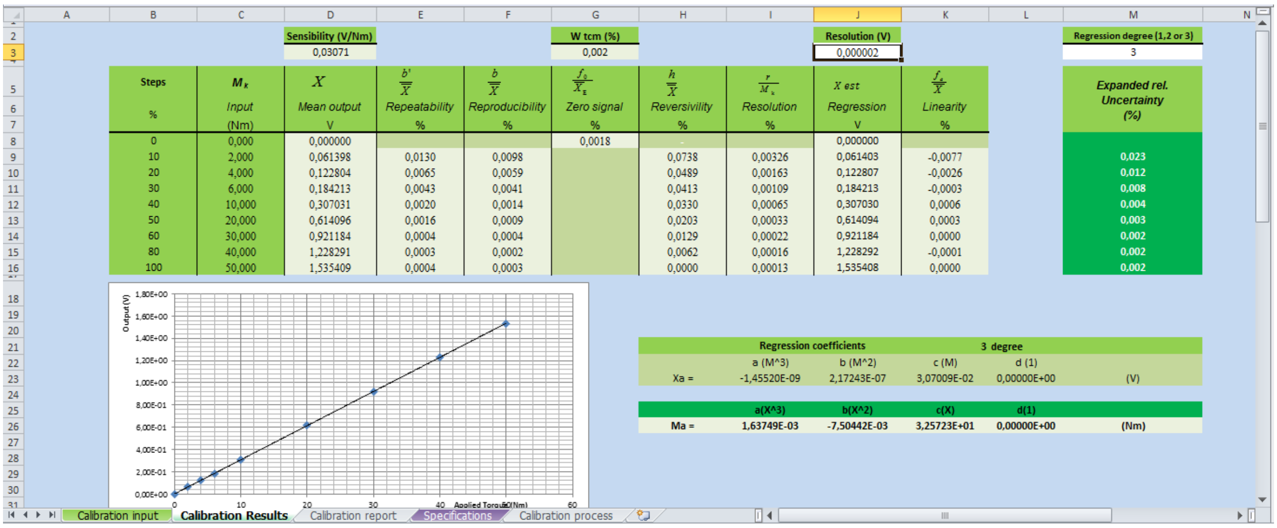


Fig. 11.5 - Calibration program example for 50Nm. Results

12. Conclusions

We have achieved the different goals we proposed at the beginning of the thesis:

- ✓ We have designed a Test bench to calibrate 200Nm and 500Nm torque transducers with the required accuracy. We obtained an accuracy of 0,016% for the rated torque of 500Nm torque transducers, whom best accuracy is 0,1%, and 0,018% for the rated torque of 200Nm torque transducers, whom best accuracy is 0,2%. We achieved the EA requirements for the accuracy of the transmitted torque for 0,1 and 0,2 accuracy class transducers, which is 0,02% and 0,04% respectively.

- ✓ For most of the components of the bench we used the material existing in the ETEC laboratory. Finally we designed few new components, which are interchangeable due to the standardization of their dimensions. We checked the validity of their dimensions with the stress analysis simulator of Autodesk Inventor.

- ✓ We have designed a versatile bench that can be used in the future for 50Nm, 100Nm and 1.000Nm torque transducers calibration adding some adaptors.

- ✓ Also we made an Excel sheet to automatically calculate the metrological properties of the torque transducers during the calibration test based on the EA Guidelines.

This thesis can be a consultation guide for the people who will continue with the construction of the "Test bench for static calibration of torque transducers".

13. Future works

In this thesis we have design a Test bench for static calibration torque transducers. Next step is to make the different components, check that the different parts fit and assembly the bench.

The points to hang the weights have to be measured with an accuracy of 0,01%, the weights and the hangers have to be calibrated with a accuracy of 0,01%. The Specifications sheet of the Excel calibration test program has to be filled with these measured values. The ambient temperature for the length measures and the buoyancy effects have to be taken into account too.

Some tests have to be done to check that the bench behaves as it was designed. Firstly the friction torque value has to be checked, and then the lever arm has to be balanced. Later a calibration test has to be performed with any of the torque transducers in order to check the output is as expected. Finally the calibration test software has to be checked. Once all the verifications have been done, we can start the calibration tests for the different torque transducers.

If the calibration test bench is going to be used for 1.000Nm, 100Nm or 50Nm torque transducers new adapters have to be designed. Preferably a new shaft and a new Adapter to rotatory table have to be designed for 100Nm/50Nm and others for 1.000Nm torque transducers. Also a new set of weights is needed for 50/100Nm and another for 1.000Nm calibration.

14. Annexes

14.1 Annex I: Torque transducers

14.1.1 Torque measurement

Torque, speed and power are the defining mechanical variables associated with the functional performance of rotating machinery. The ability to accurate the measure of this quantities is essential for determining a machine's efficiency and for establish operating regimes that are both safe and conductive to long and reliable services. On-line measurements of these quantities enable real-time control, help to ensure consistency in product quality and can provide early indications of impending problems.

In general, a driving torque originating within a device at one location (B) is resisted by an opposing torque developed by a different device at another location (F). The driving torque (from, e.g., an electric motor, a gasoline engine, a steam turbine, muscular effort, etc.) is coupled through connecting members (C), transmitting region (D) and additional couplings (E) to the driven device (an electric generator, a pump, a machine tool, mated threaded fasteners, etc.) within which the resisting torque is met at F.

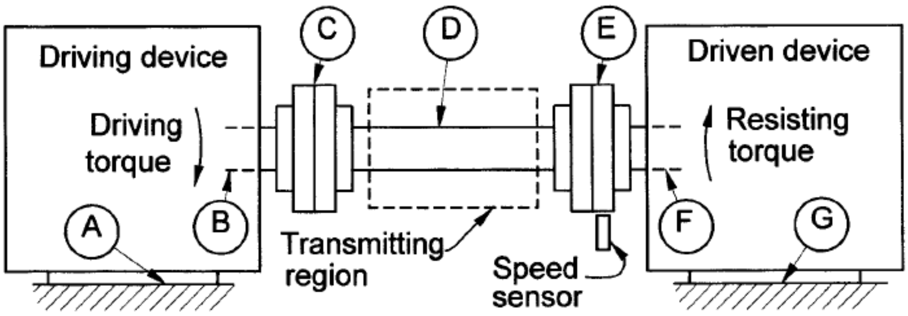


Fig. 14.1 - Schematic arrangement of devices used for the measurement of torque and power [15]

The torque at B or F is the quantity to be measured. These torques may be indirectly determined from a correlated physical quantity, e.g., an electrical current or fluid pressure associated with the operation of the driving or driven device, or more directly by measuring either the reaction torque at A or G, or the transmitted torque through D. Reaction torque at A or G is often determined from measurements of the forces acting at known distances fixed by the apparatus. [16]

Transmitted torque is usually determined from measurements, on a suitable member within region D, of τ_m, γ_m, or ϕ and applying the following equations:

$$\tau_m = \frac{16T}{\pi d^3}$$

Eq. 14.1 [16]

$$\gamma_m = \frac{\tau_m}{G} = \frac{16T}{G\pi d^3}$$

Eq. 14.2 [16]

$$\phi = \frac{32LT}{\pi d^4 G}$$

Eq. 14.3 [16]

When the angular acceleration (α) is nonzero and it's measurable, torque (T) may also be determined from $T = I \cdot \propto$, being "I" the inertia of the studied element. Requiring only noninvasive observational measurements, this method is especially useful for determining transitory torques. [16]

Instruments, variously called sensors or transducers, are used to convert the torque quantity into a linearly proportional electrical signal.

-*Sensor:* is a device which detects a change in a physical stimulus and turns it into a signal which can be measured or recorded.

-*Transducer*: is a device that transfers power from one system to another in the same or in the different form.

As seen, the torque transducers we are going to study will have to be fed with an electrical signal to provide us an output signal proportional to torque.

14.1.1.1 Static/ Dynamic measurement

We can differentiate two methods to measure torque:

-*Static torque measurement*: when the torque remains virtually unchanging. It is considered static if it has no angular acceleration.

-*Dynamic torque measurement:* when there are more rapid variations of torque. Torque is calculated applying Eq. 14.4.

$$T = I \cdot \propto_c$$

Eq. 14.4 [1]

T: torque, I: inertia, α: angular acceleration.

Also, for dynamic torque measurement, we have to consider the dynamic effects, such as the natural frequency (fo), which can be calculate applying Eq. 14.5:

$$fo = \frac{1}{2\pi}\sqrt{Kt\frac{J_1 + J_2}{J_1 \cdot J_2}}$$

Eq. 14.5 [1]

K_t: torsional stiffness.
J_1: moment inertia (driving machine + coupling + ½ measuring shaft).
J_2: moment inertia (driven machine + coupling + ½ measuring shaft). [1]

14.1.2 Operating principle

As seen before, torque transducers convert an input signal into an output electrical signal proportional to torque. Various physical interactions serve to convert a proportional value from torque, such as surface strain (γ_m), Twist angle (ϕ) or Stress (τ_m), into proportional electrical signals.

Surface strain γm

This method consist in convert the surface strain of the transmitting region, which is proportional to torque as seen in Eq. 14.2, to a proportional electrical output. Fig. 14.2 illustrates a sensing region configured to convert surface strain (γ_m) into an electric signal proportional to the transmitted torque using *strain gauges*. Strain is sensed as a change in gauge resistance. These changes are generally too small to be accurately measured directly and so it is common to employ two to four gages arranged in a Wheatstone bridge circuit.

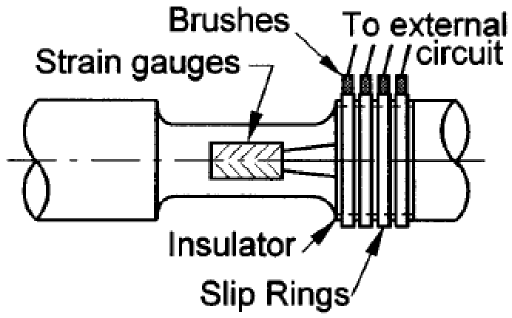


Fig. 14.2 -. Surface strain conversion method.

Strain gauges take advantage of the physical property of electrical resistance and its dependence on not merely the electrical conductivity of a conductor, which is a property of its material, but also the conductor's geometry. When an electrical conductor is stretched within the limits of its elasticity such that it does not break or permanently deform, it will become narrower and longer, changes that increase its electrical resistance end-to-end. Eq. 14.6 shows the change of the strain gauge resistance depending of its deformation in one direction. Based [16]

$$R = R_0 \cdot (1 + K\varepsilon)$$

Eq. 14.6 [16]

R: resistance of the deformed gauge, R_0: resistance of the undeformed gauge, K: strain gauge's sensibility, ε: deformation of the gauge.

Independence from axial and bending loads as well as from temperature variations are obtained by using a four-gauge Wheatstone bridge comprised of two diametrically opposite pairs of matched strain gauges, each aligned along a principal strain direction. In round shafts (and other shapes used to transmit torque), tensile and compressive principal strains occur at 45° angles to the axis. [16]

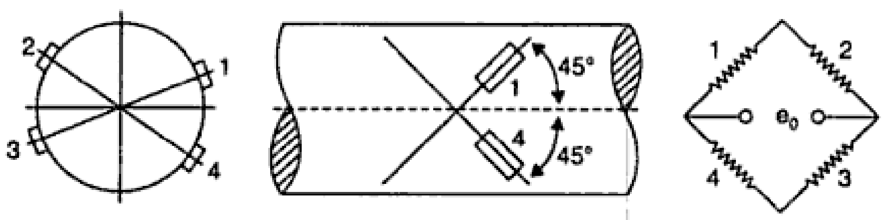


Fig. 14.3 - Torque measurement using strain gages [1]

Using a four-gage Wheatstone bridge we obtain an output electric signal directly proportional to torque as we can demonstrate with the following equations.

$$\frac{E_o}{E_i} = \frac{R(\epsilon) - R(-\epsilon)}{R(\varepsilon) + R(-\varepsilon)} = \frac{R_0 \cdot (1 + K\varepsilon) - R_0 \cdot (1 - K\varepsilon)}{R_0 \cdot (1 + K\varepsilon) + R_0 \cdot (1 - K\varepsilon)} = K\varepsilon = \frac{K \cdot \gamma_{max}}{2} = \frac{8 \cdot K \cdot T}{G \cdot \pi \cdot d^3}$$

Eq. 14.7

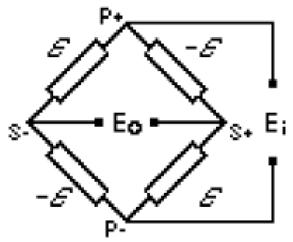


Fig. 14.4 - Four-gage Wheatstone bridge.

$$G = \frac{E}{2(1 + \vartheta)}$$

Eq. 14.8

T: torque, G: shear modulus, E: elastic modulus, υ: Poisson's coefficient, K: strain gauge's sensibility, d: diameter.

Typical practice is to increase the compliance of the sensing region (e.g., by reducing its diameter or with hollow or specially shaped sections) in order to attain the limiting strain at the highest value of the torque to be measured. This maximizes the measurement sensitivity. [16] There are various configurations: like solid circular, hollow circular, cruciform, hollow cruciform, solid-square and hollow tube with flats, which are shown in Fig. 14.5. [17]

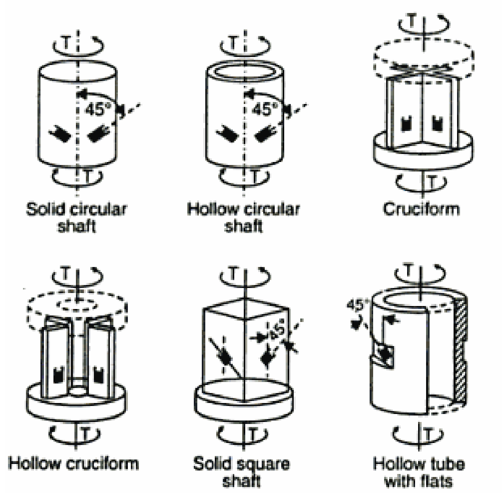


Fig. 14.5 - Sensing region designs [17]

Twist angle ϕ

Torque induces a proportional twist angle in the element to which it's applied as seen in Eq. 14.3. Measuring this angle we can obtain the applied torque. If the shaft is slender enough (e.g., L > 5 d) the twist angle ϕ , at limiting values of stress T_m for typical shaft materials, can exceed 1°, enough to be resolved with sufficient accuracy for practical torque measurements.

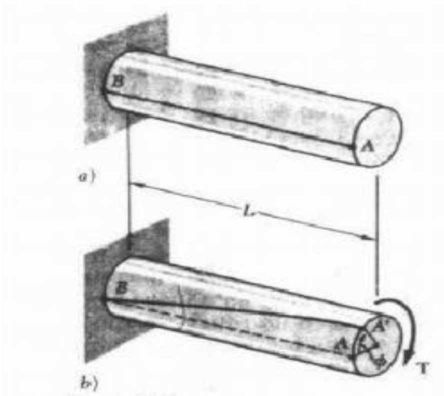


Fig. 14.6 - Twist angle.

Fig. 14.7 shows a common arrangement wherein torque is determined from the difference in tooth-space phasing between two identical "toothed" wheels attached at opposite ends of a compliant "torsion bar." The phase displacement of the periodic electrical signals from the two "pickups" is proportional to the twist angle of the torsion bar and thus to the torque.

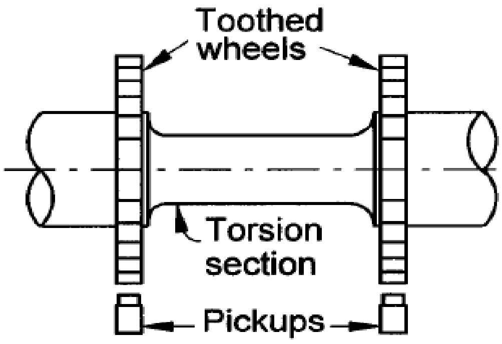


Fig. 14.7 - Pick-ups measuring system. [16]

$$\phi = \frac{32LT}{\pi d^4 G}$$

Eq. 14.9 [16]

In other constructions, a shaft-mounted variable displacement transformer or a related type of electric device is used to provide speed independent output signals proportional to ϕ. [16]

Stress T_m

In addition to elastic strain, the stresses by which torque is transmitted are manifested by changes in the magnetic properties of ferromagnetic shaft materials. This "magneto-elastic interaction" provides an inherently non-contacting basis for measuring torque.

Two types of magneto-elastic (sometimes called Magneto-strictive) torque transducers are in present use:

-*Type 1*: these transducers derive output signals from torque-induced variations in magnetic circuit permeances. They typically employ branch, cross or solenoidal constructions.

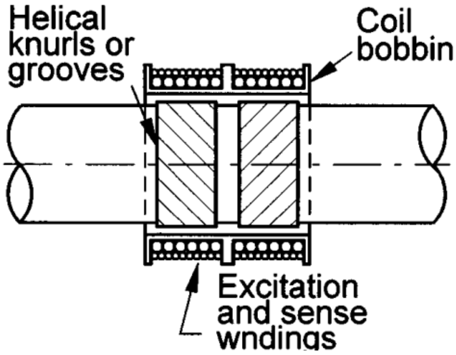


Fig. 14.8. - Magneto-elastic torque transducer type 1 [16]

In branch and cross designs, torque is detected as an imbalance in the permeabilities along orthogonal 45° helical paths (the principal stress directions) on the shaft surface or on the surface of an ad-hoc material attached to the shaft. In solenoidal constructions torque is detected by differences in the axial permeabilities of two adjacent surface regions, pre-endowed with symmetrical magnetic "easy" axes (typically along the 45° principal stress directions).

