The Manoeuvring Committee

Final Report and Recommendations to the 24th ITTC

1. INTRODUCTION

1.1 Membership

The 24th ITTC Manoeuvring Committee consisted of:

- Dr. Jakob Buus Petersen (Chairman). Formerly, FORCE/DMI, Denmark.
- Prof. Frederick Stern (Secretary). IIHR-Hydroscience & Engineering, U.S.A.
- Dr. Riccardo Broglia (from November 2003). Istituto Nazionale per Studi ed Esperienze di Architettura Navale, Italy.
- Dr. Maurizio Landrini (until June 2003). Istituto Nazionale per Studi ed Esperienze di Architettura Navale, Italy.
- Dr. Andres Cura Hochbaum. Schiffbau Versuchsanstalt Potsdam, Germany.
- Prof. Zu-Yuan Liu. Wuhan Transportation University, China.
- Dr. Pierre Perdon. Bassin d'Essais des Carènes, France.
- Prof. Key-Pyo Rhee. Seoul National University, Korea.
- Mr. Peter Trägårdh. SSPA Sweden AB, Sweden.
- Prof. Yasuo Yoshimura. Hokkaido University, Japan.

1.2 Meetings

The committee met four times:

- INSEAN, Italy, March 2003.
- FORCE/DMI, Denmark, October 2003.
- Seoul National University, Republic of Korea, April 2004.
- Bassin d'Essais des Carènes, France, November 2004.

A non-mandatory editorial meeting was also held 31st March to 1st April 2005, Potsdam, Germany.

1.3 Tasks and Report Structure

The following lists the 24th Manoeuvring Committee (MC) tasks with explanation of how these have been carried out.

- Review the state-of-the-art, comment on the potential impact of new developments on the ITTC and identify the need for research and development for predicting the manoeuvring behaviour of ships including high speed and unconventional vessels such as planing boats and catamarans. Monitor and follow the development of new experimental techniques and extrapolation methods.

State-of-the-art reviews are given regarding progress in systems (Section 2) and CFD (Section 3) based manoeuvring simulation methods; benchmark data (Section 4); high speed and unconventional vessels (Section 5); confined waters (Section 6); and manoeuvring standards and safety (Section 7). The reviews focus on the last three years, except for topics not covered in recent MC reports, which cover a longer period of time.

- Review the ITTC recommended procedures, benchmark data and test cases for validation and uncertainty analysis and update as required. Identify the requirements for new procedures, benchmark data, validation and uncertainty analysis and stimulate the research necessary for their preparation.

1. Improve the procedure 7.5-02-06-02 "Manoeuvring Captive Model Test Procedure", in particular by addition of a section on circular motion tests. There is required a set of validation data for captive model tests in form of time records of forces for a given ship and a given motion history (i.e. Planar motion mechanism).

   7.5-02-06-02 was improved for addition of circular motion tests. The MC found that the current uncertainty analysis (UA) for captive model tests included in 7.5-02-06-02 while discussing some of the important error sources, does not follow procedure 7.5-02-01-01 Uncertainty Analysis in EFD" and thereby provide quantitative uncertainty estimates. Therefore, the MC initiated an effort of developing UA procedures for captive model tests. Significant progress made as described in Section 8 with expectation that the 25th MC can complete the procedure. Benchmark data is reviewed in Section 4.

2. Continue work on procedures or guidelines for numerical applications in manoeuvring.

   The MC reviewed current systems and CFD based manoeuvring simulation methods and found that a wide variety of methods is used and there is a crucial need for validation at both model (i.e., using free model tests results) and full scale to distinguish their strengths and weaknesses; thus, premature for the development of procedures or guidelines for applications. Towards this goal, the 24th MC initiated organization of the "Workshop on Verification and Validation of Ship Manoeuvring Simulation Methods" http://www.simman 2008.dk/.

3. Further improve and update the procedure 7.5-02-06-01 "Manoeuvring Free-sailing Model Test Procedure" and include an uncertainty analysis, primarily linked to the position measurement. Validation data for free-sailing model tests are necessary for hull forms other than "Esso Osaka" hull.

   7.5-02-06-01 was improved and updated: however, progress on including UA for free model test was slower than anticipated. UA for free-model tests greatly facilitated by completion UA for captive model tests such that the 25th MC can compete both, as described in Section 8.

4. Application specific numerical methods should be sought for confined waters and bank effects as well as ship/ship interaction. Work should be conducted to improve the regression methods regarding confined waters.

   Section 3 and 6 review current status of computational methods and research for confined waters.

2. PROGRESS IN SYSTEM-BASED SIMULATION METHODS

Since the release of the Interim Standards for Ship Manoeuvrability by IMO (1993) and the final adoption of these standards (2002), ship designers are clearly faced with the design criteria for manoeuvrability and thus, procedure and tool for optimum hull form design should be established to satisfy the criteria under the basic demand of propulsive performance. As the IMO standard covers the course-keeping and yaw- checking ability as well as the conventional turning and stopping abilities, the prediction of course stability has become important.

For the prediction of the manoeuvring parameters such as advance, turning circle, tactical diameter, unstable loop width, overshoots of zigzag manoeuvres and so on, simulation techniques are required. There are many methods to predict the manoeuvring trajectories. Figure 2.1 demonstrates the typical methods for this purpose. When the database of manoeuvring parameters are provided from many full-scale trail results and free model tests, the manoeuvring parameters of a new designing ship can be estimated correcting from these database without simulations. These methods belong to the empirical “No Simulation” method. They are effective when the dimensions and frame lines of designing ship are closely similar to those of database’s. When the dimensions and frame lines of designing new ship is far from those of database, free model tests are necessary. In this case, the scale effects on the trajectories should be carefully taken account.

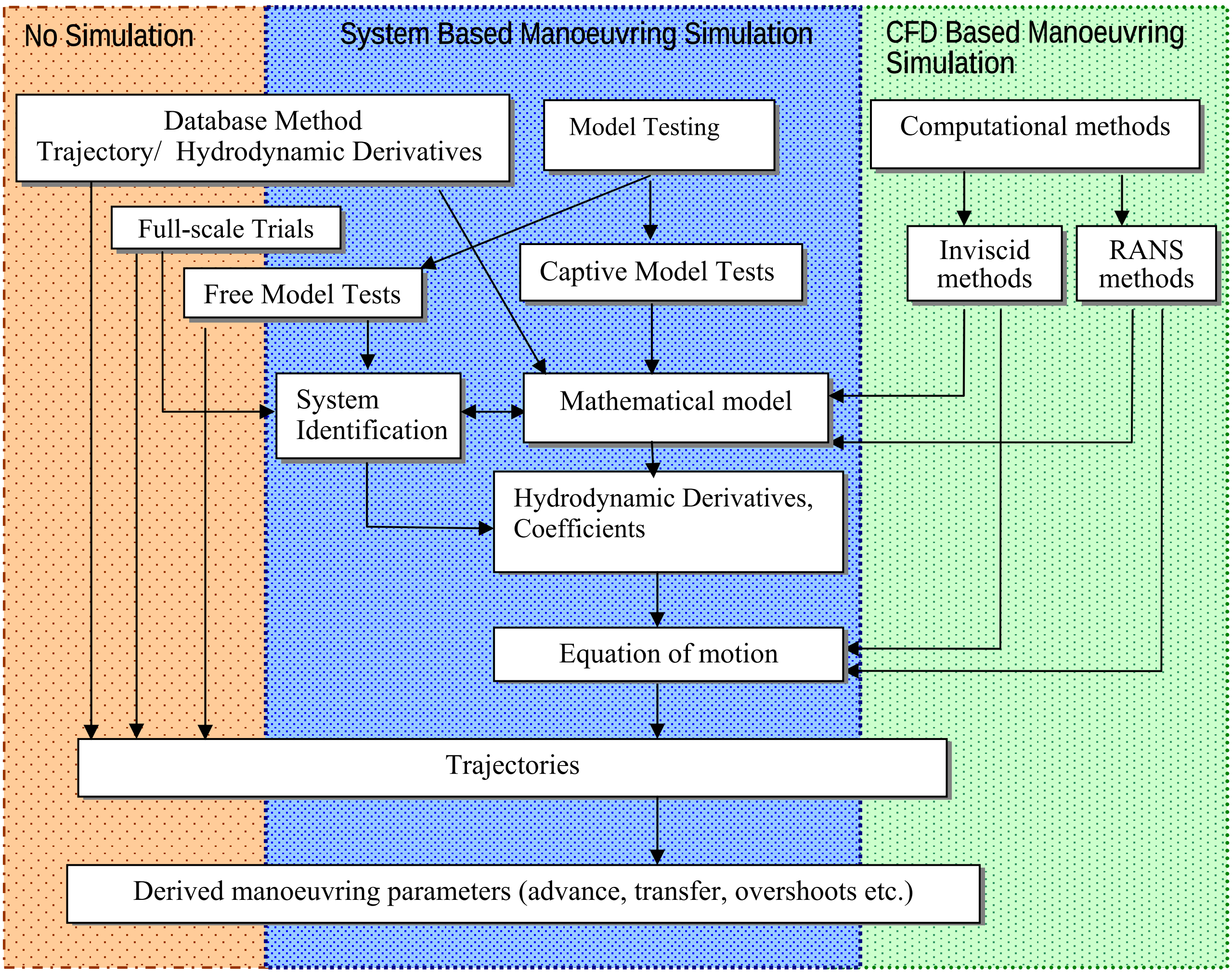


Figure 2.1- Manoeuvring prediction methods and definition of “System Based Manoeuvring Simulation”.

Obviously, the above mentioned methods cannot provide the manoeuvring parameters that are not included in the database. For this purpose, the simulation method is necessary. When the database of hydrodynamic forces and derivatives are provided and they are calculated by empirical formulas using ship's dimensions or frame lines, the trajectories can be simulated. If the dimensions and frame lines of designing new ship is far from those of database, it is necessary to supply the hydrodynamic forces and derivatives from the captive model test. For this kind of simulation, mathematical model is required. Although the hydrodynamic forces during manoeuvring motion have steady and very complex unsteady contributions, the expression of hydrodynamic forces in the mathematical model assumes that they just depend on velocity and acceleration components, according to the usual quasi-steady approach. The mathematical model for 3 degrees of freedom (DOF) is generally described as in the followings.

$$\left.\begin{aligned} m(\dot{u}_G - v_G r) &= X_G = X_{GA} + X_{GS} \\ m(\dot{v}_G + u_G r) &= Y_G = Y_{GA} + Y_{GS} \\ I_{zz}\dot{r} &= N_G = N_{GA} + N_{GS} \end{aligned}\right\} \tag{2.1}$$

or

$$\left.\begin{aligned} m(\dot{u} - vr - x_G r^2) &= X_M = X_A + X_S \\ m(\dot{v} + x_G \dot{r} + ur) &= Y_M = Y_A + Y_S \\ I_{zz}\dot{r} + m x_G(\dot{v} + ur) &= N_M = N_{GA} + N_{GS} \end{aligned}\right\} \tag{2.2}$$

where,
m : mass of ship
I_{ZZ} : moment of inertia of the ship about the vertical axis

Equation 2.1 is the equation of motion with the centre of gravity of ship: *G,* and the notation of u_G, v_G and r are velocity components at centre of gravity of ship, X_G, Y_G and N_G represent the hydrodynamic forces and moment acting on *G*. Meanwhile, Eq. 2.2 is the equation of motion referred to mid-ship: *M*, and the notation of *u, v* and *r* are velocity components at mid-ship, X_M, Y_M and N_M acting on *M*. x_G represents the location of *G* in *x*-axis direction. In each equation, the suffix "*A*" represents forces and moment by acceleration components such as $\dot{u}, \dot{v}, \dot{r}$, and the suffix "*S*" by velocity components *u, v, r* including rudder angle δ and propeller revolution *n*. For the expression of these hydrodynamic forces and moment, some polynomial functions with acceleration and velocity components are used. The coefficients of them correspond to hydrodynamic derivatives and can be obtained from:

a) Captive model test such as oblique towing test (OTT), rotating arm test (RAT), circular motion test (CMT) and planar motion mechanism (PMM) test.
b) CFD calculation
c) Identification to the free-model tests or full-scale trials
d) Database of hydrodynamic derivatives

In this report, this prediction method using hydrodynamic derivatives is defined as "System Based Manoeuvring Simulation Method". This method is widely used to predict the manoeuvring trajectories. However, as mentioned before, the unsteady character of the hydrodynamic forces makes it sometimes difficult to properly express the forces in the System Based Simulation Method. The rapid progress of Computational Fluid Dynamics (CFD) techniques makes it possible already to calculate complex hydrodynamic forces for steady and unsteady cases. This method is defined as "CFD Based Manoeuvring Simulation Method", and it is expected to be a useful method in future. CFD Based Manoeuvring Simulation Method is described in Section 3 in detail.

2.1 Model Testing

Free Model Tests. As mentioned above, free model tests are very effective as the manoeuvring parameters are directly obtained without simulation. The revised standard procedure described in "7.5-0.2-06-01" becomes useful for the tests. From these tests, the hydrodynamic derivatives and coefficients for simulations can be also obtained. The progress of this method is described in Section 2.4.

Captive Model Tests. Although the simulation based on captive model tests is the most traditional method, the test results as well as the dimensions of tested ship model recently tend to be kept confidential because of commercial matters. During these years, few data have been published regarding captive model tests.

Static Test: Rotating arm test (RAT) and Circular motion test (CMT) including oblique towing test are major static captive model tests. As the parameter such as drift angle and turning rate are constant and measuring time is enough long to average measured hydrodynamic forces, the obtained derivatives and hydrodynamic coefficients tend to be reliable. However, as one test produces results for one steady motion case only, it takes many times to carry out for the whole motion range. Therefore, few new hydrodynamic data are recently presented.

Yoshimura (2003) presented the hydrodynamic derivatives and coefficients including rudder and interactive forces with 5 fishing vessels using CMT. Using these coefficients, turning motions are well simulated.

Dynamic Test: PMM is very common as a dynamic captive model test. As the dynamic model test forces the continuous ship motion to the model, sufficient hydrodynamic derivatives including added mass and moment of inertia can be obtained from relatively few runs. It is efficient for the simulation. However, it is noted that the obtained hydrodynamic derivatives are somehow affected by memory effect that depends on the frequency of PMM, the chosen model speed, etc.

Simulations using PMM results are reported by Kim et al. (2003). They carried out tests with a 4-DOF PMM system for large container ship models, and obtained the hydrodynamic derivatives including the roll motion. Simulated turning trajectories and zigzag manoeuvres were predicted well comparing with free model tests as shown in Fig. 2.2.

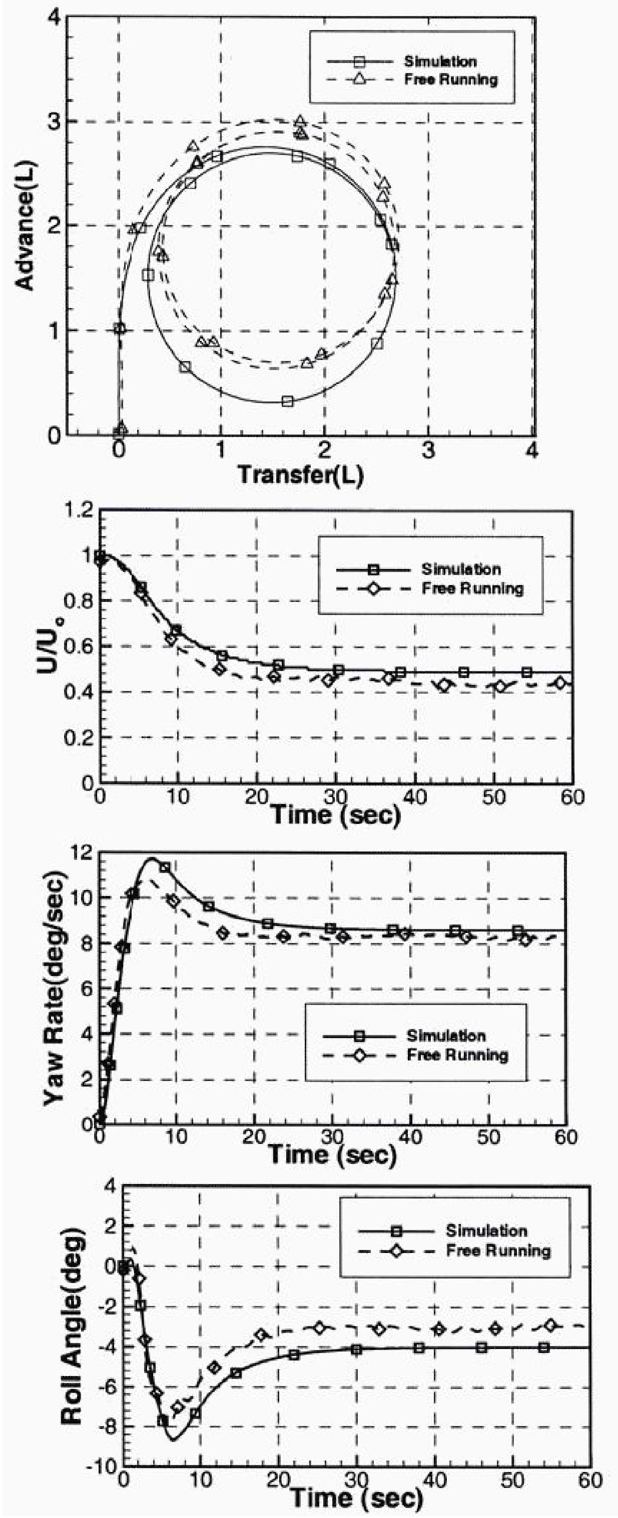


Figure 2.2- Comparison of 35° turning trajectories between simulated and free model test (taken from reference Kim et al. 2003).

Scattering Factors on Measured Hydrodynamic Forces: The measured hydrodynamic forces from a captive model test are scattering due to the experimental conditions and analyzing procedures. In order to avoid such scattering, the standard procedures are provided. With regarding to the scattering factors in the captive model test, Yoshimura (2002) summarized as shown in Fig. 2.3.

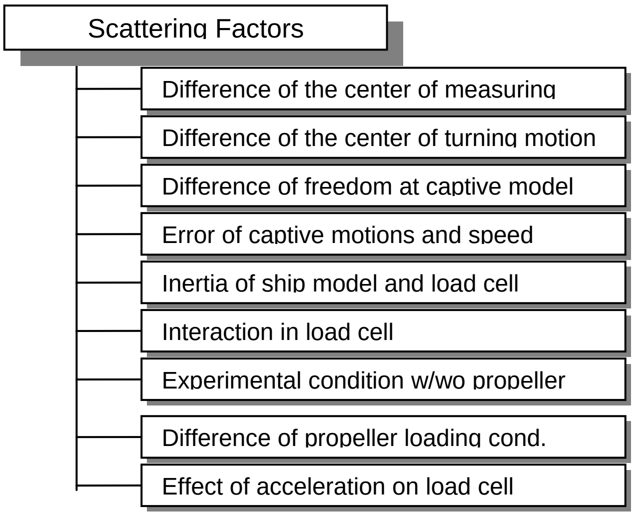


Figure 2.3- Scattering Factors on hydrodynamic forces (taken from reference Yoshimura 2002).

2.2 CFD Calculation

Recently, CFD computation is rapidly applied to the estimation for manoeuvrability. Detail of the CFD based simulation is reviewed in the next Section 3. In this section, the simulation through hydrodynamic derivatives using CFD is reviewed. Most of them, CFD is used for the prediction of course stability calculating the linear hydrodynamic derivatives Y_v, Y_r, N_v, and N_r.

Ishiguro and Ohmori (2003) tried to use CFD for the evaluation of manoeuvrability at the initial design stage, particularly proper estimation of directional stability for the IMO standard as demonstrated in Fig. 2.4. They used NICE CFD code to obtain hydrodynamic derivatives. Figure 2.5 shows an example of this simulation for zigzag manoeuvring motion using both computed hydrodynamic derivatives by CFD and measured one by captive model test (CMT), the results of which show a good agreement to each other.

Yamasaki et al. (2001) simulated trajectories for many tankers including SR-221 A, B, and C models using NICE CFD code. The computed linear hydrodynamic derivatives are compared with model test results (CMT) in Fig. 2.6, where the good correlations can be seen between CFD calculation and model tests.

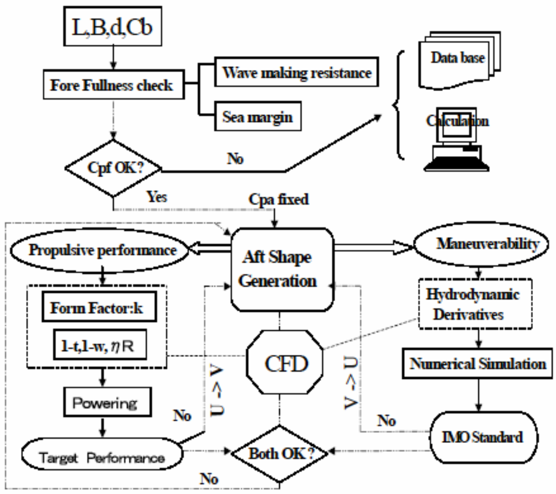


Figure 2.4- Hull form design system (taken from reference Ishiguro and Ohmori 2003).

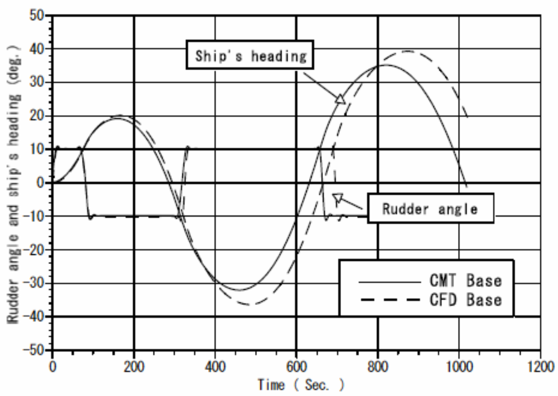


Figure 2.5- Comparison of Z-manoeuvre (taken from reference Ishiguro and Ohmori 2003).

In addition to the hydrodynamic derivatives, the calculation of rudder force is also important to predict the manoeuvring trajectories. Tanaka and Kimura (2003) computed the flow field around the stern by the NICE CFD code, and rudder force including interactive forces in the oblique conditions, then obtained the effective rudder inflow velocity and angle as shown in Fig. 2.7.

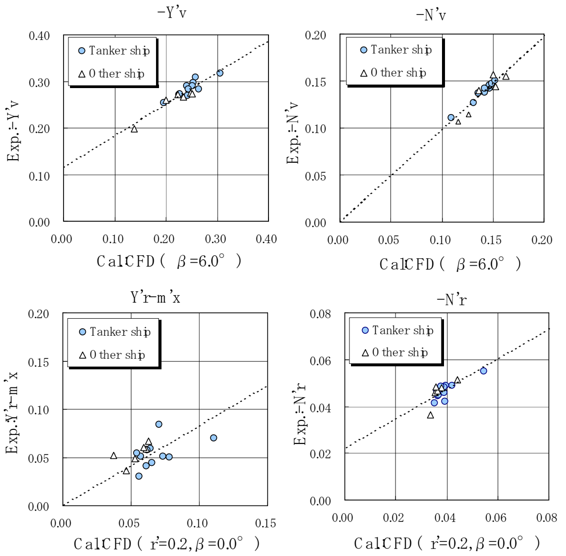


Figure 2.6- Linear derivatives, between experiment and CFD (from ref. Yamasaki et al. 2001).

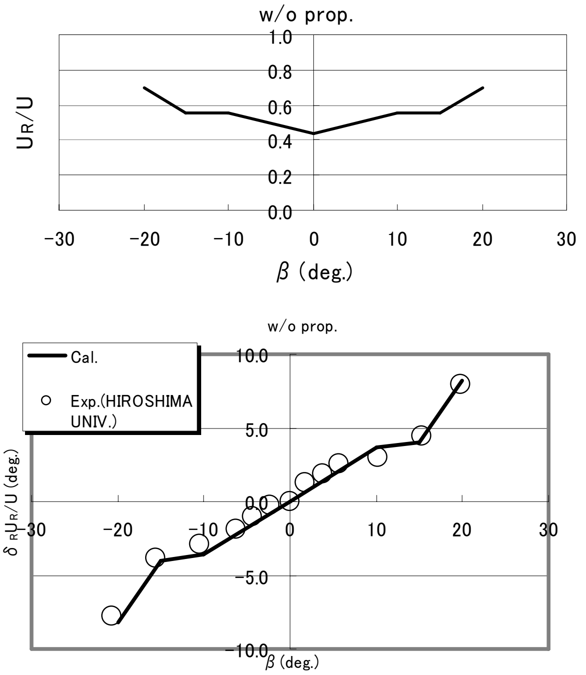


Figure 2.7- Effective rudder inflow velocity in oblique motion, between CFD and experiment (taken from reference Tanaka et al. 2003).

2.3 Database Method

The hydrodynamic derivatives and coefficients in simulation can be obtained by above mentioned model tests or theoretical calculations such as slender body theory or CFD. But at a ship design stage, a quick application will be needed to access the ship manoeuvring performance. From these points of view, data base method that expresses some simple formulae for hydrodynamic coefficients becomes very effective.

Oltmann (2003) reviewed the database method during the period of 1970 to 1993 as follows. Wagner Smitt (1970/71) regressed nearly 30 data sets from PMM tests. Norrbin (1971) investigated a similar database. In contrast to Wagner Smitt and Norrbin, Inoue et al. (1981) considered also non-linear coefficients and various load conditions including trim. Clarke et al. (1983) and Oltmann (1992) presented empirical formulas for the added mass coefficients. Kijima (1990, 1993) revised Inoue's formulas. Common to nearly all papers mentioned so far is that the corresponding regression formulas were developed during a time when full-bodied bulk carriers and tankers were the dominating ship types on the main shipping routes. Furthermore, all authors used global hull parameters like *Cb, L/d, B/d*, etc.

In the 1990, it proved so conclusively that the aft body has a considerable influence on the manoeuvring performance and the dynamic yaw stability. This holds in particular for container vessels whose aft hull forms were drastically changed from the viewpoint of the propulsion performance. To lend more weight to the aft body, the following parameter was introduced for an example from propulsion consideration.

$$\sigma_a = (1 - C_{wa})/(1 - C_{ps}) \tag{2.3}$$

where,
C_{wa} and C_{pa} : the water plane area coefficient and prismatic coefficient of the aft half hull.

Moreover, the linear derivatives have become important because of the demand of the course stability prediction in IMO standard. Many researches have been concentrated about the expression of them, as the linear derivatives strongly affected by the stern hull form particularly for full ships.

Artyszuk (2003) pointed out that the Kijima model gives slightly worse manoeuvring prediction, it is much more stable than the Inoue model. The major differences between both models are associated with the hull sway hydrodynamic force and the rudder flow straightening factor.