-*Type 2:* these transducers create a magnetic field in response to torque. They are generally constructed with a ring of magneto-elastically active material rigidly attached to the shaft.

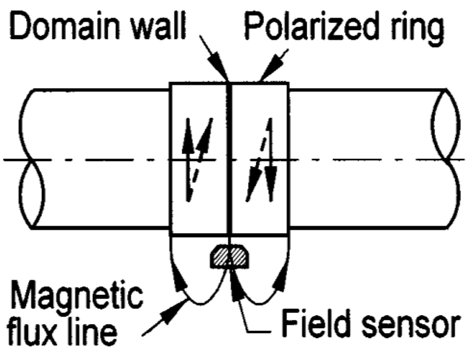


Fig. 14.9. - Magneto-elastic torque transducer type 2 [16].

The ring is magnetized during the manufacture of the transducer, usually with each axial half polarized in an opposite circumferential direction as indicated by the solid arrows in Fig. 14.9. When torque is applied, the magnetizations tilt into helical directions (dashed arrows), causing magnetic poles to develop at the central domain wall and (of opposite polarity) at the ring end faces. Torque is determined from the output signal of one or more magnetic field sensors (e.g., Hall Effect, magneto-resistive, or flux gate devices) mounted so as to sense the intensity and polarity of the magnetic field that arises in the space near the ring. [16]

14.1.3 Mechanical considerations

Transducers are available with capacities from 0,001 Nm to 200 kNm. Maximum operating speeds vary widely; upper limits depend on the size, operating principle, type of bearings, lubrication, and dynamic balance of the rotating assembly. Forced lubrication can allow operation up to 80,000 rpm. High-speed operation requires careful consideration of the effects of centrifugal stresses on the sensed quantity as well as of critical (vibration inducing) speed ranges. [16]

14.1.3.1 Construction types

There are different torque transducer construction types depending on the couplings they need to be attached to the driven/driving device.

Flange	
Shaft: key end	
Shaft: hexagonal drive	
Square drive	

Table 14.1 - Torque transducers construction types [18]

14.1.3.2 Couplings

Couplings are used to connect two shafts in order to transmit torque between them or to connect the torque transducer with the driving/driven device shaft. There are several types of couplings depending of the joint type and the system operation type. Below we explain the most common used couplings.

Rigid couplings

These couplings join rigidly both axes. We choose this junction if the axes are exactly in line and there is no bending or axial movements. The rigid connection applies in all cases where we must measure the dynamic changes in torsion, while flexible couplings amortize these small variations. [1]

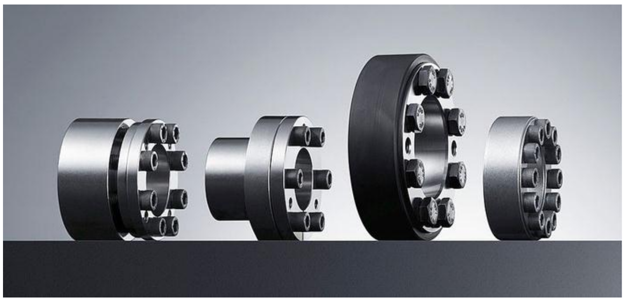


Fig. 14.10 - Rigid couplings [19]

Flexible couplings

Flexible couplings are used if there is a slight difference between the trees or the axes are not in line. They are available from very small to very high rotational speeds.

Due to products of different manufacturers have different characteristics it isn't possible to specify the angular deflection and the offset maximum permissible. The flexible couplings avoid measuring changes in torsion. [1]

In most cases it is recommended to install the torque transducer between flexible couplings, which have axial play in them, to protect the transducer from damage. [17]

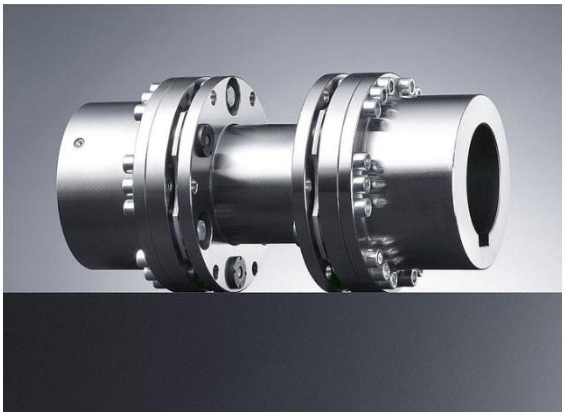


Fig. 14.11 - Flexible disc coupling [19]

Curved teeth couplings

This coupling is a combination of rigid and flexible couplings for linking trees with a very small radial clearance. These couplings have curved teeth which are used for measuring changes in torsion. When using these couplings, bushings junction must be screwed on both sides of the sensor shaft. [1]

Fig. 14.12 - Curved teeth couplings [20]

Cardan couplings

Besides the characteristics of the flexible coupling, Cardan couplings have the advantage of being free of radial clearance. To mount them, it is generally necessary to provide additional support for the torque transducer. [1]

Fig. 14.13 Cardan couplings [20]

14.1.3.3 Assemblies

There are different ways to assembly the torque transducer to the driving/driven machine and to the frame depending of the application. For high-speed applications, fixed/supporting mounting is mandatory. In all supported installations, double-element or metal bellows couplings are recommended.

For safety reasons, suspended mounting should only be used for low-speed applications. Single-element couplings (selected in relation to torque, speed and transducer weight) are then employed to create a shorter drive train. Bellows couplings and double-element couplings are not suited for suspended (floating) installations.

Suspended installation

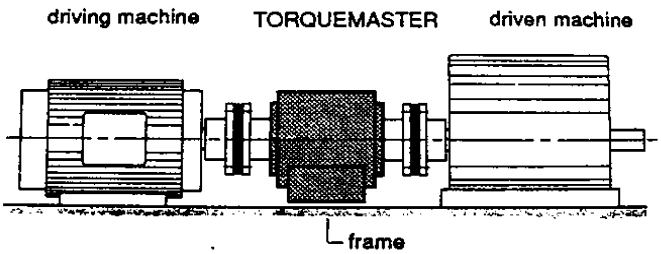


Fig. 14.14 - Suspended installation [1]

Advantages:

- Cheaper (no pedestal, single-element couplings).
- Shorter installation.

Disadvantages:

- Increased shaft and consequently reduce critical speed due to bending.
- Lack of transducer support makes changing of associated machinery more difficult. [1]

Pedestal installation:

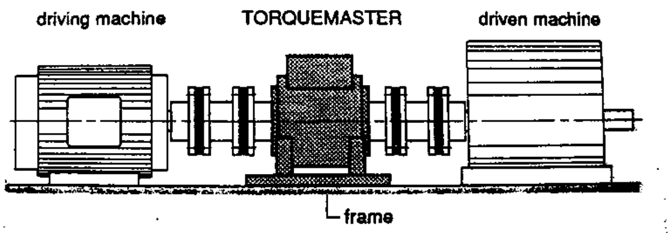


Fig. 14.15.- Pedestal installation [1]

Advantages:

- Higher critical speed due to less bending.
- Support makes changing of associated machinery easier to accomplish.

Disadvantages:

- Higher price (pedestal, double-element coupling required).
- Longer installation (double-element couplings). [1]

Bearing-less torque transducer:

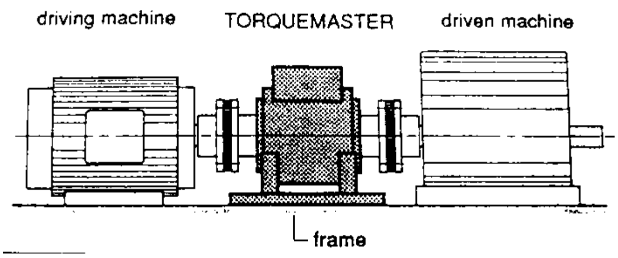


Fig. 14.16 - Bearing less torque transducer [1]

The transducer housing with its pedestal has purely a measuring function and does not carry the measuring shaft. The latter is supported by the machine shafts via couplings.

Advantages:

- Very suitable for extremely high speeds due to the short torque measuring shaft and resultant increase of stiffness.
- No possibility of bearing damage due to vibration or misalignment.

Disadvantages:

- Painstaking installation necessary to achieve minimal eccentricity between shaft and housing.
- Relative axial and radial displacements must not exceed specified limits.
- Increased possibility of damage caused by non-expert mounting. [1]

14.1.4 Electrical considerations

Transducers require some electrical input power or excitation. They usually generate 1 or 2.5 mV of output per volt of excitation, so they also generally require conditioning into a level and format appropriate for display on a digital or analog. Excitation and signal conditioning are supplied by electronic circuits designed to match the characteristics of the specific sensing technology. [16]

The schematic diagram of a sensor with non-contacting power supply and signal pick off is shown in Fig. 14.17. The bridge supply is a constant amplitude high-frequency sine wave, and the output is a sine wave of the same frequency but whose amplitude is a function of torque. Consequently, the power supply electronics must include an oscillator to generate the carrier frequency. The output electronics includes a demodulator to produce a DC signal in proportion to the peak of the output sine wave. [17]

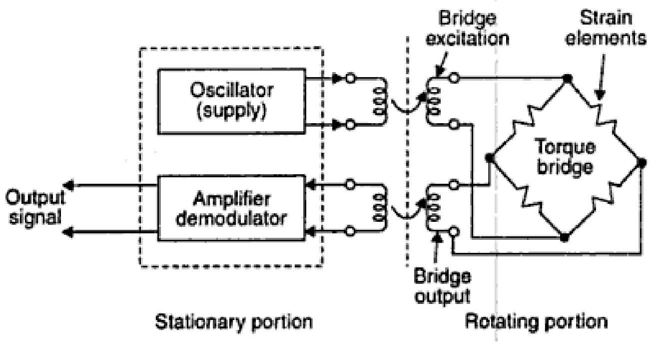


Fig. 14.17 - Torque transducer electrical system [17]

For example, strain gage bridges are typically powered with 10 V to 20 V (dc or ac) and have outputs in the range of 1.5 mV to 3.0 mV per volt of excitation at the rated load. Raising these millivolt signals to more usable levels requires amplifiers having gains of 100 or more. With AC excitation, oscillators, demodulators (or rectifiers) are also needed.

Strain gages, differential transformers, and related sensing technologies require that electrical components be mounted on the torqued member. Bringing electrical power to and output signals from these components on rotating shafts require special methods. [16] Flexible cabling minimizes incidental torques and makes for a long and reliable service life. All such wiring considerations are avoided when noncontact technologies are used. [15]

We can divide the methods to bring the electrical power to the torque transducers and to obtain the output from them in two main categories: contacting methods and non-contacting methods.

14.1.4.1 Contacting methods

14.1.4.2 Slip rings

The most direct and common approach is to use conductive means wherein brushes (typically of silver graphite) against (silver) slip rings. They are an economical solution that performs well in a wide variety of applications.

At low to moderate speeds, the electrical connection between the rings and brushes are relatively noise free, however at higher speeds noise will severely degrade their performance. They have low speed limits. Rings wear and dust generated by the brushes can quickly impede signal transfer, so you must routinely maintain the rings and the brushes to ensure clean signal transfer. [16] Also the friction produces heating of the transducer, which may affect the measurement. The maximum speed they can reach is usually 6000 rpm. [17]

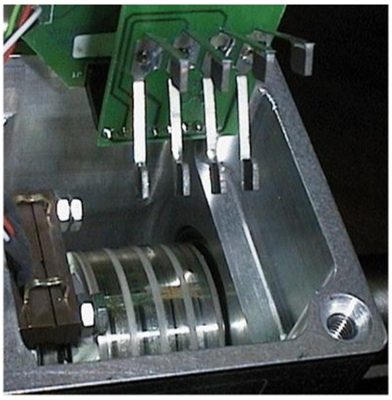


Fig. 14.18 - Slip rings.

14.1.4.3 Non-contacting methods

Several non-contacting methods are also used. For example, power can be supplied via inductive coupling between stationary and rotating transformer windings, by the illumination of shaft mounted photovoltaic cells, or even by batteries strapped to the shaft (limited by centrifugal force to relatively low speeds). Output signals are coupled off the shaft through rotary transformers, by frequency-modulated (infrared) LEDs or by radio-frequency (FM) telemetry. Where shaft rotation is limited to no more than a few full rotations, as in steering gear, valve actuators or oscillating mechanisms, hard wiring both power and signal circuits is often suitable. [16]

Rotatory transformer

Compared to the slip ring, the rotary transformer method tolerates higher speed, is non-contact and typically more accurate. However, it is less tolerant to extraneous loading conditions like bending moments and thrust loads. It also requires more sophisticated signal conditioning instrumentation using an AC carrier excitation. [16]

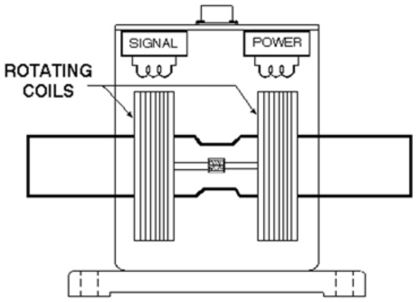


Fig. 14.19 - Rotatory transformer [16]

Analog telemetry

Analog FM wireless telemetry systems use a sensor with a built-in radio transmitter module, a power supply and a receiver. Low voltage signals from the strain gauges are amplified and modulated to a radio frequency signal by the transmitter. This radio signal is picked up by a hoop antenna and decoded into analog voltage by the receiver.

While this form of telemetry is an improvement over other mechanical methods, it is bulky and requires additional receivers for multiple channels. [16]

Fig. 14.20 - Analogue telemetry

Digital telemetry

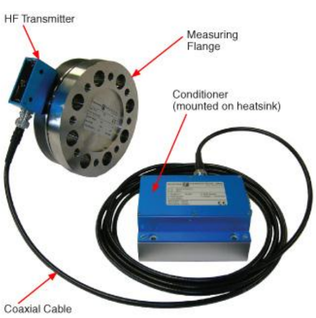


Fig. 14.21 - Digital telemetry

A rotor electronics circuit board module is embedded in the sensor. Signal conditioning and digitizing is done on the rotating sensor using this module.

Resolution, stability and accuracy are all improved over the early digital telemetry systems. The new systems transfer digital data at very high speeds, providing a frequency response up to 3,000 Hz. [16]

14.1.4.4 Power input

Power can be supplied several ways depending on the application. In the following paragraphs we do an overview of the characteristics of each method:

Standard Battery Powered Systems

Power is supplied by standard batteries, which can be purchased practically anywhere. An auto-on option is available, which activates the transmitter when a slight vibration is sensed. When motion ceases the transmitter is turned off after a factory set period of time.

Rechargeable Li-Ion Powered Systems

Power is supplied by a Li-Ion battery pack. A Power Control Module is built into the transducer, which allows for in-collar recharging. The transmitter can be turned on and off remotely from the receiver.

Inductively Powered Systems

Power is supplied to the transmitter inductively for continuous, non-interrupted measurements. No batteries are required. [21]

14.1.4.5 Output display equipment

Electronic circuitry falls broadly into two types, analogue and digital, with most electronic measurement systems comprising a mixture of the two. There are also whole analogue electronic systems, but these are rare in torque measurement. Most systems start with an analogue signal. The point at which the signal is converted defines the type.

- *Analogue systems:* one in which the signal is processed before being converted to digital.
- *Digital systems:* the original analogue signal is converted to digital before processing it. [4]

14.2 Annex II: Torque transducer characteristics

In this chapter we are going to define the different terms that commonly appear in the specification sheets of torque transducers. We are going to divide the in: Ambient conditions and load limits, and Metrological properties.

14.2.1 Ambient conditions and load limits

- Rated speed.
- Rated torque.
- Maximum service torque.
- Limit torque.
- Breaking torque.
- Permission oscillation bandwidth.
- Axial limit force.
- Lateral limit force.
- Bending limit moment.
- Reference temperature.
- Nominal temperature range.
- Service temperature range.
- Storage temperature range.

14.2.1.1 Rated speed

The rated speed is the upper limit of the revolution range starting from zero. It applies both directions. [22]

14.2.1.2 Rated torque

Rated torque is the torque that defines the upper limit of the range within which not exceed the tolerances specified in the properties of the transducer. [22]

14.2.1.3 Maximum service torque

The maximum torque of service is the upper limit of the range in which there is a clear relation between the output and torque. Above the rated torque, however, does not have to keep the tolerances indicated in the specifications.

If the transducer has been used between the rated and maximum torque of service, it keeps the limits indicated in the specifications when it is used again within the nominal torque. It might be produced a slight deviation from the zero signal, which is not regarded as a violation of the specifications. The torque transducer can be used for measurements up to the maximum torque of service, though it must be assumed that the technical properties may be disadvantaged.

The maximum torque limit of service may result from electronic properties (such as the field modulation of the internal electronics of the amplifier) or mechanical (such as an overload limit). In the case of transducers that have neither internal electronic, nor mechanical protection against overload, the service torque and the torque limit are often identical. [22]

14.2.1.4 Limit torque

The limit torque is the torque to which the measurement characteristics of the transducer do not suffer any variation.

If the transducer has been used between the nominal torque and the torque limit, it keeps the limits indicated in the specifications when used again within the nominal torque. It may be produced a slight deviation from the zero signal, which is not regarded as a violation of the specifications. [22]

14.2.1.5 Breaking Torque

It is the torque, which if exceeded, may cause mechanical destruction.

Intermediate torque values between limit torque and breaking torque don't cause any mechanical destruction. However, they can cause damages to the transducer, which ultimately could disable it permanently. [22]

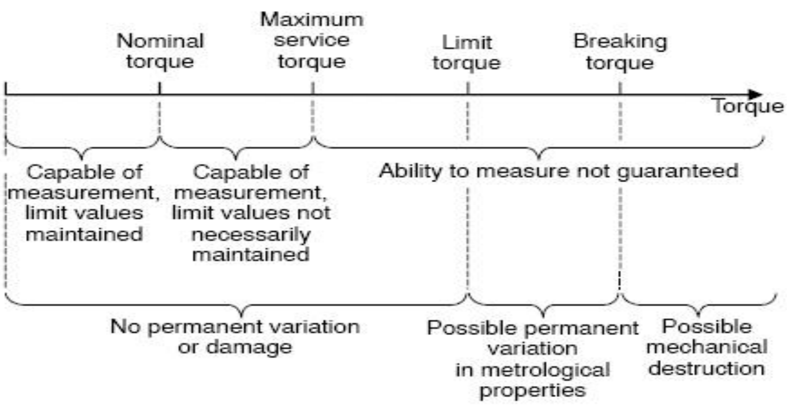


Fig. 14.22 - Torque range. [22]

14.2.1.6 Permission oscillation bandwidth

The permission oscillation bandwidth of a sinusoidal variable torque is the vibration amplitude that resists the transducer under a load of 10^6 cycles without causing any significant change in their properties.

The amplitude is designated as peak-peak (the difference between maximum and minimum torque). [22]

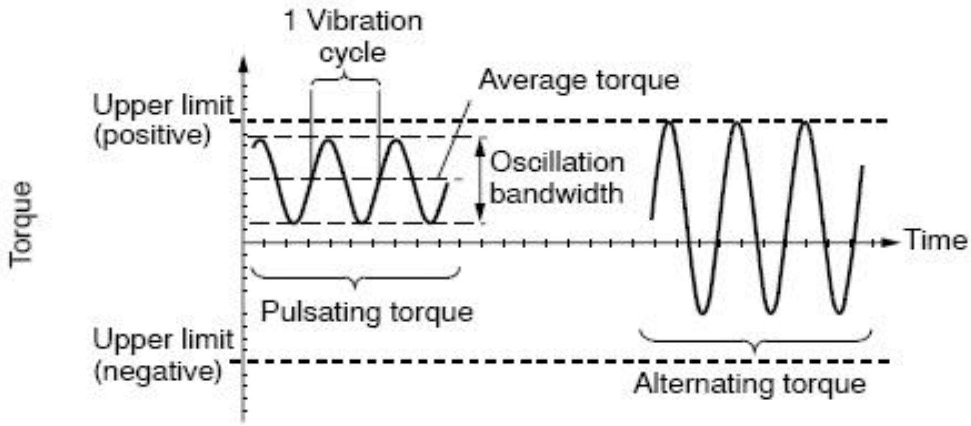


Fig. 14.23 - Permission oscillation bandwidth graphic. [22]

14.2.1.7 Axial limit force

The axial limit force is the maximum allowable longitudinal force (or axial force), defined as 'F_a' in Fig. 14.24. If this axial force limit is exceeded, the transducer may be damaged permanently.

Permissible axial force must be reduced relative to the axial force limit indicated if there is another irregular effort acting simultaneously, such as bending moment, shear force or exceeding the rated torque. Otherwise, the limits should be reduced. For example, if given the 30% limit flexion torque and 30% limit lateral force, the maximum permissible axial force is only 40%, not being able to exceed the rated torque. [22]

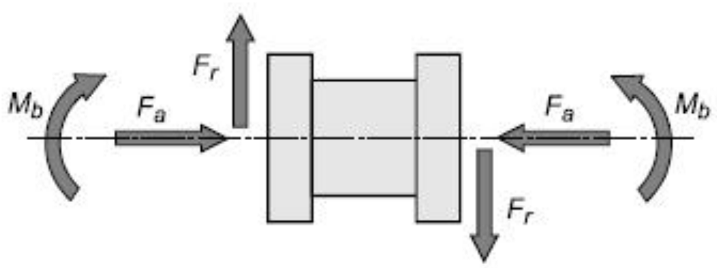


Fig. 14.24 - Parasite loads: axial force Fa, lateral force Fr, bending moment Mb. [22]

14.2.1.8 Lateral limit force

The lateral force limit is the maximum allowable shear force (or radial force), defined as ‘F_r’ in Fig. 14.24. If the shear force limit is exceeded, the transducer may be damaged permanently. The allowable shear force must be reduced relative to limit lateral force indicated if there is another effort acting simultaneously, such as axial force, bending moment or exceeding the rated torque. Otherwise, the limits should be reduced. [22]

14.2.1.9 Bending limit moment

Flexion torque limit is the maximum bending moment, defined as ‘Mb’ in Fig. 14.24. If this bending limit is exceeded, the transducer may be damaged permanently. The bending moment allowed has to be reduced from the bending limit moment indicated if there is effort acting simultaneously, such as axial force, shear force or exceeding the rated torque. If it is the case, then the limit values should be reduced [22]

14.2.1.10Reference temperature

The reference temperature is the temperature to which the specifications of the transducer refer, provided that you don’t apply any range of temperatures. [22]

14.2.1.11Nominal temperature range

The nominal temperature range is the ambient temperature range in which the transducer can be operated for all practical applications and in which the limit values for the metrological properties listed in the specifications are guaranteed. [22]

14.2.1.12Operating temperature range

The operating temperature range is the ambient temperature range in which the transducer can be operated without permanent alteration of its metrological properties. Within the operating temperature range, but outside the nominal temperature range, there is no guarantee to maintain the limit values of the metrological properties listed in the specifications. [22]

14.2.1.13Storage temperature range

The storage temperature range is the ambient temperature range in which the transducer can be stored without mechanical or electrical load without permanent alteration of its metrological properties. [22]

14.2.2 Metrological properties of the torque measurement system

- Accuracy class
- Sensitivity C
- Nominal sensitivity C_{NOM}
- Sensitivity tolerance D_C
- Temperature effect on the sensitivity TK_C
- Temperature effect on the zero signal TK_0
- Linearity deviation D_{LIN}
- Relative reversibility error D_{HY}
- Linearity deviation including hysteresis D_{LH}
- Hysteresis error
- Relative standard deviation of reproducibility

14.2.2.1 Accuracy class

The accuracy class indicates that most of the individual deviations is less than or equal to the value of accuracy class. The tolerance of the nominal value is not included. The accuracy class includes the following measurement characteristics explained below:

- Linearity deviation including hysteresis (dlh)
- Relative standard deviation of reproducibility (σ_{rel})
- Effect of temperature (10 K) on the zero signal (TK_0)
- Effect of temperature (10 K) on the face value (TK_C)

For transducers with multiple electrical outputs (output voltage and frequency) the accuracy class is defined as the output with the highest accuracy. Not to be confused with total accuracy in practical use, in which all the influences affecting individual at the same time. [22]

14.2.2.2 Sensitivity C

The margin between the output signal at nominal torque and torque is zero. Torque transducers commonly have two independent sensibility values: one for clockwise rotation and one for counter clockwise rotation.

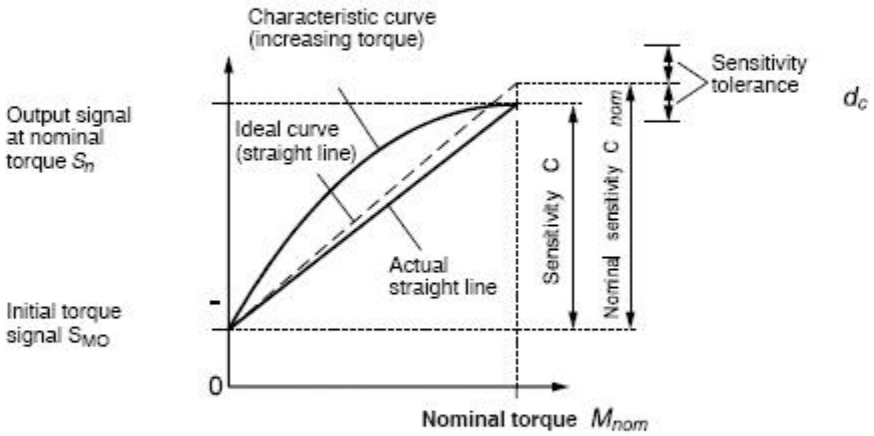


Fig. 14.25 - Sensitivity graphic. [22]

The sensitivity value 'C' defines the slope of the characteristic curve. The *characteristic curve* is also defined as the line connecting the output signal before applying a load and the output signal with nominal torque. [22]

14.2.2.3 Nominal sensitivity C_{NOM}

Normally, the nominal value is the same for torques in both rotatory directions. The nominal value is a characteristic value depending on the type and measuring range of the transducer. The characteristic value of the individual corresponds to the nominal value within a tolerance margin. [22]

14.2.2.4 Sensitivity tolerance D_C

The sensitivity tolerance is the allowed deviation between the effective sensitivity value and the rated value. This tolerance is indicated in per cent in relation to the rated value. [22]

14.2.2.5 Temperature effect on the sensitivity TK_C

The temperature effect on the sensitivity (also sensitivity temperature coefficient) is an index of the influence of temperature on the output signal with a load on the transducer. The starting torque value with the same temperature must be subtracted from the output signal. A stationary temperature should be established. The temperature effect on the characteristic value produces a change in the slope of the curve. [22]

14.2.2.6 Temperature effect on the zero signal TK_0

The temperature effect on the zero signal (also: zero coefficient of zero signal) is determined by measuring the change of the output signal free of load caused by a change in temperature of 10 K, after having established a stationary state temperature. In this case, the critical temperature is the temperature of the transducer. By a defect or a zero balance in the operating temperature of the transducer, you can eliminate the measurement error due to temperature effect on zero signals.

As we see in the temperature effect on zero signals produces a parallel shift of the characteristic curve. The temperature effect on the zero point and the temperature effect on the characteristic value overlap. [22]

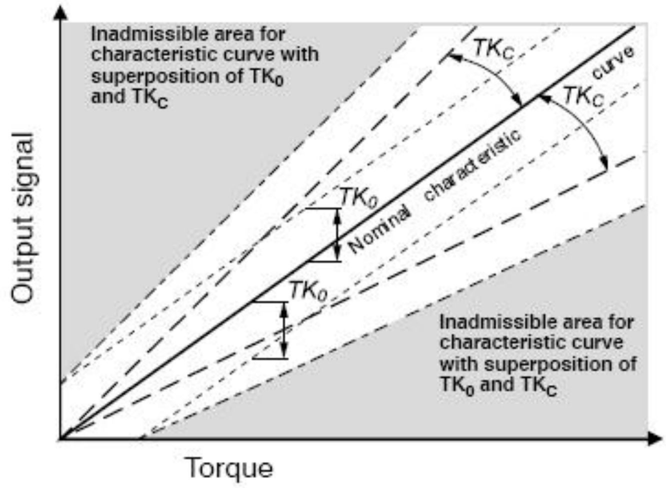


Fig. 14.26 - Temperature effect graphic [22]

14.2.2.7 Linearity deviation D_{LIN}

Absolute value of the maximum deflection of a transducer curve determined by an increasing load with respect to the reference straight line which approximates the characteristic curve of a straight line (ideal). The specified value is expressed in respect of the nominal value. To determine the deviation from linearity we make a measurement series over a load increased from zero to rated torque. [22]

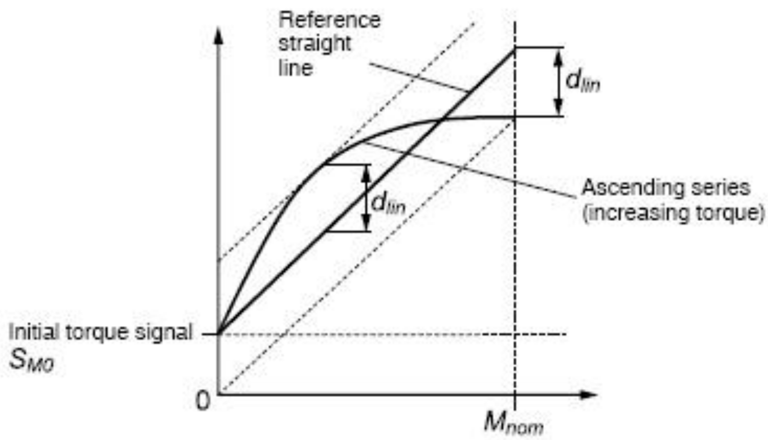


Fig. 14.27 - Linearity deviation. [22]

14.2.2.8 Linearity deviation including hysteresis D_{LH}

It indicates the maximum deviation of the output signal from the reference line. We take into consideration both the deviation from linearity and hysteresis. The specified value is expressed with respect to the nominal value. The only difference with the deviation from linearity is a load cycle that also includes the load falling from rated torque value to zero torque value. [22]

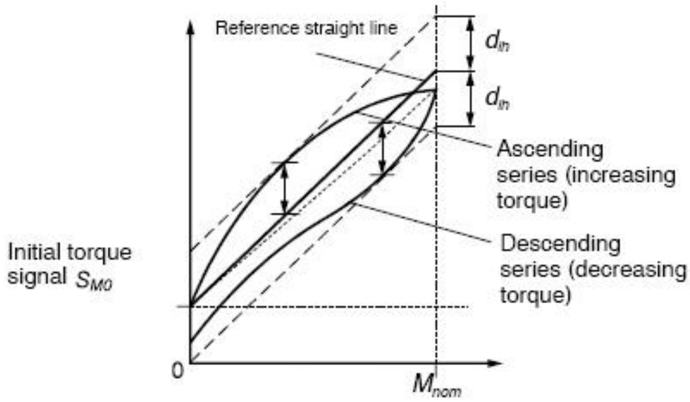


Fig. 14.28 - Linearity deviation including hysteresis. [22]

14.2.2.9 Hysteresis error

The hysteresis error is the difference between curves determined by an increasing and decreasing torque. [22]

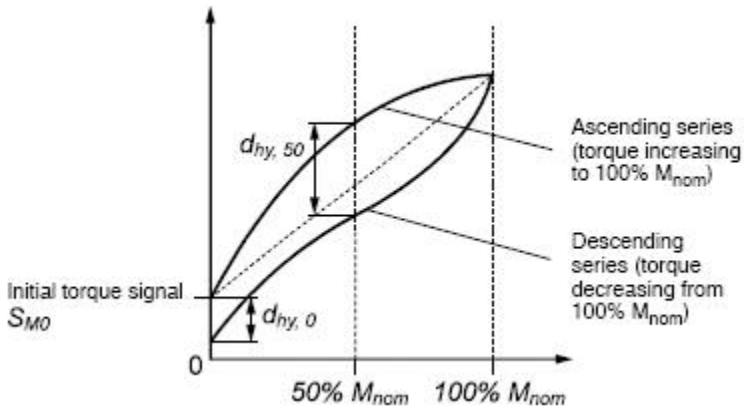


Fig. 14.29 - Hysteresis error. [22]

14.2.2.10 Relative standard deviation of reproducibility

Reproducibility is defined as the property of the output signal to have the same value on repeated measurements of the same torque. The *standard deviation* means the average deviation between several measurements of the same torque performed under the same conditions. The standard deviation is determined by static calibration equipment. [22]

14.3 Annex III - Torque transducer metrological properties calculation

In this chapter we are going to show the equations we are going to use to calculate the different metrological properties of the torque transducers for its calibration. All these properties are defined in Annex II: Torque transducer characteristics. The following formulas were collected from the "EA Guidelines on the Calibration of Static Torque Measuring Devices".

14.3.1 Indicated value

The indicated value, *X,* is defined as the difference between an indication, *I*, in loaded condition and an indication in unloaded condition. The indication at the beginning of each measurement series should be zeroed, or taken into account by computation during the evaluation following the measurement.

14.3.2 Sensibility

The sensitivity *S* shall be calculated according to the following equation:

$$S = \frac{\overline{X}_{\mathrm{E}}}{M_{\mathrm{E}}}$$

Eq. 14.10 - Sensibility [5]

X_E: mean value of the torque measuring device at maximum of the measuring range (mV/V^2).
M_E: maximum torque value of the measuring range (Nm). [5]

14.3.3 Mean value

The mean value $\overline{X}$ for each torque step shall be calculated according to equation as the mean value of the measurement results obtained in the increasing series in changed mounting positions:

$$\overline{X} = \frac{1}{n}\sum_{j=1}^{n}\left(I_j - I_{j,0}\right)$$

Eq. 14.11 – Mean value [5]

Where:

j *index of selected series*
n *number of increasing series in different mounting positions*

NOTE: The values measured in the 0° position in the second series at increasing torque are not included in the calculation of $\overline{X}$.

I: indication of torque measuring device at torque step with increasing torque.
Io: indication of torque measuring device of the zero signal prior to load application in mounting position. [3; 5]

14.3.4 Repeatability

The repeatability in unchanged mounting position (*b'*) shall be calculated for each torque step according to the following equation:

$$b' = \left| X_1 - X_2 \right|$$

Eq. 14.12 - Repeatability [5]

Where: X_1 and X_2 are the values measured for the different series in unchanged position. [5]

14.3.5 Reproducibility

The reproducibility in changed mounting position *b* shall be calculated for each torque step according to the following equation:

$$b = \sqrt{\frac{\sum_{j=1}^{n} \left(X_j - \overline{X} \right)^2}{n-1}}$$

Eq. 14.13 - Reproducibility [5]

Where: *n* number of increasing series in different mounting positions.