Kijima and Nakiri (2003) further revised the database formulas particularly for aiming the difference of stern hull shape. The formulas of whole hydrodynamic derivatives were obtained by the database that consists of 15 kinds of ships and their 48 loading conditions, and by theoretical investigation. The calculated hydrodynamic forces by the formulae are well estimated comparing from measured one, and simulated trajectories such as turning motion and zigzag manoeuvre are well predicted. Revised Kijima's formulae of linear derivatives are shown as the followings.

$$\left.\begin{aligned}
Y'_\beta &= \frac{1}{2}\pi\kappa + 1.9257(C_b B/L)\sigma_a \\
Y'_r - (m' + m'_x) &= \frac{1}{4}\pi\kappa + 0.052e'_a - 0.457 \\
N'_\beta &= \kappa\begin{bmatrix}15.0668\{d(1-C_b)/B\cdot e'_a K\}^2 \\ -23.819\{d(1-C_b)/B\cdot e'_a K\} + 1.802\end{bmatrix} \\
N'_r &= -0.54\kappa + \kappa^2 - 0.0477e'_a K + 0.0368
\end{aligned}\right\} \tag{2.4}$$

where,

e_a, e'_a : fullness of aft run

σ_a: aft sections fullness metric

K : form factor

they are described as,

$$e_a = \frac{L}{B}(1 - C_{pa}),$$

$$e'_a = e_a \Big/ \sqrt{\frac{1}{4} + \frac{1}{(B/d)^2}},$$

$$\sigma_a = \frac{1 - C_{wq}}{1 - C_{pa}},$$

$$K = \left(\frac{1}{e'_a} + \frac{1.5}{L/B} - 0.33\right)(0.95\sigma_a + 0.40)$$

Lee et al. (2003) presented another database formula. The database was obtained from PMM tests for modern ship hulls including various ship types and drafts. For more precise manoeuvring analysis, a simple parameter representing stern hull form is introduced in the formulae. The ranges of ship particulars are *Cb*: 0.55 ~ 0.87, *d/L*: 0.022 ~ 0.071, *L/B*: 5.0 ~ 8.8 and *Cb*(*B/L*): 0.075 ~ 0.166.

$$\left.\begin{aligned}
Y'_v &= -(0.145 + 2.25(d/L) - 0.2\Delta_{SR}) \\
Y'_r - m &= -(0.282 + 0.1\Delta_{SR}) \\
&\quad + (0.0086\Delta_{B/L} + 0.004)(L/d) \\
N'_v &= -(0.222 + 0.1\Delta_{SR}) + 0.00484(L/d) \\
N'_r &= -(0.0424 - 0.03\Delta_{SR}) \\
&\quad - (0.004\Delta_{Cb} - 0.00027)(L/d)
\end{aligned}\right\} \tag{2.5}$$

$$\begin{aligned}
\Delta_{Cb} &= 1 - C_b/P \\
P_{Cb} &= 1.12(d/L) + 0.735 \\
\Delta_{B/L} &= 1 - (B/L)/0.18 \\
\Delta_{SR} &= 1 - S_R/P_{SR} \\
P_{SR} &= 28.7\nabla' + 0.54 \\
S_R &= B_{P07}/B_{PS} \\
\nabla' &= \nabla/L^3
\end{aligned}$$

The linear hydrodynamic derivatives are also affected by the ship's trim. Kijima et al. (1990) proposed the following correction formulae.

$$\left.\begin{aligned}
Y'_\beta &= Y'_{\beta 0}(1 + (25C_b B/L - 2.25)\tau') \\
Y'_r - (m'_x + m') &= \{Y'_{r0} - (m'_x + m')\} \\
&\quad \times \{1 + (571p^2 - 81p)\tau'\} \\
N'_\beta &= N'_{\beta 0}(1 - \tau') \\
N'_r &= N'_{r0}\{1 + (34C_b B/L - 2.6)\tau'\}
\end{aligned}\right\} \tag{2.6}$$

where,
τ'=trim/d_m
$Y'_{\beta 0} = 0.5\pi k + 1.4Cb/(L/B)$
$(Y'_r - m'_x)_0 = 0.5Cb/(L/B)$
$N'_{\beta 0} = k$
$N'_{r0} = -0.54k + k^2$
$p = d(1 - C_b)B/L$
$k = 2d/L$

Yoshimura et al. (2003) show that Kijima's trim corrections for linear derivative are insufficient for the large trimmed ships. They introduced the empirical formulas for fishing vessels based on the obtained database. These formulae are well predictable particularly large trim by the stern.

$$\left.\begin{aligned} Y'_\beta &= Y'_{\beta 0}\left(1 + 0.6\tau'^2\right) \\ Y'_r - m'_x &= (Y'_r - m'_x)_0\left(0.4 + 1.8\tau'^2\right) \\ N'_\beta &= N'_{\beta 0}(1 - 0.9\tau') \\ N'_r &= N'_{r0} \end{aligned}\right\} \tag{2.7}$$

2.4 System Identification Method

Hydrodynamic derivatives are usually obtained from the measured hydrodynamic forces. In the system identification method, they can be obtained from the measured ship motion and applied rudder angle. Since the ship motion is identified accurately by this method, it is well used for the design of control and navigation systems. System identification method can be defined as a systematic approach to find a model of unknown system from the given input-output data. For the successful system identification, three items should be properly selected or designed; mathematical model of a system, input-output data and parameter estimation scheme.

Modelling of Manoeuvring Motions. There have been many researches on the modelling of manoeuvring motions of a ship, and those can be well described by the various mathematical models such as the regression type models or the modular type models.

Aryszuk (2003) analysed ship manoeuvring equations for the ESSO OSAKA tanker in the case where both full scale motion data and experimental scale model force measurements are combined together.

Approaches for modelling of ship manoeuvring motion in different ways have been investigated by several researchers. Hess and Faller (2000) presented an improved Recursive Neural Network (RNN) manoeuvring simulation model for surface ships. Inputs to the simulation, cast in the form of forces and moments, were redefined and extended in a manner that more accurately captures the physics of ship motion.

Moreira and Soares (2003) presented also a Recursive Neural Network (RNN) manoeuvring simulation model for surface ships. They set the inputs to be the orders of rudder angle and ship's speed, and the recursive outputs to be velocities of sway and yaw, and concluded that the RNN is a robust and accurate tool for manoeuvring simulation.

Estimation of Manoeuvring Coefficients. As the system identification method is an approach to find a model of unknown system, estimation of manoeuvring coefficients can be said as an application of system identification. Several schemes were suggested for the estimation of manoeuvring coefficients. Oltmann (2003) presented the state of the art regarding empirical regression formulas for linear sway and yaw damping coefficients, and suggested a regression approach for the formulation of damping coefficients which focuses on the aft hull form of a ship. Viviani et al. (2003), and Depascale et al. (2002) tried to obtain the hydrodynamic derivatives from the standard trial data in order to be able to extract data from the wide experimental database available with sea trial and free model tests. Viviani et al. (2003) presented two different procedures for the identification of the hydrodynamic derivatives were presented, with the comparison of their merits and shortcomings. The scheme of the identification procedure is shown in Fig.

2.8. The procedures allow obtaining values of the coefficients in a reasonable agreement with experimental data derived from PMM tests.

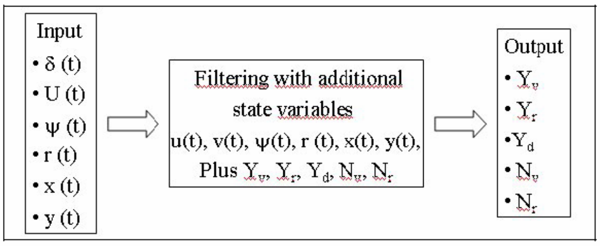


Figure 2.8- Scheme of identification procedure with filters (taken from reference Viviani et al. 2003).

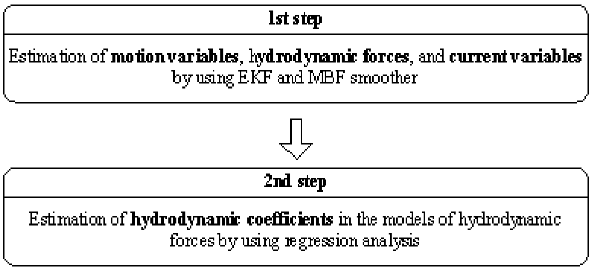
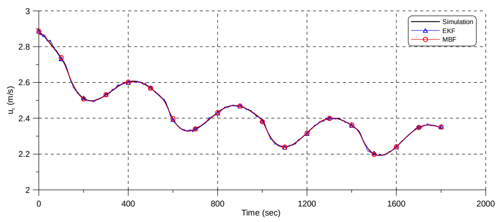
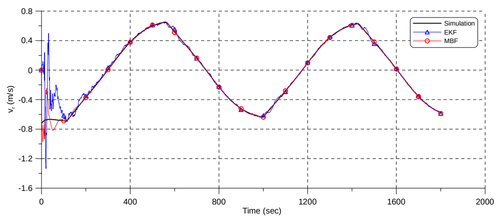


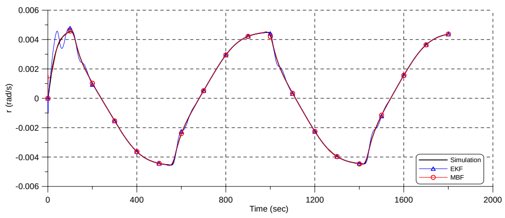
Figure 2.9- Schematic procedure of EBM technique (taken from reference Yoon et al. 2003).



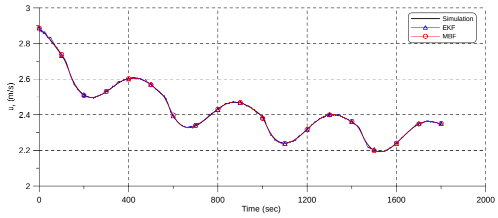
(a) Relative surge velocity



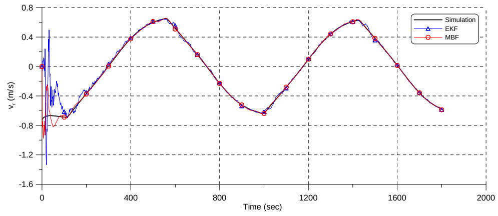
(b) Relative sway velocity



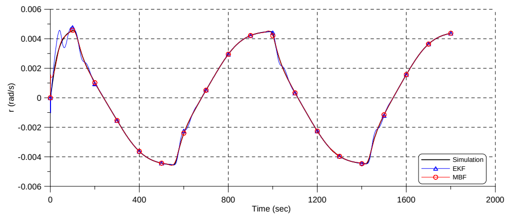
(c) Yaw rate



(d) Relative surge velocity



(e) Relative sway velocity



(f) Yaw rate

Figure 2.10- Schematic procedure of EBM technique and Identified 20°/20° zigzag trajectories (ESSO OSAKA, taken from reference Yoon et al. 2003).

Yoon et al. (2003) suggested Estimation-Before-Modelling (EBM) technique for the estimation of manoeuvring coefficients. In the EBM technique, the extended Kalman Filter and modified Bryson Frazier smoother are used to estimate motion variables hydrodynamic force, and the speed and the direction of current, with given sea trial data. With the estimated forces and environmental effects, the ridge regression method is used to estimate the manoeuvring coefficients. Figure 2.9 shows the schematic procedure of EBM technique. They used this technique for the estimation of the manoeuvring coefficients of ESSO OSAKA with simulated sea trial data as shown in Fig. 2.10.

3. PROGRESS IN CFD-BASED MANOEUVRING SIMULATIONS

A primary conclusion of the Ship Hydrodynamics CFD Workshop held in Tokyo in 1994

(Kodama et al., 1994) was that the better codes performed reasonably well for steady flow resistance and propulsion, such that extensions for unsteady ship motions, manoeuvring and industrial applications were therefore warranted. The present review indicates that great progress has in fact been made towards this goal in spite of the difficulties related to time accurate Reynolds Averaged Navier-Stokes (RANS), six degree of freedom (DOF), inclusion of appendages and propulsors and environmental effects such as waves and shallow water, all of which are required for such applications. Nonetheless, as also indicated by the present review considerably more progress is needed to achieve the ultimate goal of simulation based design (SBD). In particular, acceptance of SBD by industry requires credibility of simulations, which can only be achieved through verification and validation (V&V) for practical geometries and conditions and through international workshops on benchmarking of CFD capabilities for ship hydrodynamics (most recently, Larsson et al., 2000 and Hino, 2005), and successful CFD based designs. V&V studies and benchmarking of unsteady RANS for ship hydrodynamics remains a challenge due in part to lack of available EFD validation data, especially for ship motions and manoeuvring. In this regard, the last CFD Workshop Tokyo 2005 (see Hino, 2005) is an important step because an extensive validation for a ship in steady oblique motion and for a ship in head waves has been included for the first time.

The present review covers forward speed diffraction/radiation and ship motions, static manoeuvres, dynamic manoeuvres, trajectories, and high speed and restricted water and references over 60 CFD papers. In fact, the progress has been so significant that the MC decided to have separate Sections reviewing Systems Based Manoeuvring Simulation Methods (Section 2) and the present Section. The former refers to methods based on computerized systems, whereas the latter refers to methods based on formulation and solution of the physics-based CFD initial boundary value problem for ship hydrodynamics applications, as explained previously in the Section 2 introduction. Inviscid Methods are also reviewed; since, they continue to play a large role in systems based manoeuvring simulation methods.

As CFD matures towards SBD, the traditionally separate Naval Architecture fields of resistance and propulsion, seakeeping, and manoeuvring merge and Sections 3.2 and 3.3 may have some overlap with the 24th ITTC Resistance and Seakeeping Committee Reports. For instance, studies about RANS predictions for a ship in waves have been included and discussed. However, herein focus is specifically on applicability to manoeuvring and therefore justified. Main reasons for the slow widening of RANS techniques for manoeuvring simulation have been the need of an accurate model for the propeller effect on the flow and especially the need to turn the rudder in the numerical grid during the simulated manoeuvre. However both have been successfully implemented in some applications already and will be standard soon. As a previous step, simulations of prescribed dynamic manoeuvres resembling PMM tests are expected to yield hydrodynamic coefficients with sufficient precision for classical manoeuvring simulations. Because accuracy requirements for manoeuvring predictions are in practice less exorbitant than for resistance and propulsion, the methods treated here are expected to become practical tools very soon.