NOTE: For the 0° position, the second series at increasing torque is not included in the calculation of *b.* [5]

14.3.6 Hysteresis / Reversibility

The reversibility shall be determined according to the following equation as the mean of the absolute values of the differences between the values indicated for the series of increasing and decreasing torque series for each torque step:

$$h = \frac{1}{k} \sum_{j=1}^{k} \left| I_j - I'_j \right|$$

Eq. 14.14 - Hysteresis/ Reversibility [5]

Where: k number of torque series

NOTE: In this section a series is defined as increasing and decreasing torque. [5]

14.3.7 Deviation from the fitting curve / Non-linearity

The deviation from the fitting curve f_a shall be determined for each torque step for the indication as a function of the torque using an equation of the 1st, 2nd or 3rd degree without absolute term. The equation used shall be stated in the calibration certificate.

The equation shall be calculated as the least squares fit. The deviation from the fitting curve shall be calculated from the following equation.

$$f_a = \left(\overline{X} - X_a\right)$$

Eq. 14.15 - Non-linearity [5]

> NOTE: An alternate method consists to calculate the fitting curve and the associated standard uncertainty (u_{fa}) using the orthogonal polynomial method (Forsythe's algorithm). If this approach is adopted, it should be stated in the certificate. [5]

14.3.8 Deviation of indication

The deviation of indication f_q shall be determined only for such torque measuring devices where the measured value is directly indicated in the unit of torque and the indicated value is not fitted. It shall be determined from the mean value of the increasing series in changed mounting positions, equation:

$$f_q = \left(\overline{X} - M_{\mathrm{k}}\right)$$

Eq. 14.16 - Deviation of indication [5]

M_k: applied calibration torque (Nm). [5]

14.3.9 Creep and Creep recovery

The creep is the change in the output which occurs with time while a torque transducer is subjected to a constant torque and with all environmental conditions and other variables remaining constant. Creep recovery is the change in the output which occurs with time after the torque applied to the torque transducer has been removed, with all environmental conditions and other variables remaining constant.

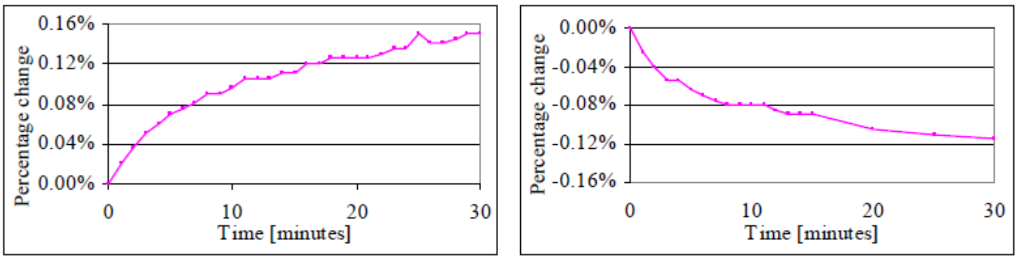


Fig. 14.30 - Graphs showing creep and creep recovery curves for a typical transducer [3]

To calculate the creep and the creep recovery we have to do the following test:

1. Preload the device.
2. With zero applied torque, wait for between 30 and 60 minutes. Record the zero torque output.
3. Apply the maximum calibration torque. Immediately record the initial torque output and record the reading thereafter at intervals not exceeding 5 minutes, over a period not less than 30 minutes.
4. Subtract the initial torque output from each subsequent output. Express these differences as a percentage of the maximum deflection. The creep shall be taken as the maximum difference obtained in the first 30 minutes.
5. Remove the applied torque, immediately record the initial zero torque output and record the reading thereafter at intervals not exceeding 5 minutes, over a period not less than 30 minutes.
6. Subtract the initial zero output from each subsequent output. Express these differences as a percentage of maximum deflection. The creep recovery shall be taken as the maximum difference obtained in the first 30 minutes. [3]

Creep:

$$\text{Max}\left[\frac{d_i - d_{init}}{d_{max}} x100\right]$$

Eq. 14.17 – Creep [3]

Creep recovery:

$$\text{Max}\left[\frac{I_{0\_i} - I_{0\_init}}{d_{max}} x100\right]$$

Eq. 14.18 – Creep recovery [3]

Where:
d_i is the i^{th} deflection at maximum applied torque;
d_{init} is the initial deflection at maximum applied torque;
d_{max} is the maximum deflection at maximum applied torque;
I_{0_i} is the i^{th} indicator output at zero applied torque;
I_{0_init} is the initial indicator output at zero applied torque.

14.3.10 Overload effect

The overload effect is the effect of 110% overload on zero output and maximum calibration torque. To calculate this effect we have to do the following test:

1. Apply the maximum calibration torque 3 times, recording the zero torque output and the maximum calibration torque output for each torque application. Calculate the mean zero torque output and the mean deflection.
2. Apply an overload of approximately 110% of the maximum calibration torque three times, recording the zero torque output and the overload torque output for each application.
3. Repeat step 1.
4. Calculate the difference between the mean zero torque output obtained in step 1 and the mean zero torque obtained in step 3. Express this as a percentage of the maximum deflection.
5. Calculate the difference between the mean deflection obtained in step 1 and the mean deflection obtained in step 3. Express this as a percentage of the maximum deflection. [3]

$$\mathrm{Max}\left[\frac{\bar{\mathrm{I}}'_0 - \bar{\mathrm{I}}''_0}{\bar{\mathrm{d}}'}\mathrm{x}100\right] - \text{zero output}$$

$$\mathrm{Max}\left[\frac{\bar{d}' - \bar{d}''}{\bar{d}'}x100\right] - \text{maximum calibration torque}$$

Eq. 14.19 – Overload effect [3]

Where:

$\bar{I}'_0$ *is the mean zero torque output before the overload;*
$\bar{I}''_0$ *is the mean zero torque output after the overload;*
$\bar{d}'$ *is the mean deflection before the overload;*
$\bar{d}''$ *is the mean deflection after the overload.*

14.3.11 Output stability at zero torque

The output stability at zero torque measures the drift in zero value. Through regular use, knowledge of a transducer's long-term stability can be built up. For a new transducer or one with little history the following tests can give some indication of the stability of the transducer.

1. Mount the transducer in the calibration device and allow at least 30 minutes settling.
2. With zero applied torque take a series of 15 readings at regular time intervals over a period of at least 48 hours simultaneously recording temperature.
3. Record the spread and express as a percentage of maximum deflection. [3]

$$\frac{\mathrm{I}_{0\,\mathrm{max}} - \mathrm{I}_{0\,min}}{d_{max}}x100$$

Eq. 14.20 - Output stability at zero torque [3]

Where:
$I_{0\,max}$ *is the maximum output with zero applied torque;*
$I_{0\,min}$ *is the minimum output with zero applied torque;*
d_{max} *is the deflection at the maximum calibration torque.*

14.3.12 Output stability at maximum torque

The output stability at maximum toque measures the drift at maximum calibration torque.

1. Preload the transducer.
2. Apply the maximum calibration torque.
3. While maintaining the maximum calibration torque take a series of 15 readings at regular time intervals over a period of at least 48 hours simultaneously recording temperature.
4. Record the spread and express as a percentage of maximum deflection. [3]

$$\text{Max}\left[\frac{d_{max} - d_{min}}{d_{max}} x100\right]$$

Eq. 14.21 – Output stability at maximum torque [3]

Where:
d_{max} *is the maximum deflection recorded over the measurement period;*
d_{min} *is the minimum deflection recorded over the measurement period.*

14.3.13 Other available test:

- *Alternating torque*: zero shifts when applying a clockwise torque followed by an anticlockwise torque. The toggle effect occurs when a full alternating load cycle takes place or when the directions of preloading and loading differ.

- *Torque versus angle.*

- *Excitation voltage effects:* change in output caused by a change in the excitation voltage for a mV/V measurement system.

- *Environmental tests:*
  - ✓ Temperature sensitivity at zero torque: changes in zero output with change in temperature.
  - ✓ Temperature sensitivity at maximum calibration torque: changes in the sensitivity of the transducer with change in temperature.

- *Dynamic tests:*
  - ✓ Frequency response (electronic): characterized by magnitude of the system's response.
  - ✓ Speed effect on zero: changes in the zero output of the transducer due to rotational effects. [3]

14.4 Annex IV - Local gravity

We need to know the local gravity value at the location where the Calibration bench will be placed and its accuracy in order to be able to measure the torque transmitted error.

14.4.1 Theoretical gravity or normal gravity

The gravitational acceleration at sea level on the Earth's surface would be constant if the Earth were stationary and a perfect sphere. However Earth's shape departs considerably from perfect spherity. The major departure is due to its oblateness which results in a variation of gravity with latitude. Because the Earth is rotating there is also a centrifugal acceleration opposing gravity. This acceleration is inversely proportional to the distance of the point of observation from the Earth's axis of rotation and is therefore also dependent.

The combined effect of these two effects is that the acceleration at the poles is about 5300 mgal higher than at the Equator. In general, for practical purposes, the value of gravity, g_o, in mgal at sea level at latitude Φ is given by the International Gravity Formula: [23]

$$g_o = 978031{,}85 \cdot (1 + 5{,}278895 \cdot 10^{-3} sin^2\theta + 2{,}3462 \cdot 10^{-5} sin^4\theta)$$

Eq. 14.22- International gravity formula [23]

Or also:

$$g_o = 978032{,}7 \cdot (1 + 0{,}0053024 \cdot sin^2\theta - 0{,}0000058 \cdot sin^2 2\theta)$$

Eq. 14.23 - International gravity formula [24]

14.4.2 Corrections to observed gravity values

Usually the observed gravity values vary from the values obtained from the International gravity formula. We have to analyse other factors in order to have an accurate value of the local gravity. The acceleration due to gravity varies over the Earth's surface for the following reasons:

Latitude:

This effect has been discussed above and can be avoided using the international gravity formula. [23]

Elevation – Free air anomalies

Between stations, the acceleration due to gravity varies inversely as the square of the distance from the centre of the Earth. The gravity formula gives the acceleration at sea level, g_o, which is:

$$g_0 = \frac{G \cdot M_e}{R^2}$$

Eq. 14.24 [23]

The gravity value, g_h, at a height h vertically above is:

$$g_h = \frac{G \cdot M_e}{(R+h)^2} = \frac{G \cdot M_e}{R^2}\left(1 - 2\frac{h}{R} + 3\frac{h^2}{R} + \cdots\right)$$

Eq. 14.25 [23]

The change in gravity due to the height h is:

$$(g_o - g_h) \cong \frac{2g_o h}{R} \quad \text{As h<<R}$$

Eq. 14.26 [23]

The change per unit height is $2g_o/R$ or 0,3086 mgal/m and is known as the Free Air Gradient. [23]

Bouguer anomaly

Bouguer anomalies take into account the gravitational effect of the rock present, the observation point and the anomaly due to the elevation. This anomaly is due to the density of the Earth's interior is not homogeneous as assumed. Approaching the rock layer of density ρ under the station to a horizontal infinite lamina tangent to the reference ellipsoid of thickness h whose gravitational attraction is $2\pi G r H = 0{,}04193 \cdot \rho \cdot h$. The Bouguer gravity anomaly (Δg_B) is defined by applying all the corrections described before [23]:

$$\Delta g_B = g_m - g_n + 0{,}3086 \cdot h - 0{,}04193 \cdot \rho \cdot h$$

Eq. 14.27 [23]

g_m: Observed gravity value.
g_n: Normal or theoretical gravity value.
h: elevation.
ϱ: rock density.

14.4.3 Local gravity formula

Taking into account all the effects explained above we obtain the following equation to calculate the local gravity:

$$g = g_n + \Delta g_B - 0{,}3086 \cdot h + 0{,}04193 \cdot \rho \cdot h$$

Eq. 14.28 – Local gravity formula [23]

g: local gravity.
g_n: Normal or theoretical gravity value.
h: height.
ϱ: rock density.

14.4.4 Local gravity calculation using the gravity formula

Firstly we are going to apply the local gravity formula to calculate a first approximation to our local gravity. The Calibration bench will be placed in the ETEC laboratory of the VUB in Etterbeek (Brussels), which has 50,83º latitude and 62m height.

We used a Bouguer anomaly contour map of Belgium taken from the "Observatoire Royal de Belgique". From that contour map we got a Bouguer anomaly value for Brussels of: Δg_B= -9 mgal. The rock density value used by the "Observatoire Royal de Belgique" was $\rho = 2{,}74\frac{g}{cm^3}$. Replacing these values in the gravity formula we obtain the local gravity value:

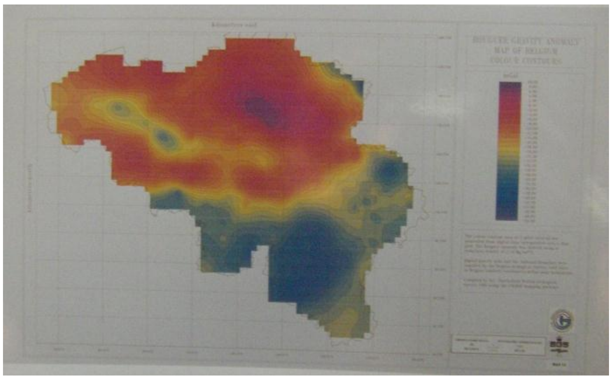


Fig. 14.31 - Belgium's Bouguer anomaly contour map

$$g = 978032{,}7 \cdot (1 + 0{,}0053024 \cdot sin^2 50{,}83º - 0{,}0000058 \cdot sin^2 2 \cdot 50{,}83º) - 9 - 0{,}3086 \cdot 62 + 0{,}04193 \cdot 2{,}74 \cdot 62 = 981.123\, mgal = 9{,}81123\ \frac{m}{s^2}$$

Eq. 14.29

14.4.5 Local gravity value from the Gravity Information System (SIS)

The Physikalisch-Technische Bundesanstalt (PTB) is the national metrology institute of Germany, who provides scientific and technical services with the highest accuracy and reliability. Within the project "Gravimetry" numerical procedures were developed based upon geodetic gravity field models and free available gravity data in view of the accessibility of local gravity field information for an extended user circle from physics and industry. [24]

Using the SIS internet application we could have an accurate value for the local value and its uncertainty. This will be the value we will use for the calibration and for the calculations of the torque transmitted uncertainty.

$$g = 9{,}81130 \ {}^{m}/_{s^2}$$

$$Ug_{95\%} = \pm 0{,}000002 \ {}^{m}/_{s^2}$$

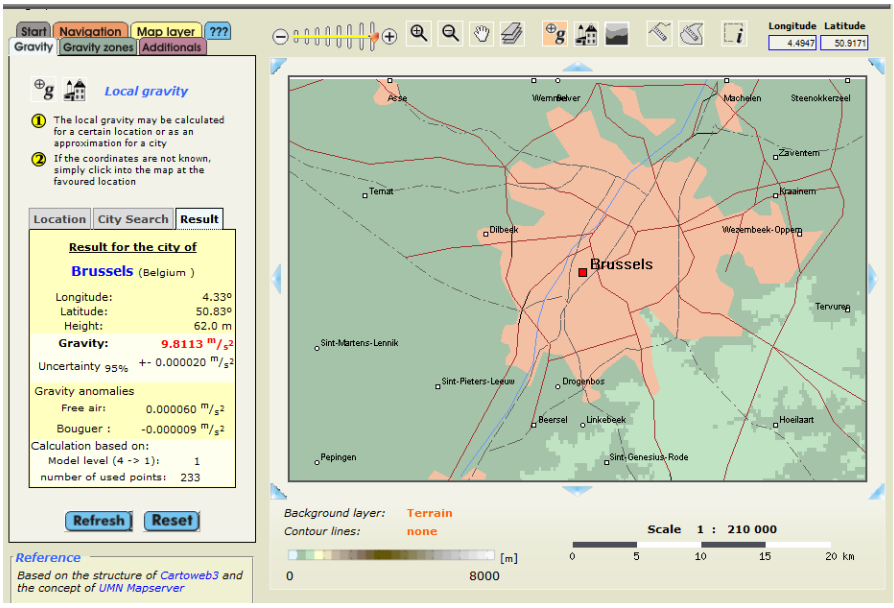


Fig. 14.32 - Brussels gravity value on SIS application. [24]

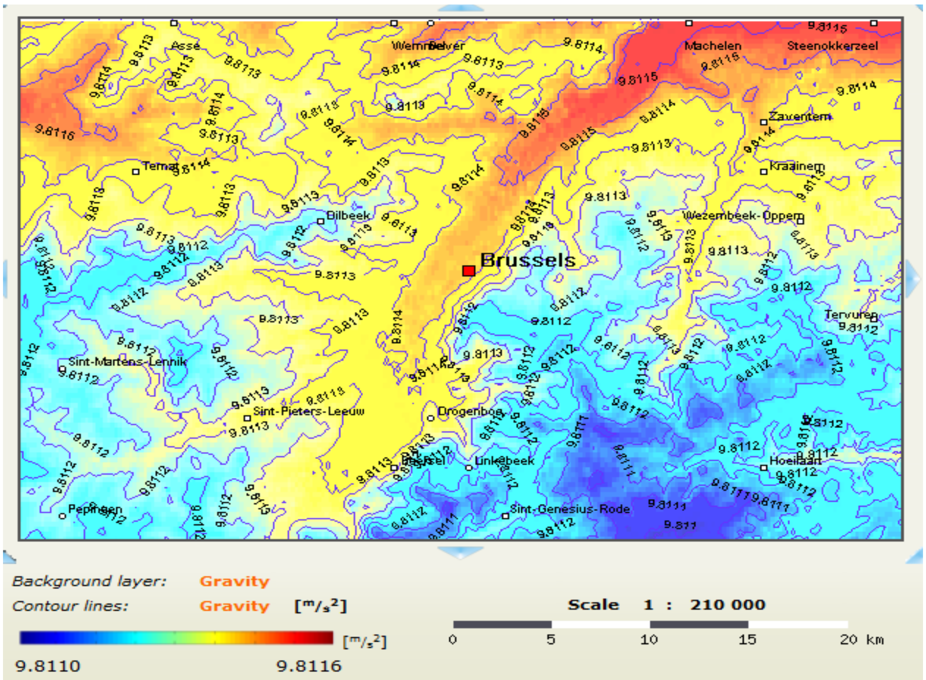


Fig. 14.33 - Brussels gravity contour map. [24]

14.5 Annex V - Uncertainty theory

14.5.1 Standard uncertainty

The standard uncertainty *u(y)* in a measurement indicates the reliability or precision of the measurement. It has the same units as the nominal value. The standard uncertainty *u(y)* of a measurement result *y* is the estimated standard deviation of *y*.

If the probability distribution characterized by the measurement result y and its standard uncertainty $u(y)$ are approximately normal (Gaussian), and $u(y)$ is a reliable estimate of the standard deviation of y, then the interval $y - u(y)$ to $y + u(y)$ is expected to encompass approximately 68 % of the distribution of values that could reasonably be attributed to the value of the quantity Y of which y is an estimate. This implies that it is believed with an approximate level of confidence of 68 % that Y is greater than or equal to y - $u(y)$, and is less than or equal to $y + u(y)$, which is commonly written as $Y = y \pm u(y)$. [25]

14.5.2 Relative standard uncertainty

The quality of a measurement cannot be solely determined by its standard uncertainty *u(y)*. For example, an absolute uncertainty *u(y)=1mm* has different meaning when referring to a length $\bar{y}$*=1cm* or to a length $\bar{y}$*=10m*. The quality of a measurement is better expressed by the relative standard uncertainty. The relative standard uncertainty *w(y)* of a measurement result *y* is defined by $w(y) = \frac{u\,(y)}{|y|}$, where *y* is not equal to 0.

The smaller the relative standard uncertainty is, the higher the quality of the measurement. By definition, the relative standard uncertainty is always a dimensionless quantity. It can be much smaller than one. To avoid the use of many decimal digits, it's sometimes multiplied by 100 and indicated by %. [25]

14.5.3 Combined standard uncertainty

The combined standard uncertainty of the measurement result y, designated by $u_c(y)$ and taken to represent the estimated standard deviation of the result, is the positive square root of the estimated variance $u_c^2(y)$ obtained from.

$$u_c^2(y) = \sum_{i=1}^{N} \left(\frac{\partial f}{\partial x_i}\right)^2 u^2(x_i) + 2\sum_{i=1}^{N-1} \sum_{j=i+1}^{N} \frac{\partial f}{\partial x_i} \frac{\partial f}{\partial x_j} u(x_i, x_j)$$

Eq. 14.30 [25]

This equation is based on a first-order Taylor series approximation of the measurement equation $Y = f(X_1, X_2, \ldots, X_N)$ and is conveniently referred to as the *Law of propagation of uncertainty*. The partial derivatives of f with respect to the X_i (often referred to as *sensitivity coefficients*) are equal to the partial derivatives of f with respect to the X_i evaluated at $X_i = x_i$, $u(x_i)$ is the standard uncertainty associated with the input estimate x_i ; and $u(x_i, x_j)$ is the estimated covariance associated with x_i and x_j.

This equation often reduces to a simple form in cases of practical interest. For example, if the input estimates x_i of the input quantities X_i can be assumed to be uncorrelated, then the second term vanishes. Further, if the input estimates are uncorrelated and the measurement equation is one of the following two forms, then the equation becomes simpler still.

Measurement equation:

A **sum** of quantities X_i multiplied by constants a_i.
$Y = a_1X_1 + a_2X_2 + \ldots a_NX_N$

Measurement result:

$y = a_1x_1 + a_2x_2 + \ldots a_Nx_N$

Combined standard uncertainty:

$$u_c^2(y) = a_1^2u^2(x_1) + a_2^2u^2(x_2) + \cdots a_N^2u^2(x_N)$$

Eq. 14.31 [25]

Measurement equation:

A **product** of quantities X_i, raised to powers *a, b, ... p*, multiplied by a constant *A*.
$Y = AX_1^a X_2^b \ldots X_N^p$

Measurement result:

$y = Ax_1^a x_2^b \ldots x_N^p$

Combined standard uncertainty:

$$w^2(y) = a^2w^2(x_1) + b^2w^2(x_2) + \cdots p^2w^2(x_N)$$ Eq. 14.32 [25]

Where $w(x_i)$ is the relative standard uncertainty of x_i and is defined by $w(x_i) = u(x_i)/|x_i|$, where $|x_i|$ is the absolute value of x_i and x_i is not equal to zero; and $w_c(y)$ is the ***relative combined standard uncertainty*** of *y* and is defined by $w_{c,}(y) = u_c(y)/|y|$, where $|y|$ is the absolute value of y and y is not equal to zero.

If the probability distribution characterized by the measurement result y and its combined standard uncertainty $u_c(y)$ is approximately normal (Gaussian), and $u_c(y)$ is a reliable estimate of the standard deviation of y, then the interval $y - u_c(y)$ to $y + u_c(y)$ is expected to encompass approximately 68 % of the distribution of values that could reasonably be attributed to the value of the quantity Y of which y is an estimate. This implies that it is believed with an approximate level of confidence of 68 % that Y is greater than or equal to $y - u_c(y)$, and is less than or equal to $y + u_c(y)$, which is commonly written as $Y = y \pm u_c(y)$. [25]

14.5.4 Expanded uncertainty

Although the combined standard uncertainty u_c is used to express the uncertainty of many measurement results, for some commercial, industrial, and regulatory applications (e.g., when health and safety are concerned), what is often required is a measure of uncertainty that defines an interval about the measurement result y within which the value of the measured Y can be confidently asserted to truth.

The measure of uncertainty intended to meet this requirement is termed expanded uncertainty, suggested symbol U, and is obtained by multiplying $u_c(y)$ by a coverage factor, suggested symbol k. Thus $U = k \cdot u_c(y)$ and it is confidently believed that Y is greater than or equal to $y - U$, and is less than or equal to $y + U$, which is commonly written as $Y = y \pm U$.

In general, the value of the coverage factor k is chosen on the basis of the desired level of confidence to be associated with the interval defined by $U = k \cdot u_c(y)$. Typically, k is in the range 2 to 3. When the normal distribution applies and u_c is a reliable estimate of the standard deviation of y, $U = 2 \cdot u_c$ (i.e., $k = 2$) defines an interval having a level of confidence of approximately 95 %, and $U = 3 \cdot u_c$ (i.e., $k = 3$) defines an interval having a level of confidence greater than 99 %.

In analogy with relative standard uncertainty u_r and relative combined standard uncertainty $w_{,r}$ defined above in connection with simplified forms of equation (6), the ***relative expanded uncertainty*** of a measurement result y is $W = U/|y|$, y not equal to zero. [25]

14.6 Annex VI – Fits standards

〔Technical Data〕

Description of Fits／Graphical Representation of Standard Fits

"Drawing Manual" in JIS "How to Use" Series   Excerpt from JIS B0401 (1999)

			H6	H7	H8	H9	Applicable Part	Functional Classification		Application Example
Can Be Moved Relatively	Clearance Fit	Loose Fit				c9	Part which accommodates a wide gap or moving part which needs a gap. Part which accommodates a wide gap to facilitate assembling. Part which needs an appropriate gap even at a high temperature.	Part whose structure needs a gap. { Inflates. Large position error Long		Piston Ring and Ring Groove Fitting by means of a loose set pin.
		Light Roll Fit			d9	d9	Part which accommodates or needs a gap.	Cost needs to be reduced. { Manufacturing Cost Maintenance Cost		Crank Web and Pin Bearing (Side) Exhaust Valve Box and the Sliding Part of a Spring Bearing Piston Ring and the Ring Groove
				e7	e8	e9	Part which accommodates a wide gap or needs a gap. Fairly wide gap, well greased bearing. Bearing subjected to a high temperature, high speed and heavy load (high-degree forced lubrication).	Regular Rotary or Sliding Part (Must be well greased.)		Fitting of the Exhaust Valve Box Main Bearing for the Crankshaft Regular Sliding Part
		Roll Fit	f6	f7	f7 f8		Fitting so as to provide an appropriate gap to permit movement (high-quality fitting). Regular normal-temperature bearing lubricated with grease or oil.	Regular Fitting (Often comes apart.)		Part in which a cooled exhaust valve box is inserted. Regular Shaft and Bush Link Device Lever and Bush
		Fine Roll Fit	g5	g6			Continuously rotating part of a precision machine under a light load. Fitting with a narrow gap so as to permit movement (spigot and positioning). Precision sliding part.	Part required to make a precision motion with virtually no play.		Link Device Pin and Lever Key and Its Groove Precision Control Valve Rod
Cannot Be Moved Relatively	Transition Fit	Sliding Fit	h5	h6	h7 h8	h9	Fitting so as to permit movement by hand, with a lubricant applied. (high-quality positioning) Special High-Precision Sliding Part Unimportant Stationary Part	Can be disassembled/re-assembled without damaging component parts.	Force cannot be transmitted by the fitting force alone.	Fitting a rim and a boss together. Fitting the gear of a precision gear device.
		Push Fit	h5 h6	js6			Fitting which accommodates a light gap. Precision fitting which locks both parts while the unit is used. Fitting which allows assembling and disassembling with a wooden or lead hammer.			Fitting Coupling Flanges Together Governor Path and Pin Fitting a Gear Rim and a Boss Together
		Driving Fit	js5	k6			Fitting which requires an iron hammer or hand press for assembling／disassembling (a key or the like is necessary to prevent inter-part shaft rotation). Precision positioning.			Reamer Bolt to Fix the Shaft of a Gear Pump and a Casing Together
			k5	m6			Requires an iron hammer or hand press for assembling／disassembling. Precision positioning which allows no gap.			Reamer Bolt Fixing the piston of hydraulic equipment and a shaft together Fitting a Coupling Flange and a Shaft Together
		Light Press Fit	m5	n6			Fitting which requires considerable force for assembling／disassembling. Precision stationary fitting (a key or the like is necessary for high-torque transmission purposes)		Slight force can be transmitted by the fitting force alone.	Flexible-Axis Coupling and Gear (Passive Side) Precision Fitting Insertion of a Suction Valve and Valve Guide
	Interference Fit	Press Fit	n5 n6	p6			Fitting which requires much force for assembling/disassembling (a key or the like is necessary for high- torque transmission). Light press fitting or the like is necessary for non-ferrous component parts. Standard press fitting is required for iron component parts and a bronze part and a copper part.	Hard to disassemble without damaging component parts.		Insertion of a Suction Valve and Valve Guide Fixing a Gear and a Shaft Together (Low Torque) Shaft of a Flexible Coupling and a Gear (Drive Side)
		Strong Press Fit, Shrinkage Fit, Freeze Fit	p5	r6			Fitting which requires much force for assembling／disassembling. Shrinkage press fitting, cold press fitting or forced press fitting is required for large component parts.		Considerable force can be transmitted by the fitting force alone.	Coupling and Shaft
			r5	s6 t6 u6 x6			Firmly coupled together and requires shrinkage press fitting, cold press fitting or forced press fitting. Permanent assembly, which can not come apart. Press fitting or the like is required for light alloy members.			Attaching and Fixing a Bearing Bush Insertion of a Suction Valve and Valve Guide Fixing a Coupling Flange and a Shaft Together (High Torque) Fixing the Rim of a Drive Gear and a Boss Together Attaching and Fixing a Bearing Bush

Dimensional Tolerances of Shaft, Regularly Used Fitting

Basic Size (mm) Over	Incl.	b9	c9	d8	d9	e7	e8	e9	f6	f7	f8	g5	g6	h5	h6	h7	h8	h9	js5	js6	js7	k5	k6	m5	m6	n5 *	n6	p6	r6	s6	t6	u6	x6
	Class of Tolerance Range for Shafts — Unit μm																																
–	3	-140 -165	-60 -85	-20 -34	-20 -45	-14 -24	-14 -28	-14 -39	-6 -12	-6 -16	-6 -20	-2 -6	-2 -8	0 -4	0 -6	0 -10	0 -14	0 -25	±2	±3	±5	+4 0	+6 0	+6 +2	+8 +2	+8 +4	+10 +4	+12 +6	+16 +10	+20 +14	–	+24 +18	+26 +20
3	6	-140 -170	-70 -100	-30 -48	-30 -60	-20 -32	-20 -38	-20 -50	-10 -18	-10 -22	-10 -28	-4 -9	-4 -12	0 -5	0 -8	0 -12	0 -18	0 -30	±2.5	±4	±6	+6 +1	+9 +1	+9 +4	+12 +4	+13 +8	+16 +8	+20 +12	+23 +15	+27 +19	–	+31 +23	+36 +28
6	10	-150 -186	-80 -116	-40 -62	-40 -76	-25 -40	-25 -47	-25 -61	-13 -22	-13 -28	-13 -35	-5 -11	-5 -14	0 -6	0 -9	0 -15	0 -22	0 -36	±3	±4.5	±7.5	+7 +1	+10 +1	+12 +6	+15 +6	+16 +10	+19 +10	+24 +15	+28 +19	+32 +23	–	+37 +28	+43 +34
10	14	-150 -193	-95 -138	-50 -77	-50 -93	-32 -50	-32 -59	-32 -75	-16 -27	-16 -34	-16 -43	-6 -14	-6 -17	0 -8	0 -11	0 -18	0 -27	0 -43	±4	±5.5	±9	+9 +1	+12 +1	+15 +7	+18 +7	+20 +12	+23 +12	+29 +18	+34 +23	+39 +28	–	+44 +33	+51 +40
14	18																																+56 +45
18	24	-160 -212	-110 -162	-65 -98	-65 -117	-40 -61	-40 -73	-40 -92	-20 -33	-20 -41	-20 -53	-7 -16	-7 -20	0 -9	0 -13	0 -21	0 -33	0 -52	±4.5	±6.5	±10.5	+11 +2	+15 +2	+17 +8	+21 +8	+24 +15	+28 +15	+35 +22	+41 +28	+48 +35	–	+54 +41	+67 +54
24	30																														+54 +41	+61 +48	+77 +64
30	40	-170 -232	-120 -182	-80 -119	-80 -142	-50 -75	-50 -89	-50 -112	-25 -41	-25 -50	-25 -64	-9 -20	-9 -25	0 -11	0 -16	0 -25	0 -39	0 -62	±5.5	±8	±12.5	+13 +2	+18 +2	+20 +9	+25 +9	+28 +17	+33 +17	+42 +26	+50 +34	+59 +43	+64 +48	+76 +60	–
40	50	-180 -242	-130 -192																												+70 +54	+86 +70	
50	65	-190 -264	-140 -214	-100 -146	-100 -174	-60 -90	-60 -106	-60 -134	-30 -49	-30 -60	-30 -76	-10 -23	-10 -29	0 -13	0 -19	0 -30	0 -46	0 -74	±6.5	±9.5	±15	+15 +2	+21 +2	+24 +11	+30 +11	+33 +20	+39 +20	+51 +32	+60 +41	+72 +53	+85 +66	+106 +87	–
65	80	-200 -274	-150 -224																										+62 +43	+78 +59	+94 +75	+121 +102	
80	100	-220 -307	-170 -257	-120 -174	-120 -207	-72 -107	-72 -126	-72 -159	-36 -58	-36 -71	-36 -90	-12 -27	-12 -34	0 -15	0 -22	0 -35	0 -54	0 -87	±7.5	±11	±17.5	+18 +3	+25 +3	+28 +13	+35 +13	+38 +23	+45 +23	+59 +37	+73 +51	+93 +71	+113 +91	+146 +124	–
100	120	-240 -327	-180 -267																										+76 +54	+101 +79	+126 +104	+166 +144	
120	140	-260 -360	-200 -300	-145 -208	-145 -245	-85 -125	-85 -148	-85 -185	-43 -68	-43 -83	-43 -106	-14 -32	-14 -39	0 -18	0 -25	0 -40	0 -63	0 -100	±9	±12.5	±20	+21 +3	+28 +3	+33 +15	+40 +15	–	+52 +27	+68 +43	+88 +63	+117 +92	+147 +122	–	–
140	160	-280 -380	-210 -310																										+90 +65	+125 +100	+159 +134		
160	180	-310 -410	-230 -330																										+93 +68	+133 +108	+171 +146		
180	200	-340 -455	-240 -355	-170 -242	-170 -285	-100 -146	-100 -172	-100 -215	-50 -79	-50 -96	-50 -122	-15 -35	-15 -44	0 -20	0 -29	0 -46	0 -72	0 -115	±10	±14.5	±23	+24 +4	+33 +4	+37 +17	+46 +17	–	+60 +31	+79 +50	+106 +77	+151 +122	–	–	–
200	225	-380 -495	-260 -375																										+109 +80	+159 +130			
225	250	-420 -535	-280 -395																										+113 +84	+169 +140			
250	280	-480 -610	-300 -430	-190 -271	-190 -320	-110 -162	-110 -191	-110 -240	-56 -88	-56 -108	-56 -137	-17 -40	-17 -49	0 -23	0 -32	0 -52	0 -81	0 -130	±11.5	±16	±26	+27 +4	+36 +4	+43 +20	+52 +20	–	+66 +34	+88 +56	+126 +94	–	–	–	–
280	315	-540 -670	-330 -460																										+130 +98				
315	355	-600 -740	-360 -500	-210 -299	-210 -350	-125 -182	-125 -214	-125 -265	-62 -98	-62 -119	-62 -151	-18 -43	-18 -54	0 -25	0 -36	0 -57	0 -89	0 -140	±12.5	±18	±28.5	+29 +4	+40 +4	+46 +21	+57 +21	–	+73 +37	+98 +62	+144 +108	–	–	–	–
355	400	-680 -820	-400 -540																										+150 +114				
400	450	-760 -915	-440 -595	-230 -327	-230 -385	-135 -198	-135 -232	-135 -290	-68 -108	-68 -131	-68 -165	-20 -47	-20 -60	0 -27	0 -40	0 -63	0 -97	0 -155	±13.5	±20	±31.5	+32 +5	+45 +5	+50 +23	+63 +23	–	+80 +40	+108 +68	+166 +126	–	–	–	–
450	500	-840 -995	-480 -635																										+172 +132				