3.1 Inviscid Methods

General Study. Lee and Kerwin (2003) developed a high order panel method based on B-spline representation for both the geometry and the solution of the flow around two dimensional lifting bodies. The influence functions are separated into singular and non-singular parts. Through a de-singularization process, the accuracy of the present method increased without limit to any order by selecting a proper numerical quadrature. Kara and Vassalos (2003) studied solutions for bodies with forward speed directly in the time domain using Neumann-Kelvin method. The exact initial boundary value problem was linearized using the free stream as a basic flow, replaced by the boundary integral equation applying Green's theorem over the transient free surface Green function. Zhang et al. (2003) proposed an unsteady vortex lattice method and panel method for unsteady performance of the propeller and rudder system. An iterative procedure is needed until thrust and torque coefficients converge. Zhang et al. (2004) adopted non-uniform rational B-spline (NURBS) to express the geometry of the ship hull and the manoeuvring hydrodynamics of a Wigley ship at an angle of incidence using an approximate Kutta condition.

Wind and Waves. Tao and Zhang (2002) derived non-linear motion equations of surf-riding from 3D manoeuvring motion equations that take into account the wave-exciting term based on the combination of manoeuvre and seakeeping theory. By simplified the non-linear equations into linear equations, they calculated the broaching area of a destroyer. Funaji et al. (2003) performed numerical simulations of turning motion and zigzag manoeuvres to evaluate the effect of external disturbances such as wind and waves on the manoeuvring performance. Wind forces were calculated by Toshifumi Fujiwara's method and the wave forces were calculated by a singular point distribution method. Yasukawa and Kose (2003) presented a practical simulation method for predicting ship stopping manoeuvres in wind and waves. Zhu et al. (2004) established a prediction method for ship manoeuvring in regular waves based on the MMG model in calm water, and the regular wave forces were directly added to the manoeuvring motion equations as exciting forces. Strip method was used to calculate added masses, radiation and Froude-Krylov forces.

High Speed and Restricted Water. Rigby et al. (2001) used Rankine-Source method to study about the ship running in deep water in different speeds which exactly simulated the Millward and Bevan (1986)'s model geometry and experimental conditions, in order to test the capabilities of their numerical method in predicting the flow past the ship hull for higher *Fn* (up to 1.0). Their solver gives a good agreement to a certain *Fn* in wave making resistance, and after that, it gives lower values compared to EFD values. Sahoo and Doctors (2003) applied the potential theory based solver SHIPFLOW to restricted depth condition in order to compute the wave resistance. The results showed that the decrease in resistance at supercritical speed and the increase at high sub critical speeds became more pronounced as the depth/length ratio decreases. Adil et al. (2004) studied the effects of depth and forward speed on the hydrodynamic coefficients of ships with parabolic shape using a modified Green function and strip theory to calculate 2D and 3D coefficients. Both depth and forward speed have considerable effects on the hydrodynamic coefficients. Chun and Zao (1998) performed a similar study but manoeuvrability was taken into consideration. They did simulations using ESSO OSAKA in manoeuvring motion at finite speed in shallow water by adopting slender body theory and matched asymptotic expansion method. The results showed good agreement with experimental data. Kijima (2004) studied details of the manoeuvring characteristics in shallow water. Results showed that hydrodynamic forces and moment acting on a ship changed remarkably depending on the water depth. In terms of estimating manoeuvring motion, the turning circle becomes larger if the depth of water shallows. From comparisons between numerical and measured results in terms of both force and motion, an approximate formulae is accurate to estimate manoeuvring motion in shallow water. The most important factor on the difference of manoeuvring characteristics in shallow water, lies in the position of the acting point of both yaw damping force and sway damping force, which depend on ship's form.

3.2 Forward Speed Diffraction/Radiation and Ship Motions

Azcueta (2004) simulated the free surface flow around high speed Littoral Combat Ships (LCS) combining the commercial RANS code COMET with a 6-DOF motion integration. A two-phase VOF-technique for the free surface is used. Coarse grids have been used to predict the resistance curve at model scale for Froude numbers up to 0.9 taking dynamic lift and trim into account. The agreement with experiments is surprisingly good. Forces and motions for a LCS in regular head waves were also predicted. Azcueta (2002) shows results for the Series 60 ship moving straight ahead and obliquely. Wave breaking at the bow of a blunt-bow ship was simulated on three grids. The agreement of the wave profile with experiments is good. Simulations for a sailing boat are shown.

Cura Hochbaum and Vogt (2002) presented a method to predict ship motions and the free surface viscous flow around the ship. The motion equations in 6-DOF are coupled with an own RANS code for this purpose. A two-phase Level Set technique is used for the free surface. Good agreement of predicted waves with theoretical results is shown for single and for superimposed harmonic waves. The method is applied for a container ship in regular head waves. Forces and moments agree well with measurements. The method was able to predict the wave breaking at the bow observed during tests.

Klemt et al. (2003) also used COMET to predict flow induced forces and body motions, implementing the motion equations via user-coding. An implicit procedure is used for coupling the governing equations of body and fluid. An overlapping grids technique allows for large body motions. Results for a RoRo ship moving in regular head waves show good agreement of predicted heave and pitch motions with measurements. The method is described in Klemt (2005). A two-phase VOF-technique is used to calculate the free surface. Forces on the bow door of a RoRo ferry and the relative motion between bow and undisturbed wave agree well with experiments. Computed RAOs of heave and pitch motions are also in good agreement with measurements.

Luquet et al. (2004) presented a method for calculating wave-body interactions. Incident waves are described explicitly with a nonlinear potential flow model. Thus, only the vicinity of the ship has to be discretized with high resolution. A tracking method is used for the free surface. Results for the DTMB 5512 travelling in fixed condition in regular head waves were compared with experiments. Results on 3 grids show good convergence behaviour. Time histories of forces are predicted well. However, the computed wave pattern shows strong damping.

Miller et al. (2002) studied forced roll motions of a 3D cylinder with bilge keels at a range of roll frequencies and amplitudes. The computations have been performed with the RANS code UNCLE for the immersed and an emerged body configuration, with and without forward speed at different model scales. Predicted forces on bilge keels for a low frequency and moderate roll amplitude agree well with experiments. Snapshots of the transverse velocity at a cross section of the cylinder agree well with PIV measurements for the immersed case.

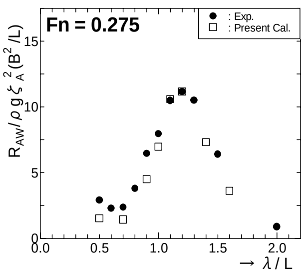


Figure 3.1- Measured and predicted RAOs of added resistance for the containership SR108.

Orihara and Miyata (2003) presented a method to predict the added resistance of practical hull forms based on the RANS code WISDAM-X. The free surface is determined with a single-phase Density Function Method. Simulations were done on an overlapping grid system for a container ship in regular head waves. Results on three different grids showed good convergence behaviour. The predicted RAOs of motions and added resistance agree well with measurements, Fig. 3.1.

Park et al. (2001) investigated the interactions of nonlinear waves with fixed bodies. The free surface is captured with a single-phase Density Function Method. The waves are generated prescribing the velocities at the inlet of the computational domain according to the motion of a wavemaker. Simulations were done at model scale for nonlinear waves of different periods running on a fixed vertical cylinder. The results agree reasonable well with experiments and with results of potential codes.

Weymouth et al. (2003) extended the RANS code CFDSHIP-IOWA to predict heave and pitch motions of ships. A single-phase free surface tracking method is used. Results for a Wigley-like hull in regular head waves are compared with measurements for a range of Froude numbers, wavelengths and amplitudes. A Fourier analysis of the free surface elevation yielded harmonic amplitudes and phases in good agreement with experiments. The paper suggests that RANS methods offer a big potential for problems involving large motion amplitudes and nonlinear effects.

Wilson and Stern (2002) extended the RANS code CFDSHIP-IOWA to predict ship motions. The method was applied to the flow around a combatant in calm water and carrying out a prescribed sinusoidal roll motion at different frequencies. Computed results are discussed and the obtained added roll moment of inertia and the roll damping coefficient are compared with measurements for a fishing vessel and with 2D computational results for a square.

Wilson et al. (2004) presented a single-phase Level Set method implemented in the RANS code CFDSHIP-IOWA. The solver has been extended to work on overset grids. The method was applied to simulate the flow around the DTMB 5512 model in calm water at different Froude numbers and in regular head waves as well. Computed results are discussed and in part compared with experiments showing a good overall agreement. In another application the capability of the method to deal with overturning waves was demonstrated, Fig. 3.2.

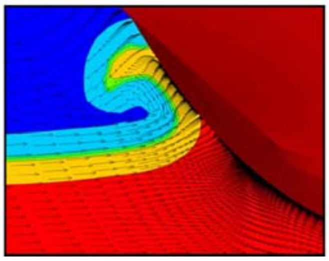


Figure 3.2- Level set contours and velocity field at the bow centre plane of a landing craft.

CFD WORKSHOP TOKYO 2005. In Test Case 4 the flow around the model of a combatant moving in regular head waves was simulated. The results of the participating groups were very encouraging. Experimental data allowed for validating the computed time histories of the forces on the hull, and the free surface elevation and nominal wake at several instants. A Fourier analysis was performed to compare amplitudes and phases of the 0^{th} and 1^{st} harmonics of these results with those obtained from measurements, Longo et al. (2002).

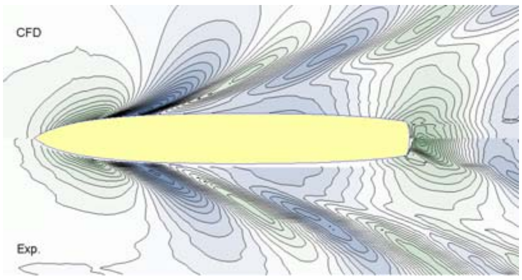


Figure 3.3- 0^{th} harmonic amplitude of wave elevation. Experiment and prediction.

The time histories of forces and pitch moment were predicted well, especially by Deng et al. and Luquet et al., see Hino (2005). The wave pattern and corresponding Fourier coefficients have been predicted particularly well by Cura Hochbaum and Pierzynski, Fig. 3.3, and by Carrica et al. The wake field and its

0^{th} and 1^{st} harmonic amplitudes and phases were also predicted well, especially by Deng et al. In short, the agreement of the results with the measurements was in part impressive, but this was not always achieved by one and the same code. The used methods are almost mature for practical application and will be applied for manoeuvring tasks in the near future.

3.3 Static Manoeuvres

Tahara et al. (2002) presented a detailed comparison of computational and experimental results for the Series 60 (C_B=0.6) ship model in steady drift manoeuvre at low and high Froude numbers. Numerical simulations were carried out by means of a surface fitting, finite volume based RANS PISO-type solver; turbulence effects were modelled by the Baldwin-Lomax model. Good agreement with experiments was observable for both integral and local quantities. The discrepancies between numerical and experimental data are due to different reasons: trim and sinkage were free in the experiments but fixed in the numerical simulations; wave breaking and air entrainment effects were not taken into account.

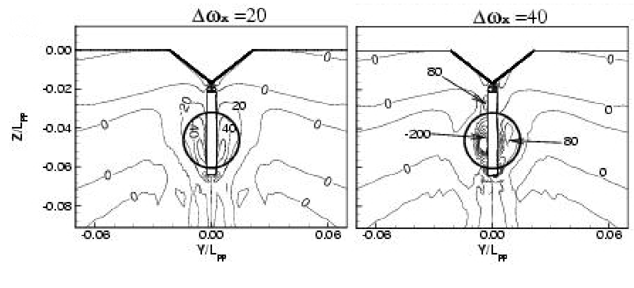


Figure 3.4- Axial vorticity at x/L=1.0 cross plane (β=4°, δ=0°). Left un-propelled, right propelled (Simonsen and Stern, 2003c).

In Simonsen and Stern (2003a, b and c) a detailed analysis of the propeller, hull and rudder interaction is made by using RANS based simulations. The computations were performed with the CFDSHIP-IOWA code, which is a finite difference, Chimera, multi block grid based RANS solver. The simulations were carried out for the ESSO Osaka model for the bare hull and full appended configurations; steady straight ahead, static rudder and pure drift manoeuvres were considered. Turbulence effects were taken into account by means of a k−ω model, free surface effects were neglected; propeller, operating at the propulsion-point, is taken into account by a body force field. The results show good agreement with experimental data for low drift and rudder angle, whereas discrepancies appear for the higher angles. Flow patterns were discussed and the behaviour of hydrodynamic forces was correlated to the main flow field features. In Fig. 3.4, the effect of the operating propeller on the cross flow at x/L=1.0, for the non zero drift angle case is shown.

Tanaka and Kimura (2003) have performed a similar analysis by using a cell-cantered finite volume based RANS solver NICE. The algebraic Baldwin-Lomax turbulence model was used, free surface effects were neglected and the effects of the propeller was taken into account by an infinitely bladed propeller model; a rather coarse mesh, with a total of around 320,000 volumes, was used. Agreement with experimental measurements is fairly good in terms of hydrodynamic forces.

In El Moctar (2001) RANS computations of the flow around a rudder in uniform flow and in the propeller slipstream, around a propeller in oblique flow, and around two different ship hulls, namely the C-Box cargo ship and the Esso Osaka tanker, in steady drift and steady turning motion are presented.

Cura Hochbaum and Vogt (2003) performed numerical simulations of a twin screw, twin rudder full appended RoRo ship in steady drift and steady turning manoeuvres. Governing equations are solved by means of a inhouse SIMPLE-type RANS solve; a two-equations k-ω turbulence model was used. Free surface was neglected, the effect of the propeller on the flow field was taken into account by a simple empirical body force model depending on drift angle and yaw rate. Side force, yaw and roll moment for different yaw/drift combinations were compared with measurements; the overall agreement was in general satisfactory for all the combinations but for the larger drift angle. Comparisons in terms of flow field quantities have shown qualitatively agreement with PIV measurements.