Dimensional Tolerances of Shaft, Regularly Used Fitting

Basic Size (mm) Over	Incl.	B10	C9	C10	D8	D9	D10	E7	E8	E9	F6	F7	F8	G6	G7	H6	H7	H8	H9	H10	JS6	JS7	K6	K7	M6	M7	N6	N7	P6	P7	R7	S7	T7	U7	X7
	Class of Tolerance Range for Holes — Unit μm																																		
–	3	+180 +140	+85 +60	+100 +60	+34 +20	+45 +20	+60 +20	+24 +14	+28 +14	+39 +14	+12 +6	+16 +6	+20 +6	+8 +2	+12 +2	+6 0	+10 0	+14 0	+25 0	+40 0	±3	±5	0 -6	0 -10	-2 -8	-2 -12	-4 -10	-4 -14	-6 -12	-6 -16	-10 -20	-14 -24	–	-18 -28	-20 -30
3	6	+188 +140	+100 +70	+118 +70	+48 +30	+60 +30	+78 +30	+32 +20	+38 +20	+50 +20	+18 +10	+22 +10	+28 +10	+12 +4	+16 +4	+8 0	+12 0	+18 0	+30 0	+48 0	±4	±6	+2 -6	+3 -9	-1 -9	0 -12	-5 -13	-4 -16	-9 -17	-8 -20	-11 -23	-15 -27	–	-19 -31	-24 -36
6	10	+208 +150	+116 +80	+138 +80	+62 +40	+76 +40	+98 +40	+40 +25	+47 +25	+61 +25	+22 +13	+28 +13	+35 +13	+14 +5	+20 +5	+9 0	+15 0	+22 0	+36 0	+58 0	±4.5	±7.5	+2 -7	+5 -10	-3 -12	0 -15	-7 -16	-4 -19	-12 -21	-9 -24	-13 -28	-17 -32	–	-22 -37	-28 -43
10	14	+220 +150	+138 +95	+165 +95	+77 +50	+93 +50	+120 +50	+50 +32	+59 +32	+75 +32	+27 +16	+34 +16	+43 +16	+17 +6	+24 +6	+11 0	+18 0	+27 0	+43 0	+70 0	±5.5	±9	+2 -9	+6 -12	-4 -15	0 -18	-9 -20	-5 -23	-15 -26	-11 -29	-16 -34	-21 -39	–	-26 -44	-33 -51
14	18																																		-38 -56
18	24	+244 +160	+162 +110	+194 +110	+98 +65	+117 +65	+149 +65	+61 +40	+73 +40	+92 +40	+33 +20	+41 +20	+53 +20	+20 +7	+28 +7	+13 0	+21 0	+33 0	+52 0	+84 0	±6.5	±10.5	+2 -11	+6 -15	-4 -17	0 -21	-11 -24	-7 -28	-18 -31	-14 -35	-20 -41	-27 -48	–	-33 -54	-46 -67
24	30																																-33 -54	-40 -61	-56 -77
30	40	+270 +170	+182 +120	+220 +120	+119 +80	+142 +80	+180 +80	+75 +50	+89 +50	+112 +50	+41 +25	+50 +25	+64 +25	+25 +9	+34 +9	+16 0	+25 0	+39 0	+62 0	+100 0	±8	±12.5	+3 -13	+7 -18	-4 -20	0 -25	-12 -28	-8 -33	-21 -37	-17 -42	-25 -50	-34 -59	-39 -64	-51 -76	–
40	50	+280 +180	+192 +130	+230 +130																													-45 -70	-61 -86	
50	65	+310 +190	+214 +140	+260 +140	+146 +100	+174 +100	+220 +100	+90 +60	+106 +60	+134 +60	+49 +30	+60 +30	+76 +30	+29 +10	+40 +10	+19 0	+30 0	+46 0	+74 0	+120 0	±9.5	±15	+4 -15	+9 -21	-5 -24	0 -30	-14 -33	-9 -39	-26 -45	-21 -51	-30 -60	-42 -72	-55 -85	-76 -106	–
65	80	+320 +200	+224 +150	+270 +150																											-32 -62	-48 -78	-64 -94	-91 -121	
80	100	+360 +220	+257 +170	+310 +170	+174 +120	+207 +120	+260 +120	+107 +72	+126 +72	+159 +72	+58 +36	+71 +36	+90 +36	+34 +12	+47 +12	+22 0	+35 0	+54 0	+87 0	+140 0	±11	±17.5	+4 -18	+10 -25	-6 -28	0 -35	-16 -38	-10 -45	-30 -52	-24 -59	-38 -73	-58 -93	-78 -113	-111 -146	–
100	120	+380 +240	+267 +180	+320 +180																											-41 -76	-66 -101	-91 -126	-131 -166	
120	140	+420 +260	+300 +200	+360 +200	+208 +145	+245 +145	+305 +145	+125 +85	+148 +85	+185 +85	+68 +43	+83 +43	+106 +43	+39 +14	+54 +14	+25 0	+40 0	+63 0	+100 0	+160 0	±12.5	±20	+4 -21	+12 -28	-8 -33	0 -40	-20 -45	-12 -52	-36 -61	-28 -68	-48 -88	-77 -117	-107 -147	–	–
140	160	+440 +280	+310 +210	+370 +210																											-50 -90	-85 -125	-119 -159		
160	180	+470 +310	+330 +230	+390 +230																											-53 -93	-93 -133	-131 -171		
180	200	+525 +340	+355 +240	+425 +240	+242 +170	+285 +170	+355 +170	+146 +100	+172 +100	+215 +100	+79 +50	+96 +50	+122 +50	+44 +15	+61 +15	+29 0	+46 0	+72 0	+115 0	+185 0	±14.5	±23	+5 -24	+13 -33	-8 -37	0 -46	-22 -51	-14 -60	-41 -70	-33 -79	-60 -106	-105 -151	–	–	–
200	225	+565 +380	+375 +260	+445 +260																											-63 -109	-113 -159			
225	250	+605 +420	+395 +280	+465 +280																											-67 -113	-123 -169			
250	280	+690 +480	+430 +300	+510 +300	+271 +190	+320 +190	+400 +190	+162 +110	+191 +110	+240 +110	+88 +56	+108 +56	+137 +56	+49 +17	+69 +17	+32 0	+52 0	+81 0	+130 0	+210 0	±16	±26	+5 -27	+16 -36	-9 -41	0 -52	-25 -57	-14 -66	-47 -79	-36 -88	-74 -126	–	–	–	–
280	315	+750 +540	+460 +330	+540 +330																											-78 -130				
315	355	+830 +600	+600 +360	+690 +360	+299 +210	+350 +210	+440 +210	+182 +125	+214 +125	+265 +125	+98 +62	+119 +62	+151 +62	+54 +18	+75 +18	+36 0	+57 0	+89 0	+140 0	+230 0	±18	±28.5	+7 -29	+17 -40	-10 -46	0 -57	-26 -62	-16 -73	-51 -87	-41 -98	-87 -144	–	–	–	–
355	400	+910 +680	+540 +400	+630 +400																											-93 -150				
400	450	+1010 +760	+595 +440	+690 +440	+327 +230	+385 +230	+480 +230	+198 +135	+232 +135	+290 +135	+108 +68	+131 +68	+165 +68	+60 +20	+83 +20	+40 0	+63 0	+97 0	+155 0	+250 0	±20	±31.5	+8 -32	+18 -45	-10 -50	0 -63	-27 -67	-17 -80	-55 -95	-45 -108	-103 -166	–	–	–	–
450	500	+1090 +840	+635 +480	+730 +480																											-109 -172				

Fig. 14.34 - Fits standards [26]

14.7 Annex VII – Drawings

- Calibration Test Bench assembly
- Lever arm
- Adapter lever arm-shaft
- Adapter to rotatory table
- Hanger holder
- Shaft
- Hanger assembly
- Hanger head
- Hanger clevis pin
- Hanger base

15. References

[1.] Vibro-Meter. *Measuring Monitoring Testing; Torque transducers TORQUEMASTER.*

[2.] Wegener, Georg and Bruns, Thomas. Traceability of torque transducers under rotating and dynamic operating conditions. [Online]

[3.] Robinson, Andy. *Guide to the calibration and testing of torque transducers.* Industry and Innovation Division, National Physical Laboratory. London, 2008.

[4.] Norbar. Norbar Torque Tools. [Online] www.norbar.com.

[5.] EURAMET (European Association of National Metrology Institutes). Guidelines on the Calibration of Static Torque Measuring Devices. [Online] July 2007. http://www.euramet.org.

[6.] Sushma Industries. Sushma Industries. [Online] http://www.indiamart.com/sushma-inds/.

[7.] Mansfield, Michael y O'Sullivan, Colm. *Understanding Physics.* Chippenham (United Kingdom) : Wiley, 2011.

[8.] KTR. *Rigiflex Operating - Assembly Instructions.* 2008.

[9.] SKF. *SKF ConCentra roller bearing.* Hyltebruks Tryckeri AB, 2003.

[10.] Hewlett-Packard Company. *HP 34401A User's guide.* 1996.

[11.] Babu, K Ganesh y Srithar, K. *Design of machine elements.* New Delhi : Tata McGraw Hill, 2010.

[12.] Bhandari, V. B. *Design of machine elements.* New Delhi : Tata McGraw-Hill, 2008.

[13.] Autodesk. *Autodesk Inventor Getting Started.* 2009.

[14.] SKF. Friction - SKF.com. [Online] http://www.skf.com.

[15.] Winczner, Ken. E.E.Times. [Online] 27 3 2006. [Cited: 22 11 2010.] www.eetimes.com.

[16.] Webster, John G. *The Measurement, Instrumentation, and Sensors Handbook.* CRC Press; Springer; IEE Press, 1999.

[17.] Krishnaswamy, K. y Vijayachitra, S. *Industrial Instrumentation.* New Delhi : New Age International Publishers, 2005.

[18.] Lorenz Messtechnik GmbH. [Online] http://www.lorenz-messtechnik.de.

[19.] Direct industry. [Online] www.directindustry.com.

[20.] Magtrol. [Online] www.magtrol.com.

[21.] ATI (Advanced Telemetrics International). Telemetry systems, wireless telemetry, remote telemetry. [Online] http://www.atitelemetry.com.

[22.] Schicker, Rainer and Wegener, Georg. *Measuring Torque Correctly.* Hottinger Baldwin Messtechnik GmbH.

[23.] Khan, M. A. *Global geology.* London : Wykeham Publications, 1976.

[24.] PTB (Physikalisch-Technische Bundesanstalt). Gravity Information System (SIS). [Online] http://www.ptb.de/en/org/1/11/115/_index.htm.

[25.] Physical Measurement Laboratory of NIST. The NIST Reference on Constants, Units, and Uncertainty. [Online] http://physics.nist.gov/cuu/Uncertainty/index.html.

[26.] Misumi. Misumi Europe - Mechanical components. [Online] http://www.misumi-europe.com.

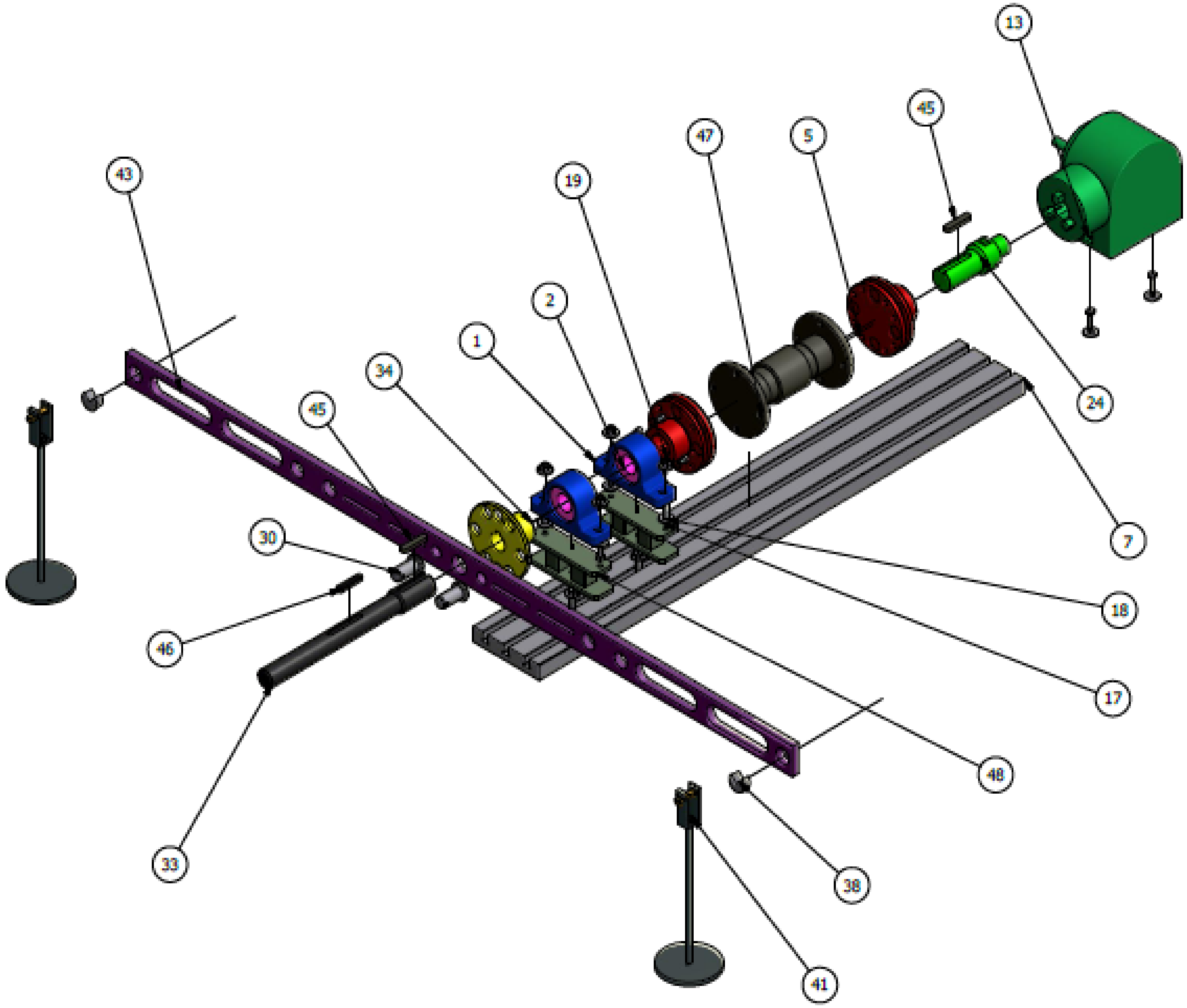
[27.] Kaman. Kaman Industrial Technologies. [Online] http://www.kamandirect.com/.

[28.] International Organization of Legal Metrology. OIML 111-1 International Recomendation. [Online] 2004. http://www.oiml.org/.

[29.] RoyMech. [Online] http://www.roymech.co.uk.

[30.] RS Components. RS Components - Electronic and Electrical Components. [Online] http://uk.rs-online.com.