Numerical simulations for submarines in steady manoeuvres were performed by several authors. Kim et al. (2003) and Van et al. (2003) presented a comparison between numerical and experimental tests around two submarine geometries at various drift and pitch angles. Park et al. (2003b) performed an analysis of the flow around a submarine in steady drift manoeuvre with three different conning tower shapes. Park and Downie, 2003 and Park et al., 2003a presented numerical simulations of a British Bombardier moving with drift, yaw and combined drift/yaw motion. Sung et al. (2002) have used a finite volume based RANS solver for the prediction of the flow field around a submarine in steady turning motion. Sung et al. (2004) computed hydrodynamic forces and moments as well as stability indices for a Series 58 bare hull submarine by using the pseudo-compressibility finite volume based RANS solver IFLOW. Steady pitch and steady turning manoeuvres were considered.

<u>CFD WORKSHOP TOKYO 2005.</u> Test case 3 of the last CFD Workshop Tokyo 2005 (Hino, 2005) deals with the modified KRISO tanker (KVLCC2M) in steady drift condition in calm water; Froude number is set to zero, Reynolds number is equal to 3.945×10^6. Drift angles considered are β = 0, 3, 6, 9 and 12 degrees. The model is fixed. A large number of authors from different institutions have provided numerical results, namely: Broglia et al. from INSEAN, with their in house code XShip, Rhee and Skinner from FLUENT, Pattenden et al. from SOTTON with the commercial CFX5.7 software, Deng et al. from ECN/CNRS with their code ISIS, Gorsky et al. from NVWCCD with the UNCLE code, Kim et al. from KRISO with WAVIS code, Chou et al. from USDDC with their in house code UVW, Eça et al. from ITS/MARIN with PARNASSOS, Hirata and Kobayashi from NMRI with the code NEPTUNE and Hino and Sato from NMRI with the SURF code.

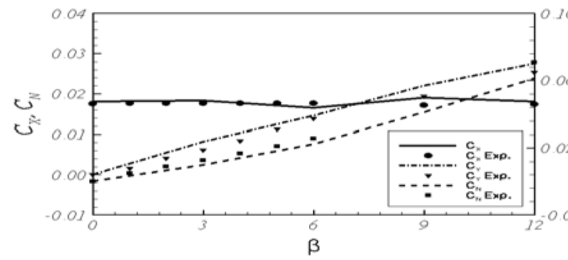


Figure 3.5- Longitudinal and lateral forces and yaw moment coefficients versus drift angle (courtesy of Broglia et al. in Hino, 2005).

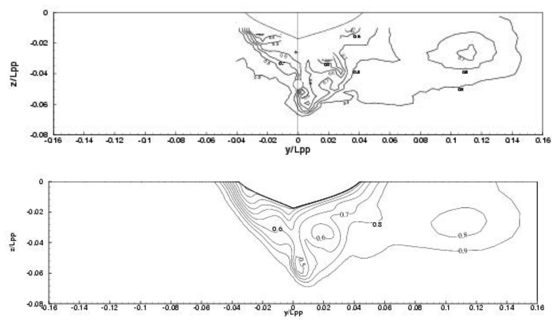


Figure 3.6- Axial velocity contours on the wake plane (β=12°); experimental measurements on the top, Deng et al. results on the bottom (courtesy of Kume et al. and Deng et al., in Hino, 2005).

Good agreement with experimental results is obtained in terms of integral quantities from almost all the authors. For instance, Fig. 3.5 shows the longitudinal, lateral force and yaw moment coefficients versus drift angle obtained by Broglia et al. (in Hino, 2005). Some discrepancies appear at larger drift angle, at least for those authors whose numerical grids where relatively coarse. In general, good agreement with experiments can be also noticed for pressure on the hull surface and pressure cuts on the hull surface, as well as for lateral force distribution along the hull. Some discrepancies appear when data on the wake plane (in the propeller region) are compared; in general the major differences are due to the turbulence model adopted and the grid employed. As an example of the better resolved wake field, in Fig. 3.6 results obtained by Deng et al. from ECN/CNRS with their code ISIS and the use of the *k-ω* EASM turbulence model is shown, in the same figure experimental data from NMRI are also reported for comparison.

As a final comment, the results for this test case were very encouraging, and the agreement with experimental data satisfactory for almost all the authors; this suggests that RANS based methods are mature for this kind of application and can be very useful.

3.4 Dynamic Manoeuvres

Burg and Marcum (2003) developed a non-linear scheme based on Farhat's algorithm to calculate the free surface elevation for unsteady RANS code. The results for a DTMB Model 5415 in small drift angles, constant turning radius and a prescribed manoeuvre although fairly good, were not grid independent and show more work needed on the turbulence models and grid deformation rate. Di Mascio and Broglia (2003) and Di Mascio et al. (2004) developed a unsteady RANS code for a non-inertial reference, attached to the body. Their first work focused in simple geometries and Series 60 hull in forced rolling motion without free surface. The second work present more complicated calculations, as a Series 60 hull in combination of sinusoidal sway and yaw motions, at Rn=4x10^6 and Fn=0.316. The calculations, for coarse, medium and fine meshes, showed that the drag coefficient converged, but the lateral force did not, probably due to the poor quality of the coarse mesh. Kim and Rhee (2002) studied the performance of different turbulent models for a manoeuvring submerged prolate spheroid using FLUENT. The calculations were carried out for Rn=4.2x10^6 and three different incident angles: 10, 20 and 30 degrees. The turbulent models analyzed were: Spalart-Allmaras, with a variation in the computation of the effective viscosity production; three different *k-ω* models (KO-1, KO-2 and Shear-Stress Transport), the first two based on the revised version by Wilcox (1998), and three different Second-moment closure Reynolds-Stress Transport Models (RSTM), each of them with small variations in the transport equation for *ε*. The *k-ω* model gave the better results, especially when adding the low-*Rn* effects (KO-2). The RSTM-1 model performed rather weakly. In addition, some eddy-viscosity based models which usually give good results in 2D performed poorly in this case. Also, it became clear that small modifications on the transport equation for ε (RSTM models) produce significant improvements.

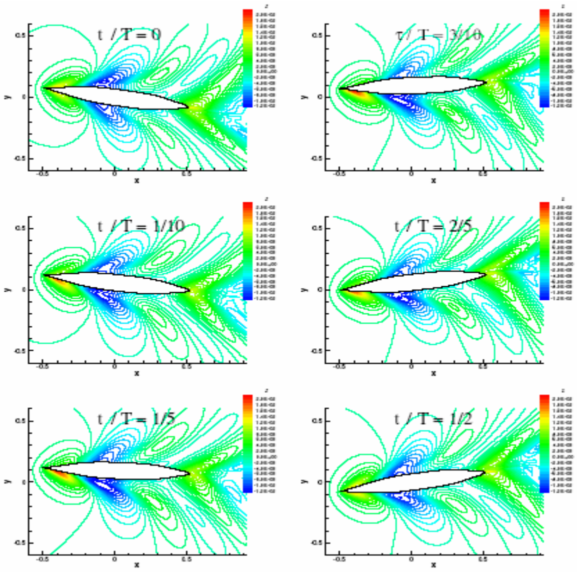


Figure 3.7- S60-PMM: Wave pattern during a half oscillation period (Di Mascio et al., 2004).

3.5 Trajectories

Jensen et al. (2004) used COMET to study a conventional passenger fast ferry advancing in head waves, with focus on the slamming effect in the bow door, and a container ship C-Box performing the turning-circle manoeuvre on its own spade rudder with a Fn=0.23.

In the case of the ferry, pitch and heave motions showed close agreement with experimental results, although the amplitude of the motions increased monotonically with finer grids. The vertical forces acting on the bow door showed also good agreement and even improved the results obtained from other state of the art methods. In the case of the container ship, the rudder was surrounded by sliding interfaces to allow its movement and the rudder was modelled by a body force field. The calculations produced reasonable results in terms of yaw rate, tactical diameter, pressure distributions and heel of the ship. Pankajakshan et al. (2002) utilized an unsteady RANS code coupled with a 6 DOF code to simulate induced submarine manoeuvres. Grids capable of deformations imposed by control surfaces deflections of up to 25 degrees were obtained using the method of Remotigue (2002). The sliding interface technique (Chen and Briley 2001) was used for the rotating propulsors. The simulations were carried out for the ONR Body 1 Radio Controlled Model at Rn=18.6x10^6 in order to use the experimental results from Faller et al. (2001). Even though agreement with experimental data was extremely good, as small errors tend to accumulate, drifting the simulation, particular attention had to be put to turbulence modelling, grid refinement and algorithmic issues.

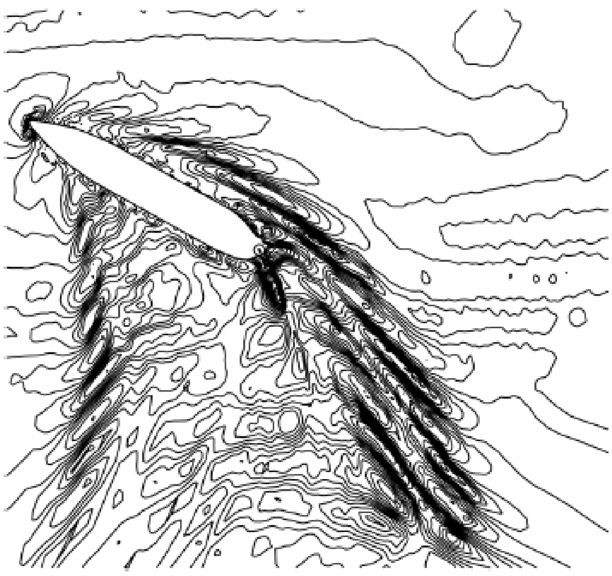


Figure 3.8- Free Surface Deformation during a Turning-Circle Manoeuvre of a Container Ship. Ψ=60˚ (Jensen et al., 2004).

3.6 High Speed and Restricted Water

High Speed. Orihara (2002) used unsteady RANS method to validate the capability of a CFD code WISDAM-VII for predicting the hydrodynamic characteristics of a high-speed ship advancing in calm water with its speed range 0.4<Fn<1.0. They validated the results such as surface pressure distributions, running attitudes, and hull resistance by comparing available experimental data. Azcueta (2003) studied planning craft running in extremely high speeds (i.e. up to Fn=4.0) using COMET. Pitch and heave motion prediction were improved when RANS with VOF method was used compared to extended Wagner's theory, however, there is still a difference when the wave length became comparatively large compared to the ship length.

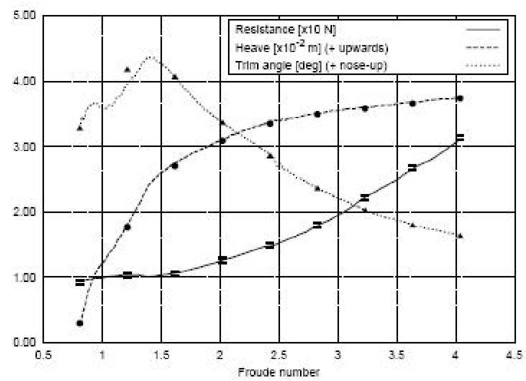


Figure 3.9- Numerical resistance test and computations at constant speeds for power boat sailing at extremely high speeds up to F_n=4. (Azcueta, 2003)

Sato et al. (2003) applied unsteady and steady RANS code SURF to predict the wave making resistance under the condition that the model had overturning wave at the bow. Comparison with experimental data generally showed good agreement even though without detail bow wave breaking due to the lack of grid resolution. Furthermore, unsteady simulation gave better convergence in terms of total resistance than steady simulation. Andrillon and Alessandrini (2004) tested the novel free-surface computation technique, 2D+T VOF fully coupled formulation to calculate breaking free-surface flow. They performed computations for both 2D and 3D case including Wigley hull computation for which *Fn* is up to 1.0.

Restricted Water. Lee et al. (2003) used both experimental and unsteady RANS approach to obtain quantitatively the hydrodynamic forces under the lateral motion, and a view to generalizing the obtained hydrodynamic forces that will be practical use. In shallow water physics for lateral motion, the water depth was an important factor to exercise influence on the inertial force and transitional lateral force acting on ship hull. They also resulted that using a concept of circulation was effective to express the lateral drag coefficient qualitatively. Hänel et al. (2003) proposed a combined RANS/EULER solver for flows past ship hulls in shallow water. The coupling of a field method in full dimensions in the near vicinity of a ship to a shallow water Boussinesq method simulating wave propagation farther apart showed promise, but stability of the coupling process was found to be critical in some cases and the position and size of the overlap zone of two solvers was chosen by empirical means and optimization will enhance the overall efficiency.

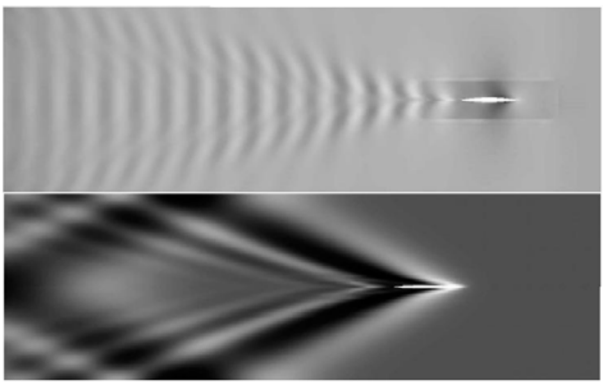


Figure 3.10- Surface elevation for coupled solution sub-critical speed F_{nh}=0.25 (top), and supercritical speed F_{nh} =1.50.(bottom) (Hanel et al, 2003) (F_{nh} : Froude number based on depth).

4. BENCHMARK DATA

4.1 Introduction

The manoeuvring community has demanded new benchmark cases for quite some time. Ideally, a list of benchmark ships for manoeuvring covering different "typical" ship types should exist, as it should for all the classical disciplines of naval architecture. However, it has not been possible to find truly relevant benchmark cases for manoeuvring, primarily because it is difficult to find relevant examples.

Manoeuvring predictions can be split into two base cases:

- Predictions in model scale
- Predictions in full-scale

Naturally, in order to predict full-scale manoeuvres, it is necessary first to address the problem of predicting manoeuvres in model scale. This is the case for free model tests, predictions based on captive model tests as well as for predictions based on CFD methods (potential theory and viscous). Validation should in principle exist in model scale before proceeding to full scale. However, the ship designers, shipyards, ship owners, all require reliable predictions of full-scale manoeuvres. This requirement naturally implies that scale effects need to be taken into account. The manoeuvring committee is of the opinion that manoeuvring prediction methods still lack validation even at model scale. This is well demonstrated by the Esso Osaka benchmark, perhaps best demonstrated by Barr (1993), where very different results are shown for hydrodynamic forces from various published captive model tests with the Esso Osaka. Even free model tests still require validation at model scale. This is the main reason for the mandate for this manoeuvring committee to work on uncertainty analysis of free model tests.

This section will suggest new benchmarks, which do not fulfil all requirements for a benchmark, primarily because no full-scale data exist, but they will be very relevant at model scale and they will enable various manoeuvring prediction methods to demonstrate their capabilities.

At first, a section will describe available literature covering possible benchmark ships for validation of manoeuvring prediction methods. The reason for performing this literature survey is to demonstrate that it is necessary to move forward with new benchmarks even though they are not perfect.

The problem of predicting ship manoeuvres by either numerical simulation or by free running model tests is difficult. One major reason is the lack of reliable and well-documented full-scale trial results of ship manoeuvres. There are many reasons for this; one is that requirements for ships manoeuvring capabilities traditionally have had second priority compared with resistance and propulsion requirements. Another reason is that manoeuvres are difficult to handle because of the unsteady nature of the problem. If we compare ship manoeuvring problems with resistance and propulsion it is obvious that resistance and propulsion data are easier to evaluate because they are somewhat steady situations, the speed of the ship, the propeller revolutions, the engine torque etc., can all be measured with reasonably accurate mean values.

Sea keeping is also a difficult problem and reliable full-scale measurements are seldom. Furthermore, large uncertainty is connected with the determination of the sea-spectrum that the ship was actually experiencing during measurements. Validation of seakeeping methods is more often done using model tests in towing tanks (head sea or following sea) or in large sea-keeping basins for oblique seas. The advantage for sea-keeping problems is that the mean values of heave, pitch and roll are around or close to zero.

Manoeuvring is more complex for many reasons. The first problem is that it is an unsteady problem and that the parameters of interest evolve in time. The number of parameters is also large, including at least position and speed vectors, rudder angle, propeller RPM. Another problem is that the system (the ship) is a highly complex non-linear system. A small difference in some conditions (initial or during a manoeuvre) can have large influence on the resulting manoeuvre. It is therefore obvious that a relevant benchmark test or trial requires the utmost accuracy with respect to execution and documentation.

4.2 Requirements for a Benchmark Case

The basic requirements for a benchmark case are:

- Full ship documentation (Lines, propeller, engine, superstructure).
- Accurate full-scale trial results, including full documentation of the ships loading condition.

Because scale effects in ship manoeuvring are poorly understood and for some ships, are assumed to be of importance, it is a requirement that trials are available in full-scale.

There are many problems related with fulfilling these requirements. The most important one being the fact that only in a very few cases it is possible to get the full ship documentation as this is usually considered proprietary data of the shipyard or the ship owner.

Another problem is of course the well-known problem to obtain accurate full-scale trial results. The problem to obtain calm deep-water condition with no sea currents is difficult to solve. The environment is difficult to control and many, if not most, sea trials have corrections for the environmental conditions during the trials.

The most famous and most used benchmark ship, the Esso Osaka, Crane (1979), is the best existing example of a benchmark ship. All relevant data for the ship exist and the trials were conducted with utmost carefulness. Still, with this classical benchmark ship there is a problem with the documented trials because it has not been possible to extract the actual drift angles during the trials. The relevance of this information was recently demonstrated by the 23rd ITTC specialist committee, which showed predictions of turning circles from three different mathematical models. The resulting manoeuvres from the three mathematical models were quite similar track wise, except for the fact that the three methods predict very different final drift angles during the turning circle, from 12 to 25 degrees. As the drift angle is the perhaps most important state variable in a manoeuvring model, it is not difficult to understand that this spread causes a lot confusion on how to interpret these results. It is not possible to determine which model actually reflects the true drift angle during the trial manoeuvres.

With today's accurate methods for determining positions and velocities, it is possible to obtain relatively accurate trial data. For possible future benchmark ships for manoeuvring this is a major advantage compared with the situation in the late 1970s during the Esso Osaka trials.

4.3 Literature Survey of Available Benchmark Ships

The most cited and used benchmark case for manoeuvring is the VLCC Esso Osaka, which was launched in 1973. Careful and controlled full-scale trial tests were conducted with this VLCC in both deep and shallow water and reported by Crane (1979). The full description of the hull, rudder and propeller is available. Numerous models of the Esso Osaka exist around the world and the Esso Osaka has been used as an example for research almost constantly since the trials in 1979. The latest examples are some new PMM tests that are reported as a written contribution to the present 24th ITTC report. These PMM results are performed at BSHC with free model tests and corresponding uncertainty analysis. The 23rd ITTC had a specialist committee on the Esso Osaka with the purpose to "close" the discussions on the Esso Osaka results and modelling. The complete list of references where the Esso Osaka has been used as a benchmark example is too long to cover here. However, a few "milestones" are mentioned. The JAMP (1985) group reported several test results with models of the Esso Osaka. Barr (1993) compared the results from various captive model tests and mathematical models with each other with discouraging correlation between the forces represented by the mathematical models. The 22nd ITTC manoeuvring committee recommended the Esso Osaka as a benchmark ship and the committee supported a conference, MAN01, arranged by BEC, with Esso Osaka as the main benchmark case. The 23rd ITTC specialist committee on the Esso Osaka gave the reasoning for selecting the Esso Osaka as the most relevant benchmark ship. A set of benchmark data for two types of models, the MMG model and the Whole Ship Model (Abkowitz type) are defined. There is no doubt that the Esso Osaka will stay as a benchmark ship for some time to come. The reasons are first of all that careful trials have been conducted in both deep water and two shallow water depths. Secondly, so much research has been going on using this ship as an example, that numerous comparisons with previous model test results or numerical calculations will take place also in the future.

The Mariner ship is another benchmark case for which carefully conducted full scale trials exists, Morse and Price (1961). The Mariner was intensively discussed as a benchmark in the reports of the 11th and 12th ITTC Manoeuvring Committees. The hull form is rather old (post-ww2 form) and very atypical for modern ships. However, full scale trial data exist, Morse and Price (1961), and several papers report captive model test results, including rotating arm results, Chislett and Strøm-

Tejsen (1965), Wagner-Smitt and Chislett (1974), Gertler (1966), Burcher (1966).

Clarke et al. (1972) published full-scale results for a series of manoeuvring trials with the 193.000 DWT tanker Esso Bernicia. The trials were conducted in both deep (T=77m) and shallow (T=26.5m) water in loaded and ballasted condition (four different conditions). Among other manoeuvres the Bech's spiral manoeuvre was conducted in all four conditions. In loaded condition both spirals showed a course-unstable ship and only with small dependence on water depth. In the ballast condition the spiral showed a marginally course stable ship in deep water and a course unstable ship in shallow water. The full-scale trial results are well documented but it has not been possible to find any references using these trials as benchmark.

A recent paper by Trägardh and Levine (2004) describes comprehensive full-scale trial results for a newly designed twin-screw tanker. The lines have been published but there is still some way to recommend these trials as benchmarks. A more comprehensive description of the trials is required and access to lines and rudder propeller data need to be granted. The manoeuvring committee finds this ship interesting as a benchmarks ship because full-scale trials exist in deep water. However, the MC decided not to recommend it for this committee, primarily because it is a "non-conventional" design but also in order to focus on a few benchmarks. It is, however, a candidate for the future.

4.4 The Search for a New Benchmark

The Esso Osaka was recommended as the benchmark ship by the 22nd ITTC manoeuvring committee and the Esso Osaka is still a relevant benchmark, especially because of the trials being performed at both deep and shallow water. The manoeuvring community will have to live with the spread in results from captive model tests and keep this spread in mind, also for future research with other benchmark ships.

There are, however, some arguments against using this ship as a benchmark, even though some of these arguments are put forward primarily by the CFD community:

- Old design
- One ship type
- No documentation of the drift angle in the full scale trials
- "Perhaps" the trials were contaminated with current
- For CFD purposes, lack of propulsion data.

It is relevant to define new benchmark cases in order to concentrate research on a limited number of well-documented ships. The manoeuvring committee has not been able to suggest a new benchmark ship that fulfils all requirements. The committee, however, feels that there is a desperate need for new well-documented benchmark cases. It is therefore required to compromise. The two most important requirements, that full-scale trials exist and that the ship is well documented, are contradictory (If trials exist the ship yard/owner is reluctant to publish the ship documentation).

The 24th Manoeuvring Committee has chosen to concentrate on manoeuvring prediction in model scale. A logical example for a manoeuvring benchmark in model scale would be the Japanese SR-221 tankers A, B and C models. Basically, these three models have the same fore body but different after body shapes, from the more "classic" U-formed shape represented by the model B to the more modern pram-formed after body represented by model A. The model C is has an "intermediate" after body shape, somewhere in-between the two other models. Several papers document that even though these models have almost identical main particulars they have very different manoeuvring behaviour. In Fujino (1996) it is shown that the modern pram formed A-type aft body gives a much more course unstable ship than the "classic" U-formed B-type aft body.

Actually, the B-model produces an almost course stable ship. Several papers with Japanese authors use the SR-221 models as a test example for showing the importance of the shape of the aft body on the manoeuvring characteristics for full-formed ships, a few examples are: Fujino (1996), Nonaka et al. (1996), Ishiguro et al. (1996), Kose et al. (1996), all from MARSIM 1996. The hull-forms for the SR-221 tankers are proprietary and they therefore cannot be used as new proposed benchmarks for ship manoeuvring.

The Korean towing tank KRISO has developed a similar "example" case. KRISO has kindly offered to provide full access to two versions of the same tanker. This includes full lines descriptions as well as rudder-propeller details. Furthermore, KRISO has performed captive model tests for both hulls and the data will be available at www.simman2008.dk as they become available. The 24th Manoeuvring Committee found these two versions of the same tanker very interesting because even though their main dimensions are almost the same, they have different manoeuvring characteristics due to slightly different stern shapes. The variation in stern shape is in the direction of V-shape towards U-shape, in line with the numerous papers from Japan using the SR-221 tests models, However, in this case, the variation in the after body shape is much less than in the Japanese SR-221 example. The main particulars of the two ships are shown in Table 4.1.

A body plan of the two after body shapes is shown in Fig. 4.1. From Fig. 4.1 it is seen that the difference between the after body shapes is quite small, in fact, it is so small, that one could expect that some prediction methods would not be able to predict large differences in the two models manoeuvring behaviour. The results of the captive model test are shown in a paper by Kim and Kim (2000). Just to show the difference in the predicted manoeuvring behaviour, Table 4.2 shows the predicted overshoot angles in the 10-10 and 20-20 zigzag manoeuvres for the two models:

Table 4.1- Main particulars of VLCC 1 and VLCC 2.

	VLCC1	VLCC2
Lpp (m)	320	320
Beam (m)	58.0	58.0
Draught (m)	20.8	20.8
Block coefficient	0.81	0.81

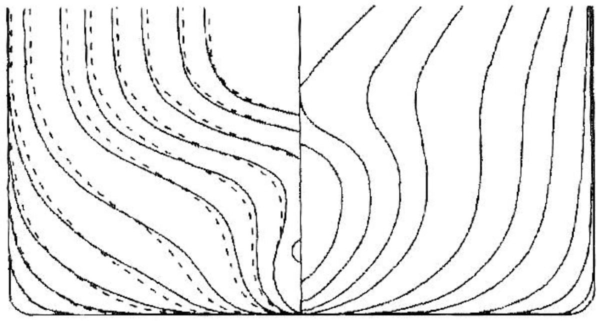


Figure 4.1- Body plan of VLCC1 and VLCC2. Solid line is VLCC1 and dashed line is VLCC2. VLCC1 is the V-shaped (pram form) and VLCC2 is less V-shaped (more towards U-shape).

Table 4.2- Predicted overshoot angles of VLCC1 and VLCC2

	VLCC1	VLCC2
1 overshoot, 10-10 zz	8.3	5.6
2 overshoot 10-10 zz	24.6	15.0
1 overshoot 20-20 zz	11.0	8.7

As it can be seen from Table 4.2, the predictions show larger overshoot angles for the VLCC1, which has the more V-shaped (pram formed) after body. This is in line with the expected result.

Now, the two examples of the VLCC represent modern hull forms of full-bodied ships. It is also well known that manoeuvring predictions of faster, slimmer lined ships like container ships and RO-RO vessels is important as well. Especially, for low block coefficient ships with low GM values, manoeuvring can be a problem. A benchmark ship of this type is therefore relevant as well. Again, the 24th Manoeuvring Committee could not find a relevant ship with available full-scale results. The 24th Manoeuvring Committee recommends the 24th ITTC to adopt the KRISO Container Ship (KCS) as a benchmark ship for manoeuvring. This ship model has been selected by the resistance and propulsion community as a benchmark ship, and the lines and rudder-propeller data are already available on the World Wide Web. The arguments for suggesting this ship are lees convincing than for the VLCC1 and VLCC2, since no manoeuvring data exist at the time of writing. However, the ship is a 3600 TEU PanMax container ship with the following main particulars:

- Lpp = 230 m
- Beam = 32.2 m
- Draught = 32.2 m
- Block coefficient = 0.65

A PanMax container vessel at high speed with low GM is very likely a challenge for any manoeuvring prediction method and the KCS is therefore a relevant benchmark model for manoeuvring. There is strong evidence that benchmark manoeuvring data in the form of both PMM and Free model tests will become available in due time before the planned workshop www.simman2008.dk for validation of manoeuvring prediction methods.