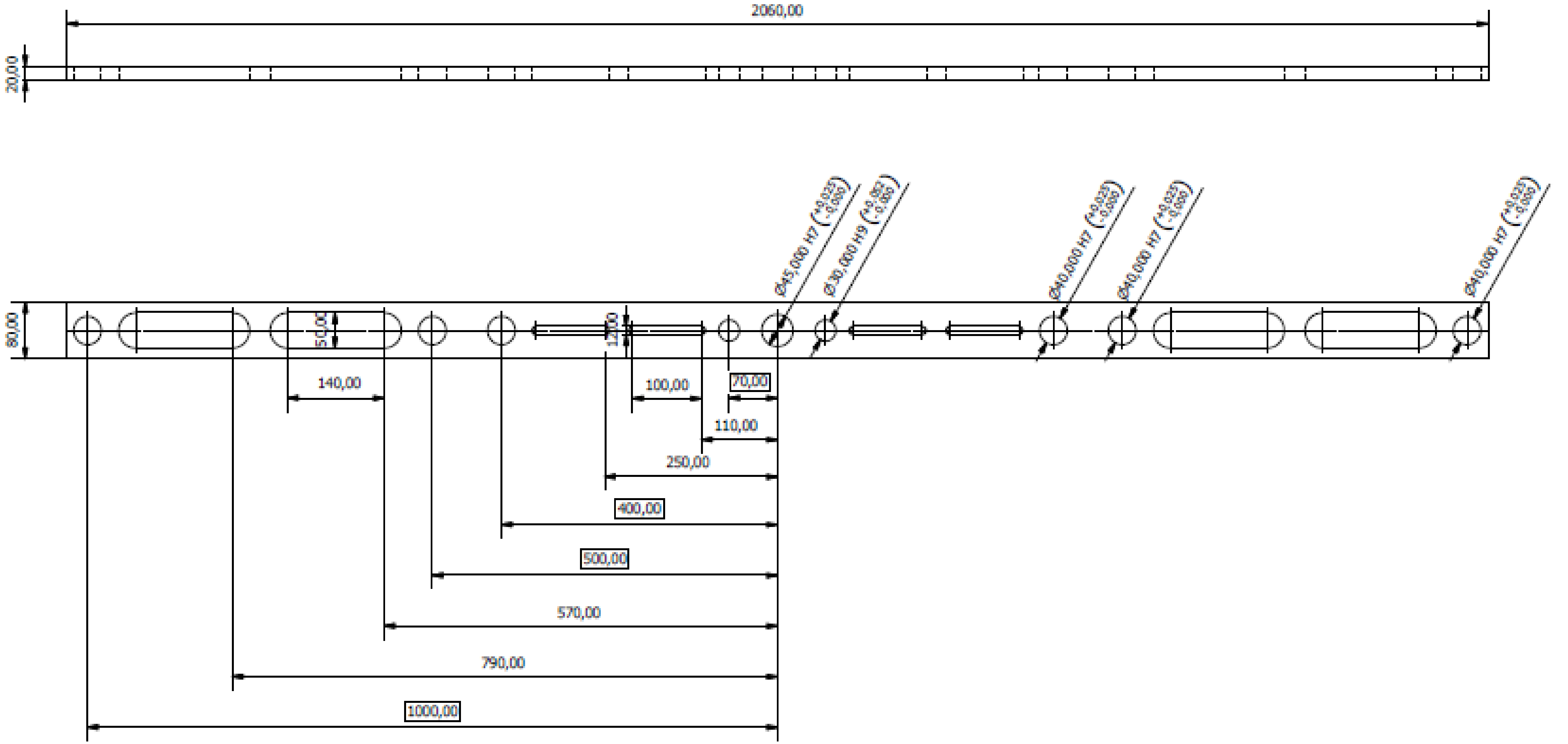
[31.] Carbonel, Jose A. Robles. Torque standard machines at CEM. [Online] 2006. http://www.imeko.org.



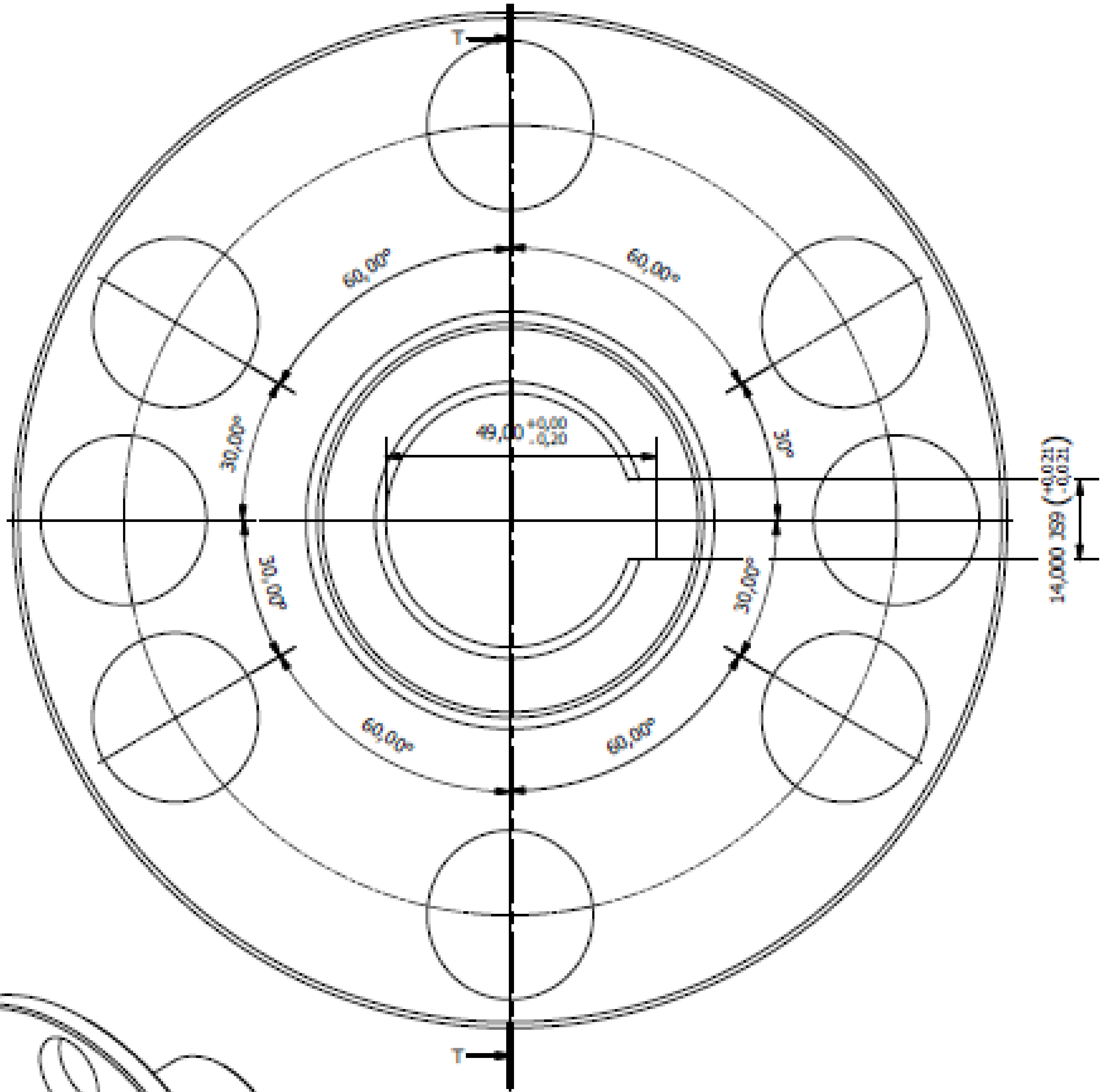
Components list		
Number	Quantity	Component
1	2	Bearing SYT 45 LTS
2	2	Spheric rolling bearing 22209
5	2	KTR Rigiflex 60 Flange-Key
7	1	Clamping table
13	1	Gear box
17	6	Screw M12 x 60
18	12	Washer M12
19	4	Nut M12
24	1	Adapter to gear box
30	2	Clevis pin - 30 x 60
33	1	Shaft
34	1	Adapter lever arm - shaft
38	2	Hanger support
41	2	Hanger
43	1	Lever arm
45	2	14x9x74mm key
46	1	16x10x76mm key
47	1	Torque transducer
48	2	Bearing base

Designed by	Revised by	Approved by	Date	Date
Iñaki Garro				03/05/2011

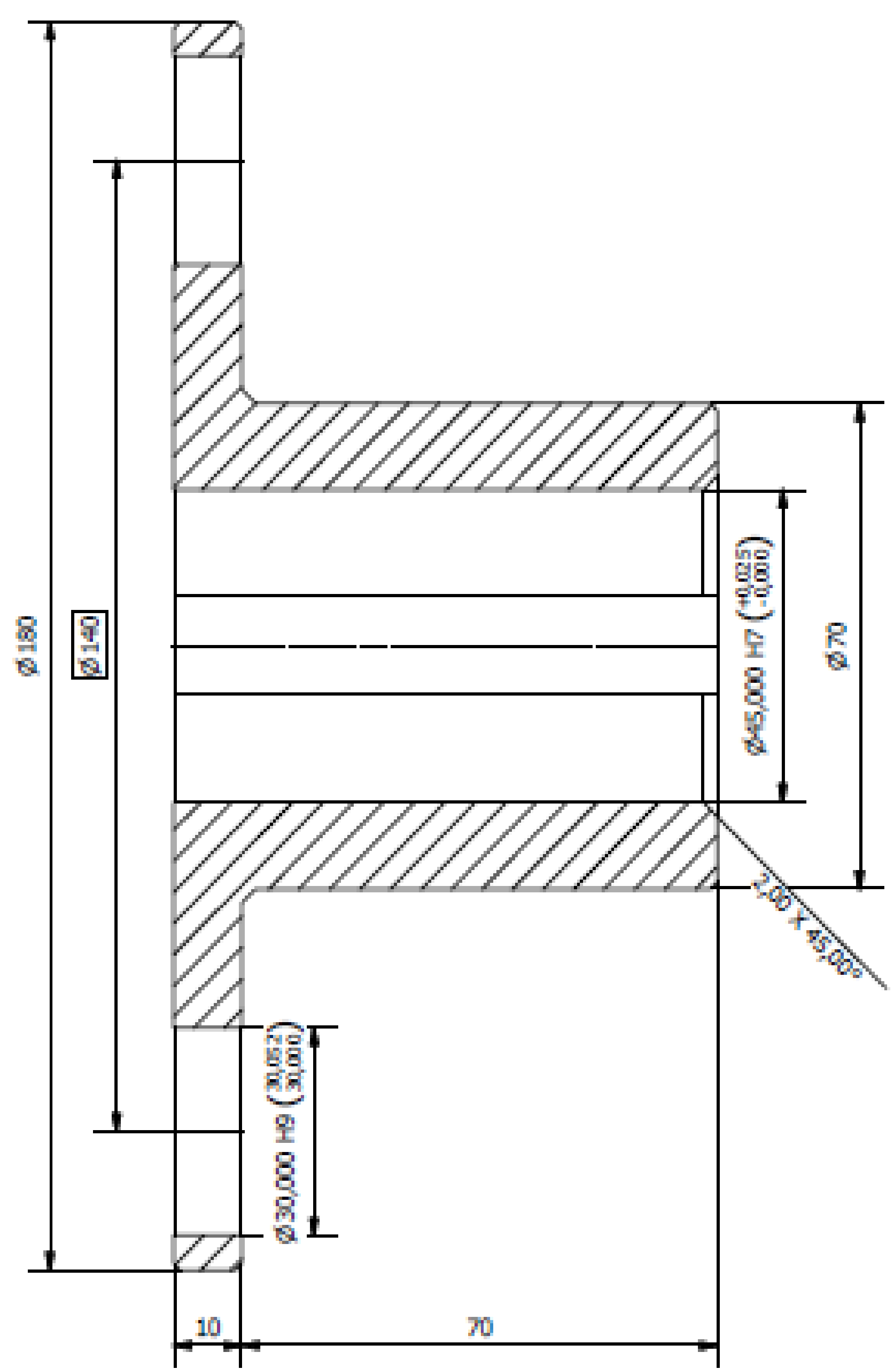
Vrije Universiteit Brussel	Torque calibration bench		
	Assembly	Edition	Sheet 14 / 14

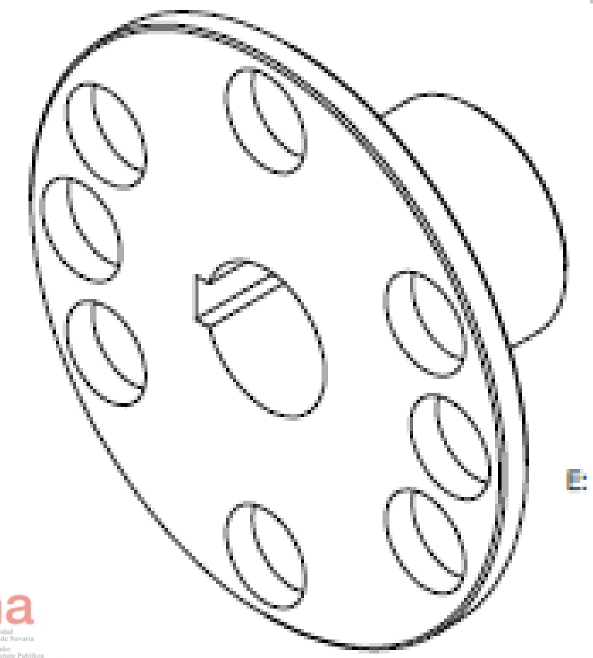


Designed by	Revised by	Approved by	Date	Escale 1:6	Date	Aluminium
Iñaki Garro					03/05/2011	
Vrije Universiteit Brussel			Torque calibration bench			
			Arm design (Quantity: 1)		Edition	Sheet 10 / 10



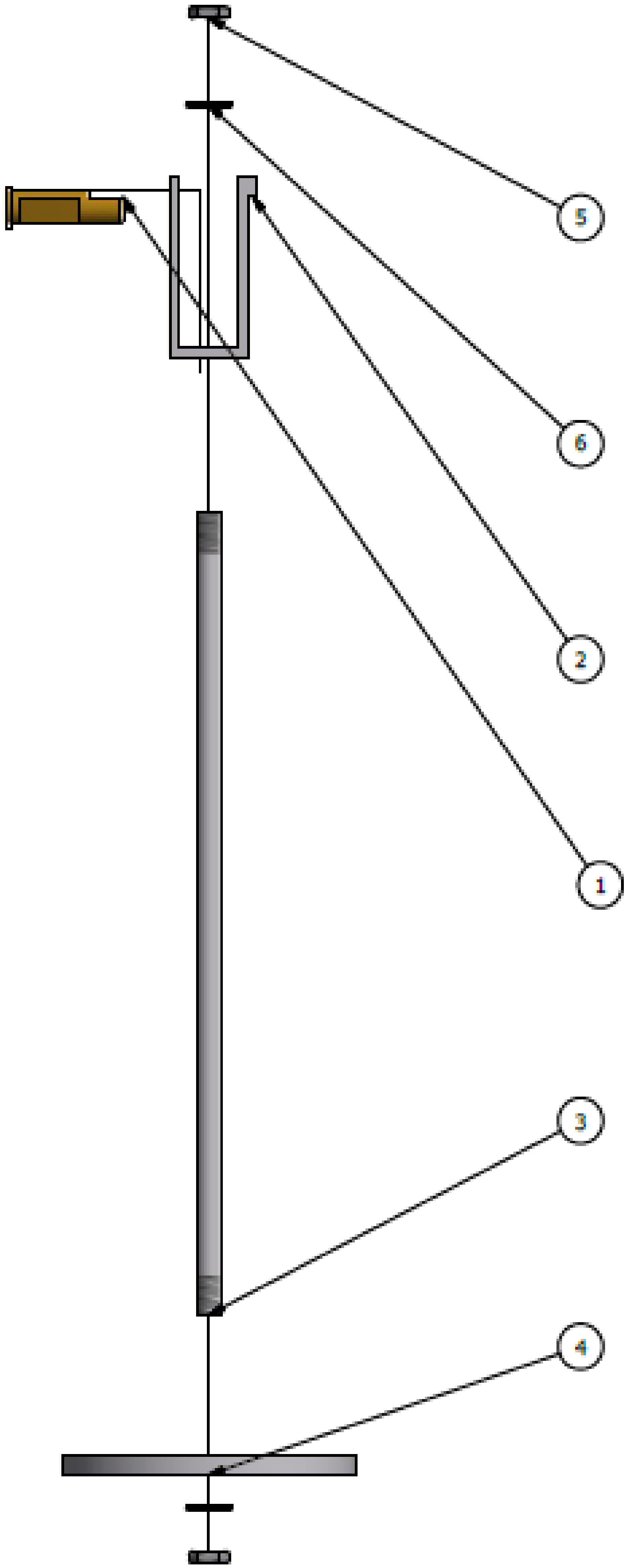
T-T (1 : 1)

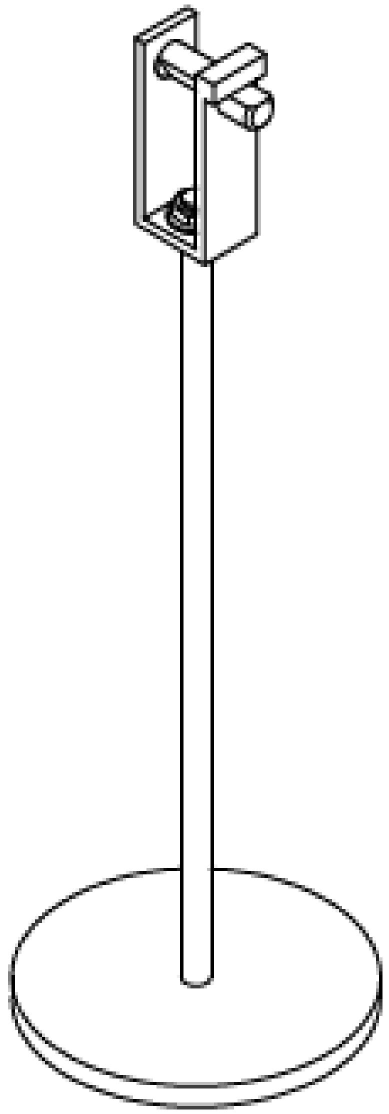




E: 1:2

Designed by	Revised by	Approved by	Date	E: 1:1	Date	Stainless steel
Iñaki Garro					03/05/2011	
Vrije Universiteit Brussel		Torque calibration bench				
		Adapter arm-shaft (Quantity: 1)			Edition	Sheet 15 / 15



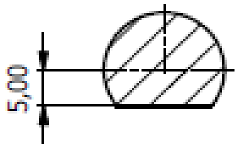


Parts List			
Element	Piece	Quantity	Comments
1	Clevis pin	1	d=16 mm x 60 mm
2	Hanger head	1	
3	Spout nipple	1	d=12 mm , 400 mm length
4	Hanger base	1	
5	Nut	2	M12
6	Washer	2	M12

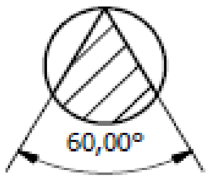
Designed by	Revised by	Approved by	Date	Date
Iñaki Garro				03/05/2011