The final suggested benchmark model is the well-known combatant model used in numerous projects, the DDG 51 frigate. The argument for including this model as a benchmark for manoeuvring is that many researchers work with combatant type hull forms. This specific hull form is relatively modern and it has been used as a benchmark for resistance and propulsion for some time. The preliminary work on verification and validation of captive model tests initiated by the present committee includes model test results using this model as an example. The results of the model tests will be made available in connection with the planned workshop www.simman2008.dk.

4.5 Recommendations for New Benchmarks

A review of existing benchmark ships for manoeuvring has been presented. The Esso Osaka is an existing benchmark ship for manoeuvring, especially relevant because of the availability of reliable full-scale trial results in both deep and shallow water. The trial data are, however, not perfect, especially because of the lack of information about the drift angle during the trials. There is a need in the manoeuvring community for validation of manoeuvring prediction methods using newer, more modern hull forms, as well as other types of ships. Four new benchmark ships have been recommended by the 24th ITTC Manoeuvring Committee for adoption as benchmark ships for manoeuvring. The benchmarks are characterised by the fact that they exist in model scale but they will never be build in full scale. The advantages of selecting these models as benchmarks are that:

- They have modern hull forms
- Lines, propeller, rudder data available
- PMM and Free model test data are or will become available for all.
- They are already selected as benchmark ships for resistance and propulsion.
- At the time of writing new model test results will be procured within the next ITTC period.

The 24th Manoeuvring Committee therefore recommends the 24th ITTC to accept these four ship models as benchmark ships for manoeuvring, even though no full-scale trial data exist for these ship types.

5. HIGH SPEED AND UNCONVENTIONAL VESSELS

5.1 High Speed Vessels

According to IMO (2000) a high-speed craft is a craft capable of maximum speed equal or exceeding V(m/s)= 3.7 $\Delta^{0.1667}$, were Δ is the design displacement in m^2.

The IMO 2000 HSC Code applies to:

a) Passenger craft which do not proceed in course of their voyage more than four hours at operational speed from a place of refuge;
b) Cargo craft of 500 gross tonnage and upwards which do not proceed in course of their voyage more than eight hours at operational speed from a place of refuge when fully laden;

Table 5.1- NATO Naval Vessel Missions (Sheinberg et al. 2004).

Mission	Submission
TRANSIT AND PATROL	Point to point
	Search and Rescue
	Offshore patrol
	Military Surveillance
HARBOUR MANOEUVRING	Mooring
	Towing
	Anchoring
ANTI SUBMARINE WARFARE	Proactive
	Reactive
SURFACE WARFARE	Ship to Ship
	Strike – Ship to Shore
ANTI AIR WARFARE	Proactive
	Reactive
MINE WARFARE	Mine hunting
	Mine sweeping
	Mine Avoidance
VEHICLE INTERACTION	RAS Alongside
	RAS Astern
AMPHIBIOUS	Land Marines
	Special Operations
OTHER	Mine Laying

(RAS means Replenishment at Sea).

Naval surface ships may be considered a combination of high speed and unconventional vessels. Sheinberg et al. (2004) present manoeuvring data for a number of U.S. Coast Guard Cutter in view of mission oriented manoeuvring requirements developed for future vessels by the NATO Specialist Team. Table 5.1 gives identified NATO Naval Vessel Missions.

The NATO Specialist Team describes different manoeuvring abilities with ranking levels for the different missions and suggests manoeuvring criteria. It is shown that the five Coast Guard Cutters meet the IMO criteria with large margins but only in some cases the suggested NATO criteria. The authors make the following general conclusions.

The NATO NG/6 efforts on manoeuvrability have provided a basis for identifying unified criteria. They have also afforded a forum for exchange and collaboration:

- Criteria need to be further subdivided by vessel size.
- There is increased interest in defining manoeuvrability in a seaway.

5.2 Unconventional Vessels

With unconventional vessels are understood ships with waterjet and/or podded propulsion, sailing vessels, tug/barge systems, dynamically controlled vessels including semi-submersible platforms and under-water vehicles.

Several papers dealing with the manoeuvrability of podded ships have been published. Within the European research projects OPTIPOD and FASTPOD extensive work have been done covering most aspects of hydrodynamic design.

Woodward et al. (2003) presents some work done in the OPTIPOD project, where four different ship types were studied.

Table 5.2- OPTIPOD ship data (Woodward et al., 2003)

Ship type	Lpp [m]	Displ [m3]	No of pods	Design speed [kts]
Cargo	160	30,000	1	15
Supply	89	9,000	2	15
RoPax	194	22,300	2	29
Cruise	289	46,300	2	24

A simple assessment of the dynamic stability of these ships gives the following stability characteristics.

Table 5.3- OPTIPOD ship stability characteristics (Woodward et al., 2003).

	Stability Test	Spiral Loop	Conclusion
Cargo Ship	-2.8×10^{-5}	10 (deg)	Unstable
Supply Ship	-3.2×10^{-4}	10 (deg)	Unstable
RoPax Vessel	2.4×10^{-5}	+ ve gradient	Stable
Cruise Liner	5.4×10^{-5}	+ ve gradient	Stable

They also propose semi-empirical equations for estimation of the manoeuvring derivatives that show a very good correlation with results from captive model tests as demonstrated by the Initial Turning Ability.

Table 5.4- Initial turning ability of OPTIPOD ships (Woodward et al. 2003).

Ship Lengths	Cruise Liner	RoPax
Estimated	1.76	1.28
Model tests	1.80	1.20

The Nichols chart given in Fig. 5.1 shows the Gain-Phase relationship for the four designs derived from simulations.

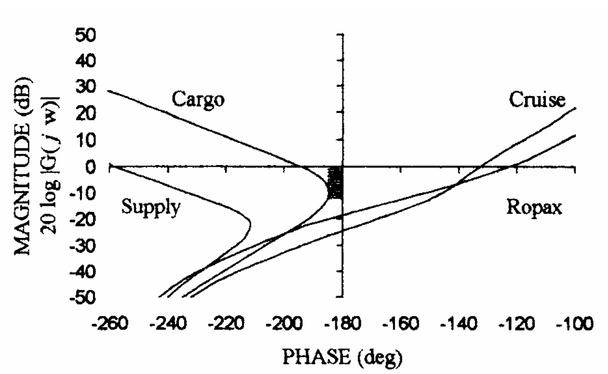


Figure 5.1- Nichols chart (Woodward et al. 2003).

Woodward et al. (2004) also presents a comparison of different stopping modes for pod driven ships based on simulations. The following results were obtained for the podded version of the OPTIPOD RoPax vessel.

Table 5.5- Simulated stopping characteristics for OPTIPOD RoPax vessel (Woodward et al. 2004).

Manoeuvre Performed	Stopping distance (Ship-lengths)	Stopping time (sec)	Max Stock Force (MN)	Max Stock Torque (MNm)
CSM	11.97	303	0.87	-
SSM1	6.66	201	2.81	2.72
SSM2	9.05	299	2.70	2.51
ISM	5.81	182	2.08	1.45

CSM means conventional stopping by reversing propeller rotation. SSM1 means turning the pods outwards 180° with constant shaft torque. SSM2 means same as SSM1 but reducing shaft torque by 40%. ISM means indirect stopping manoeuvre, i.e. turning the pods outward to 60° while ordering full astern and when ship speed has reduced by 80% ordering the helm back to zero.

Ayaz et al. (2004) give correlation data between results from free model tests and simulations. They also show that potential problems with pod-driven ships such as large heel angles in connection with manoeuvres and poor course stability could be overcome.

Depascale et al. (2005) presents some results coming out of the extensive work of the EU FASTPOD research project. Unfortunately no real manoeuvring results are given, but a photo of the 4-podded RoPax vessel that was used for free manoeuvring tests is given below.

Heinke (2004) gives results from extensive tests with another pod design at both pulling and pushing mode. Here the range of deflection angles is increased to ± 180 degrees and the maximum transverse force was reached at ± 60 degrees. The influence of cavitation was also studied showing small effects on forces and moments.

Figure 5.2- Model for manoeuvring and seakeeping tests of the FASTPOD RoPax vessel. (Photo from SSPA).

Grygorowics et al. (2004) presents results from comprehensive open water tests with two pod propulsor models. The tests were carried out in circulating water channel and the complete pod unit was mounted on a six-component tensiometer dynamometer. Forces and moments were measured for a range of advance numbers combined with a range of deflection angles from +30 to –30 degrees. Both a pushing pod and a pulling pod model were tested without propeller and with propeller at both positive and negative direction of rotation. The results are given as traditional non-dimensional coefficients.

Haragushi et al. (2003) compare a the course keeping and yaw-checking abilities of a ship equipped with a pod propulsion system and a conventional ship in view of the criteria of IMO Resolution MSC137(76). The background is a number of podded ships do not comply with IMO and still are reported to have good course keeping performance.

Junglewitz et al. (2004) presents a more theoretical study of podded propulsors. Although their concern primarily is the strength, they also compare the side force from a pod with that of a conventional spade rudder. The fractions of the side force from different components of a pod are discussed based on a RANS solution.

Kurimo et al. (2003) presents interesting comparisons of results from free model tests and sea trials for a twin-pod cruising vessel. Also simulations demonstrating the influence of scale effects on turning and yaw-checking abilities are presented. The table below shows results for turning circle tests at test speed 24 knots and 35° steering angle.

Table 5.6- Comparison of simulated turning performance (Kurimo et al. 2003).

	Full scale	Model scale
Advance / Lpp	2.21	2.12
Transfer / Lpp	0.97	0.91
Tactical Diameter / Lpp	2.26	2.11
Steady Diameter / Lpp	1.76	1.62
Drift angle	18°	19°

For the full-scale prediction the full scale resistance, wake and engine characteristics was applied, while for the model scale prediction the model resistance, wake and constant propeller rpm corresponding to model self-propulsion point, i.e. same condition as for the free model test, have been applied.

Stettler et al. (2004) presents some preliminary results of research being conducted at MIT on the dynamics of an azimuthing podded propulsor, with emphasis on the application to nonlinear vehicle manoeuvring dynamics. The results clearly illustrate unique characteristics of vectored-propulsion, including generation of a sizable normal force and increase in vectored thrust associated with propulsor azimuth. Force data is also presented in terms of surge and sway forces showing a nearly linear up to ±45° azimuth angle. Results cover azimuth angle of ±180°. Limited result of wake flow visualisation using Particle Image Velocimetry (PIV) is also presented.

Trägårdh et al. (2004) presents examples manoeuvring studies including free model tests, simulations and sea trials of a Double Acting Tanker in both ahead as in astern mode. Also examples of single-point- mooring simulations are given.

To overcome some of the problems such as excessive forces, cavitation and vibration of existing rudder and pod designs, particularly for fast vessels, studies of course keeping only using a flap on the trailing edge of the rudder or pod have been presented. In Japan a double-action flap rudder acting like a normal flap rudder at low speed but using the flap only at high speed has been developed by Nakashima Propeller Co. (2003). For podded ships flap steering was studied for instance within the European OPTIPOD research project and presented by Trägårdh (2002), where also some aspects of the manoeuvring and course-keeping performance in a seaway of a conventional and podded RoPax vessel is highlighted.

In light of the arising interest in modern sailing merchant and cruising vessels Masuyama et al. (2003) present an interesting paper revealing the sailing performance of the typical Japanese sailing trader, which has already disappeared into history.

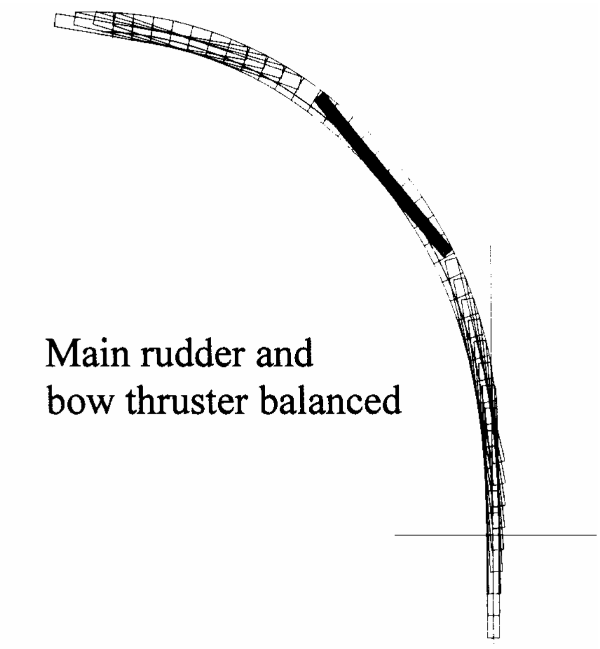


Figure 5.3- Manoeuvre using main rudder and bow thruster (taken from Gronarz, 2003).

Gronarz (2003) has made a study of traffic lane width in river bend with a barge train system. Simulations supported by model tests can be used to minimize the required lane width by combining the traditional turning by a main stern rudder and an optimised use of a bow thruster depending on river bend radius.

Another interesting paper on barge trains is presented by Kong et al. (2003). They have made a simulation study of zig-zag tests with a barge train based on full-scale tests. The classic ship manoeuvring equations were adopted and processed to linearised and non-dimensionalised form, and then a kind of Neural Network Recursive Model (NNRM) was derived and applied to full-scale barge train tests carried out in inland waterways of China.

The following figure shows the object barge train used in the full-scale tests conducted in Nantong, east China, near Shanghai. The particulars of this barge train are listed as follows:

- Ship Name: Yangtze 22018
- Length: 200m; Breadth: 65m: Mean Draft: 2.6m

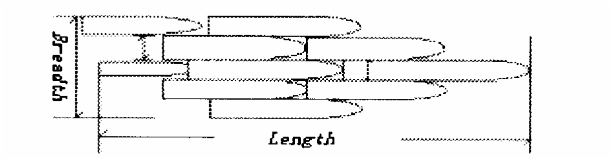


Figure 5.4- The combination of the barge train (taken from Kong et al. 2003).