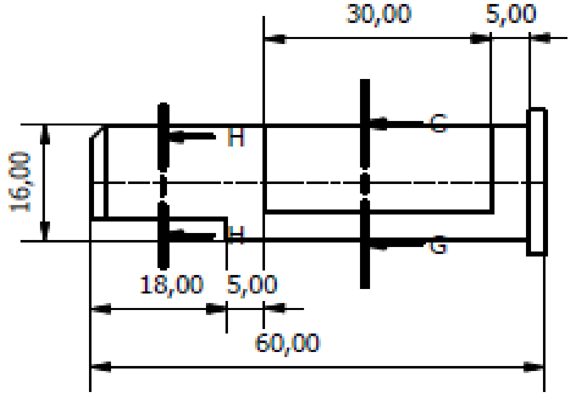
Vrije Universiteit Brussel	Torque calibration bench		
	Hanger (Quantity: 2)	Edition	Sheet 6 / 6

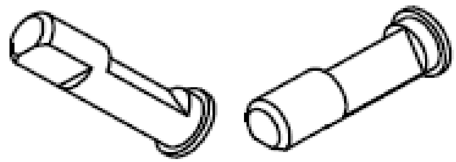
H-H (1 : 1)



G-G (1 : 1)



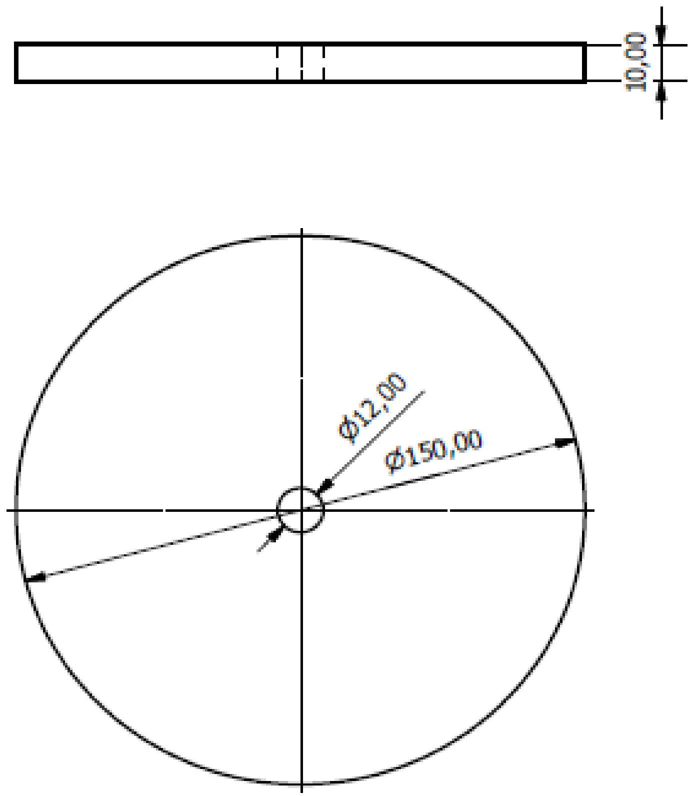


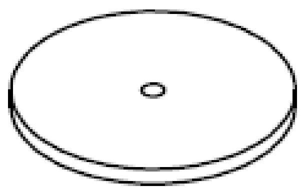


Escale 1:2

Machined from a clevis pin 20x60

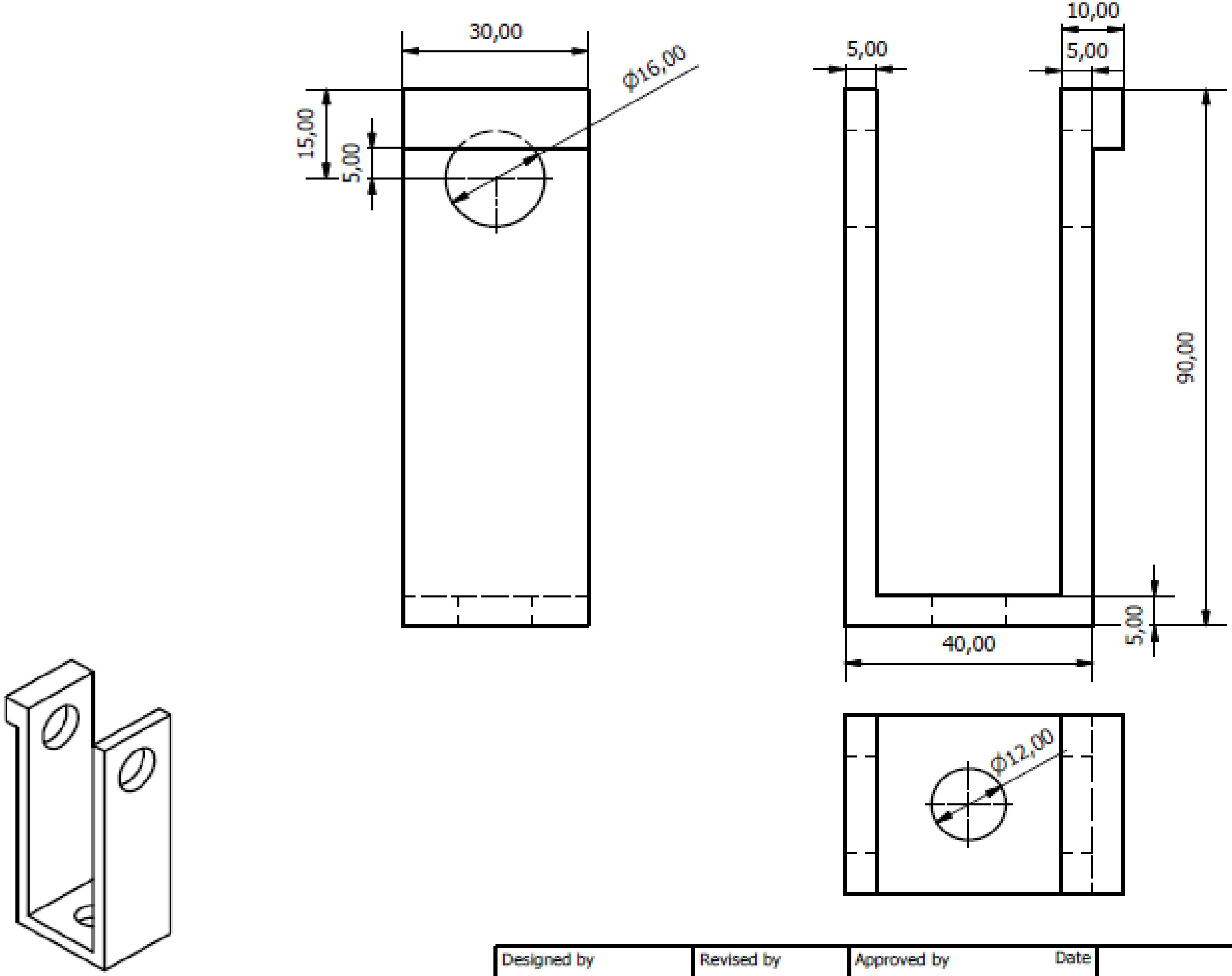
Designed by Iñaki Garro	Revised by	Approved by	Date	Escale 1:1	Date 03/05/2011	Stainless steel
Vrije Universiteit Brussel				Torque calibration bench		
				Hanger clevis pin (Quantity: 2)	Edition	Sheet 7 / 7



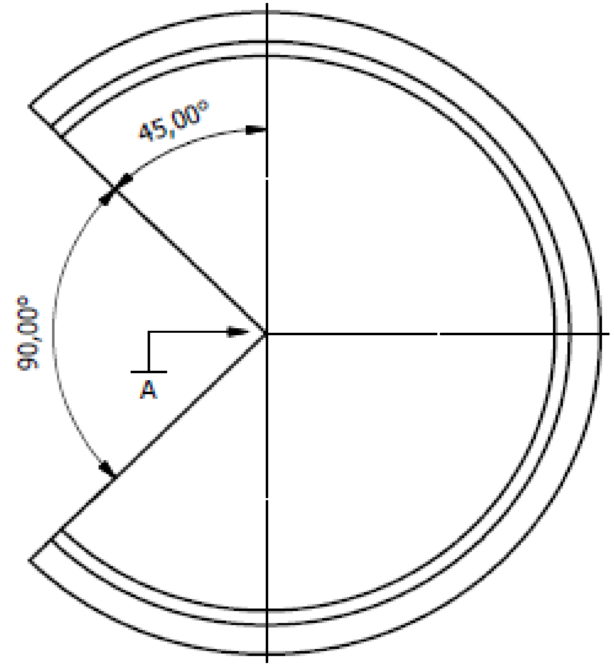


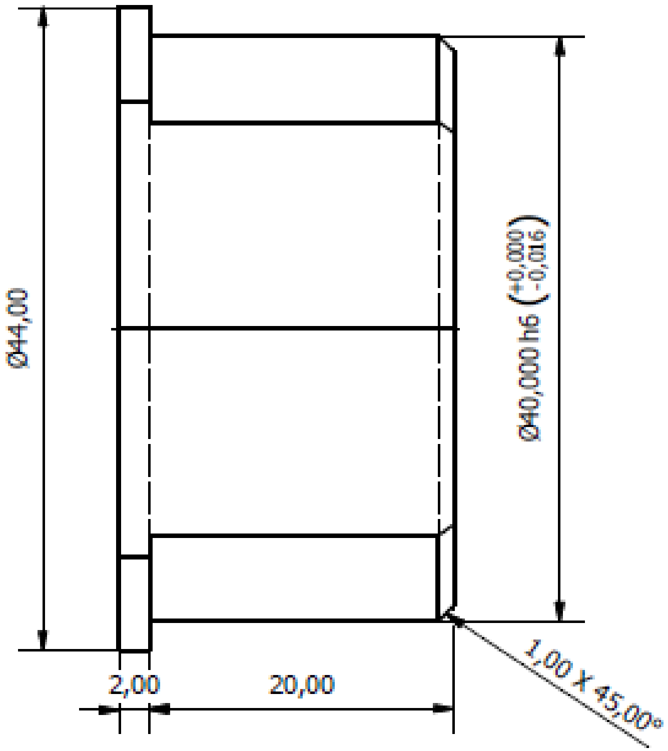
Escale 1:4

Designed by	Revised by	Approved by	Date	Escale 1:2	Date	Aluminium
Iñaki Garro					03/05/2011	
Vrije Universiteit Brussel			Torque calibration bench			
			Hanger base (Quantity: 2)		Edition	Sheet 9 / 9

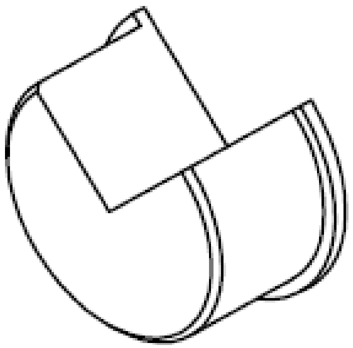


Designed by Iñaki Garro	Revised by	Approved by	Date	Date 03/05/2011	Aluminium
Vrije Universiteit Brussel			Torque calibration bench		
			Hanger head (Quantity:2)	Edition	Sheet 8 / 8



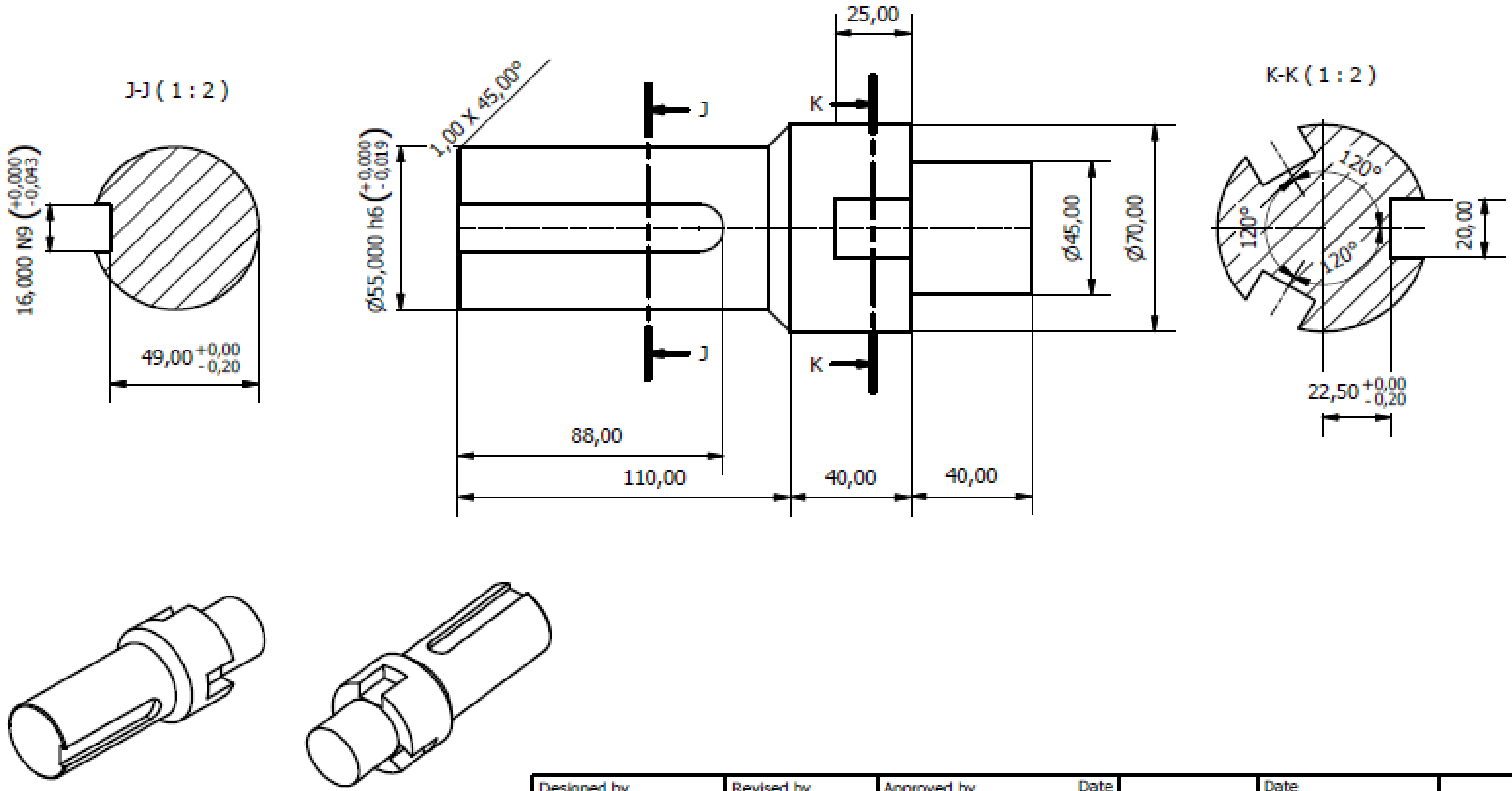


A: The sharpness of this 90º angle is important



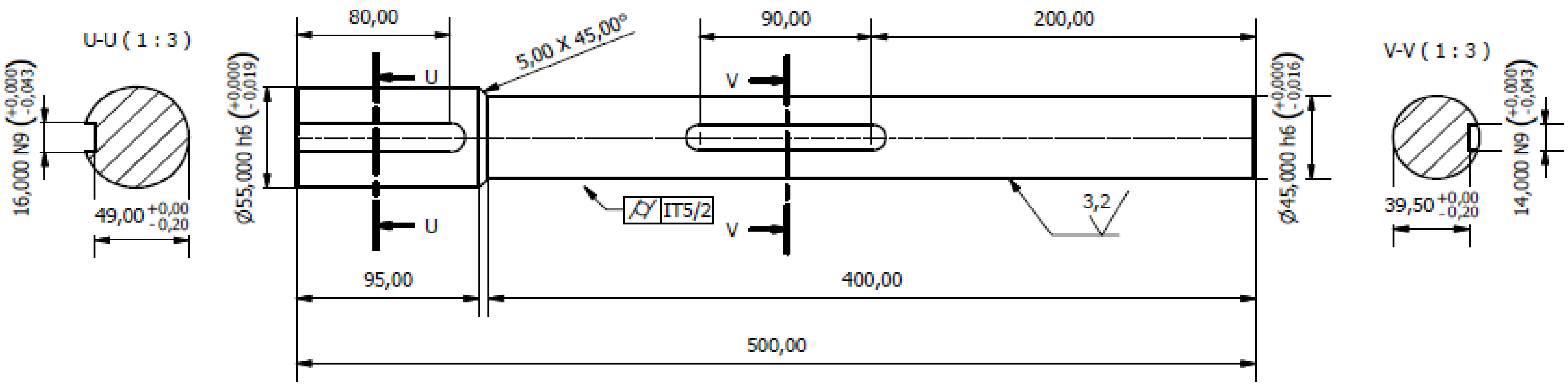
Escale 1:1

Designed by	Revised by	Approved by	Date	Escale 2:1	Date	Aluminium
Iñaki Garro					03/05/2011	
Vrije Universiteit Brussel			Torque calibration bench			
			Hanger holder (Quantity: 2)		Edition	Sheet 2 / 2



Escale 1:4

Designed by	Revised by	Approved by	Date	Escale 1:2	Date	Aluminium
Iñaki Garro					03/05/2011	
Vrije Universiteit Brussel			Torque calibration bench			
			Adapter shaft-table (Quantity: 1)		Edition	Sheet 4 / 4



Designed by	Revised by	Approved by	Date	Date	
Iñaki Garro				03/05/2011	Stainless steel

Vrije Universiteit Brussel

Torque calibration bench

Shaft (Quantity: 1)

Edition	Sheet
	16 / 16