The authors conclude that they expect that, after further process of environmental effects, external disturbance and consideration of arbitrary velocity change, the proposed NNRM may be applied to the following objects:

- Course-keeping or automatic pilot and even berthing of barge trains or ships in inland waterways.
- Online, real-time prediction and tracking of barge trains or ships during manoeuvring in restricted water.

Shigehiro et al. (2003) present a prediction method to investigate the manoeuvring performance if a Philippine outrigger fishing craft. Hydrodynamic derivatives were determined based on captive model tests. Simulated turning circle tests are compared with results from free model tests in calm water. Also the influence of wind on the turning circle and zig-zag tests was studied.

6. CONFINED WATERS

The introduction of the IMO criteria for manoeuvrability has led to larger focus on prediction of the manoeuvres related with these criteria. The criteria should be evaluated in calm seas, in loaded condition, in deep water and at a speed close to maximum.

It is, however, well known that the majority of critical ship manoeuvres are made in confined waters (shallow water, banks) and naturally the influence of other traffic and environmental forces also becomes important parameters. Normally, ship simulator models are used for analysis of confined water issues. The requirements on accuracy for such mathematical ship models are just as critical as for manoeuvring models suited for prediction of the IMO criteria. However, the complexity of the model become significantly larger due to the drastically increased number of influencing parameters involved. In the 23rd ITTC report by the Manoeuvring Committee, a thorough overview, analysis and description of various empirical methods for confined waters and ship-ship interaction was given. The overview covered empirical methods for:

- Shallow water corrections
- Muddy bottoms
- Horizontal restrictions
- Ship-ship interaction
- Ship squat

In 2001 an International workshop on channel design and vessel manoeuvrability was held in Norfolk Virginia. Gray et al. (2002) summarizes the results of the workshop:

- that there is a need for more full scale results of ship manoeuvres in restricted waters.
- that in addition to the IMO criteria in deep water, manoeuvring criteria need to include also slow speed manoeuvring in restricted waters.

There were, however, no proposals for new shallow water manoeuvring criteria, and no doubt, this is a difficult task. This section will cover new literature since the last ITTC on these topics. There is a clear trend in the work of the period that numerical methods (potential methods and RANS methods) are more frequently used to calculate the hydrodynamic influence of lateral and vertical, fixed or moving, restrictions.

A large section is dedicated to a literature survey on hydrodynamic forces in deep and shallow water for the Esso Osaka.

6.1 Shallow Water

Wang et al. (2000) developed a method for prediction of linear acceleration and velocity derivatives for ships advancing in shallow water and in a channel. The method is a boundary integral equation method and ship squat and interaction with the bottom and channel sides is taken into account. A comparison with the classic tests with the Mariner ship and the Tokyo Maru by Fujino (1968) shows good agreement for the acceleration derivatives and the linear velocity derivatives.

Varyani et al. (2002) investigated the turning characteristics of a trimaran in deep and shallow water based on a simulation method using empirical formulae for shallow water corrections of hydrodynamic derivatives.

A large series of PMM model tests with a container ship model and a model of the Esso Osaka in deep and shallow water was reported by Eloot and Vantorre (2003). In deep water PMM testing it is normally assumed that various added mass derivatives are determined by selecting certain standard test parameters. The paper shows that this is not the case for shallow water testing, on the contrary, various added mass derivatives are highly influenced by the PMM test parameters. A RANS method was used by Lee, Toda and Sadakane (2003) for prediction of the lateral force on a Wigley hull performing a berthing manoeuvre. Results are compared with model test results with good correlation, also in unsteady motions.

Kijima (2004) investigated manoeuvring characteristics in shallow water. To estimate hydrodynamic derivatives, the author proposed approximate formulae, and results showed that hydrodynamic forces and moment acting on a ship changed remarkably depending on water depth.

6.2 Bank Effects

The use of CFD to determine the interaction between a ship and other ships or banks has had some attention. Chen et al. (2002) used a RANS method to calculate ship-ship interaction effects in shallow water as well as in a channel. It was shown that the method was able to reproduce reasonably well some of the ship-ship interaction results reported by Dand (1981), both with respect to time histories and the magnitude of the interaction forces.

Miao et al. (2003) developed a numerical method using potential theory for prediction of bank forces of a ships advancing in a channel. The method gives reasonable results for $H/D > 1.5$ but needs improvements for smaller water depth to draught ratios.

For some time research in the United States has been going on to measure full scale manoeuvres using DGPS to obtain more accurate information on full scale ship motion behaviour in confined waters from full scale measurements of ship motions in the Houston Ship Channel. The long-term goal is to be able to develop more accurate simulation models. A paper by Dagget et al. (2003) focuses on the combined effect from shallow water and banks. The paper suggests the need for improved bank models and for reliable deep-water full-scale trial results to ensure that the basic simulation model actually reflects the real ship behaviour.

Several papers deal with model tests and development of empirical methods for ship-bank interaction, Vantorre et al. (2003) and Li et al. (2003). The work reported by Li et al. (2003) includes results for varying water depth, ship types (including a catamaran), sloping and vertical banks, flooded banks and the effect of propeller thrust. The paper also compares results with another empirical method by Ch'ng et al. (1993). Vantorre et al. (2003) also present an empirical method for ship-bank interaction based on model test results with two bulk carrier models and a container ship. The model includes the influence of water depth, bank distances and propeller loading. The limits for the proposed regression equations are stated.

6.3 Ship-Ship Interaction

A large simulation study in confined waters was conducted for the Norfolk harbour, as reported by Chen et al. (2002). The study included risk of grounding, manoeuvring space and required deepening of approach channels. Chen et al. (2003) present ship-ship interaction forces calculated from an unsteady chimera RANS code and comparisons are made with the "classical" Dand (1981) ship-ship interaction model test results. Very good correlation is found between the calculations and the measurements. The paper concludes that the free surface is important for the interaction between the two ships.

Yasukawa (2003) used a potential theory to calculate the ship-ship interaction forces between two ships in an overtaking situation. The motions were simulated using coupled equations of motion and the 3D panel method provided hydrodynamic interaction forces and added masses as function of the ships relative positions.

Liu and Zhang (2002) adopted a 3D Rankine source boundary element method based on the fully non-uniform rational B-spline (NURBS) to calculate the interaction forces between two ships. The geometry of the ship hull and unknowns to solve in the fluid domain expressed as the formal of NURBS. The numerical results and analysis provided for the hydrodynamic interaction forces and moments.

Varyani, McGregor, and Wold (2002) developed a potential theory method for calculation of ship-ship interaction forces. The method is validated against results of Dand (1981) and Yasukawa (1990). Results are presented for two, as well as three, ships meeting in channel.

The effect of ship-ship interaction between overtaking/overtaken ships was investigated by Lee and Kijima (2003) for four different ship types; a VLCC, a PCC, a container and a Cargo ship. It is concluded that the main concern regarding ship-ship interaction concentrates on the PCC at low speed in strong winds due to the large superstructure of the PCC.

The mooring forces on a container ship induced by a passing bulk carrier were reported by Varyani, Krishnankutty and Vantorre (2003). The mooring system was modelled as linear. The results were obtained using an empirical method for calculation of ship-ship interaction forces acting on a ship at zero speed. Related work was presented by Krishnankutty and Varyani (2003, 2004).

Vantorre, Verzhbitskaya and Laforce (2002) showed results from a large series of ship-ship interaction model experiments with four ship models in a shallow water towing tank. The model tests covered a large variety of parameters such as overtaking/overtaken, speeds, distances, water depths. An empirical method for calculating the extreme peaks in typical time traces of interaction forces is suggested. It is, however, suggested in the paper that it is an impossible task to develop a full empirical method that takes into account all the possible parameters influencing interaction forces between two ships passing each other.

Pinkster et al. (2004) performed both experimental and numerical studies on hydrodynamic forces and free surface effects around ships moored in harbours subjected to other ships passing nearby. The study was conducted for restricted water depth and a rigid still water level. For the experimental study, measurements were performed for hydrodynamic forces, velocity around the hull, and wave elevation. For the numerical study, two methods were tested; one based on the double-body potential flow model, and the other also based on the potential theory but taking into account free surface effects. The authors concluded that computations generally give a good prediction of the characteristics of the passing ship forces compared to measurement. Also, they concluded that free surface effects are negligible for the case of a vessel moored in open water, however, predictions of mooring forces for vessels moored to a quay in restricted water should not be based on hydrodynamic forces derived from open water data of passing ship forces.

6.4 Ship Squat

Gourlay and Tuck (2001) presented numerical results of the maximum sinkage and trim of a ship travelling in finite depth predicted from two slender body methods. The two methods compare well with experimental results except for supercritical depth Froude numbers. The authors suggest that the discrepancies are due to the tank walls, which inevitable are present in the model experiments.

Jiang and Henn (2003) also show calculations from a method using slender-body theory for the near field flow and an extended shallow water approximation using Boussinesq's equations for the far field flow. The method shows good results for sinkage and trim compared with model test results.

6.5 Currents

As usual, only few references treat the problem of ships advancing in strong currents. Paulauskas (2001) presents a practical method for calculating required channel width in a harbour entrance subjected to strong cross-currents. Also Niwa and Numano (2002) studied ship manoeuvring under influence of strong currents.

6.6 Esso Osaka Shallow Water Hydrodynamic Force Data Re-Visited

The present 24th ITTC Manoeuvring Committee report recommends identifying new benchmark ships for ship manoeuvring. This does, however, not disqualify the Esso Osaka as being a relevant benchmark case, primarily because of the available full-scale data but just as well because of the huge amount of model testing and subsequent analysis work that has been published throughout the years. Especially, when moving into suggesting new benchmark ships is it relevant to discuss and conclude on "lessons learned" from the Esso Osaka case.

It has been demonstrated by Barr (1993) that the hydrodynamic derivatives in deep water by no means seem to correlate well among the published results. However, the motivation for digging deeper into these rather old references was that they could serve as a validation case for CFD calculations of hydrodynamic forces in shallow water, taking into account the huge amount of test results available in the literature.

One could ask why there could be any hope that the results would correlate better for shallow water than in deep water. The idea was to utilise that the usual method for interpolating the hydrodynamic forces between different water depth's is to use the hydrodynamic forces for deep water as a base reference and use correction functions to modify the various hydrodynamic contributions as function of water depth. In this way, the base case (the deep water results) are in a way removed from the results which then only reflect the relative influence of the shallow water, i.e. the various references with results from both deep and shallow water would perhaps correlate better.

As already mentioned, Barr (1993) compares Esso Osaka results for various simulation models, including hydrodynamic derivatives in both deep and shallow water (h/T = 1.2). This reference focuses on simulation results but it also compares the hydrodynamic derivatives from various institutions. Barr (1993) references a series of reports or papers dealing with the Esso Osaka. The following references have been identified, which include model test results with the Esso Osaka in both deep and shallow water:

- JAMP (1985), tests performed at various Japanese facilities, 2.5m model; Tokyo Univ., 6m model; NMRI
- Miller (1980), tests performed at Hydronautics Inc., USA, 7.66m model.
- Dand and Hood (1983), tests performed with a 1.6m and a 3.5m model at NMI, Great Britain.
- Bogdanov et al. (1987), tests performed at the Bulgarian Hydrodynamics Centre (BSHC), 8.0m model
- Oltmann et al. (1986) and Gronarz (1988), tests performed at HSVA and in Duisburg (VBD), 5m model.
- Eda, H., Davidson Lab (reference not found), 1.6m model.
- Dahl, S. (1994), Master Thesis, Danish Maritime Institute, now FORCE Technology, 5m model (HSVA/VBD model).

It was possible, with a reasonable effort, to find all of the above references except the report from the Davidson Lab by Eda.

The comparison referenced in the following concentrates on actual forces and moments from the measurements and not on force and moment predictions represented by hydrodynamic derivatives. To overcome this loss of information, the original forces and moments have been manually digitised from the various figures in the original references.

Now, if the forces are to be compared with CFD calculations (or compared with each other) it is important to know what were the exact conditions for the tests, which provided the model test results. Captive model testing can be made in numerous ways. Some of the more important test parameters are:

- Scale of model
- Bare hull/appended hull testing
- If appended hull, what was the propeller propulsion point (ship/model/in-between)
- Approach speed and strategy for speed during test program
- Was the model free in heave and pitch?
- Origin, are the forces and moments measured around COG or amidships?
- What kind of turbulence stimulation was used?

The scale of the model, the approach speed and the method of turbulence stimulation (if applied) have influence on the boundary layer developed on the ship model during the tests. Especially at lower speeds, it is the experience of FORCE Technology that less consistent results sometimes occur. An approach speed of 7 knots for the Esso Osaka falls well within this “low speed” category in model scale.

It has not been possible to find all this information/documentation for each of the tests in the various reports.

A summary of the mentioned test parameters is given in Table 6.1. It is clear from Table 6.1 that some information on the various test parameters is un-available. The MC has gathered some information by simply asking some of the involved persons in the tests. The JAMP reference is a kind of summary document and some of the missing information is probably documented in the original reports from the various research institutes.

Table 6.1- Summary of different test parameters for shallow water tests with the Esso Osaka. A question mark indicates that either it was not documented or un-clear to author.

Source	Model length M	Bare hull appended hull	Self propulsion point Ship/ model/between	Approach speed Knots	Turbulence stimulation	Free/fixed in sinkage & trim	Origin
JAMP 6 m model	6	Appended	?	?	?	?	COG N: Amidships
JAMP 2.5 m model	2.5	Bare	N.A.	?	?	?	COG N: Amidships
Hydronautics	7.66	Appended	Ship	7.0	Yes, studs at the bow	?	COG
NMI 3.5m	3.5	Appended	Model	3.7-11.2	Yes, 2.5 mm studs at bow and along bilge keel	?	Amidships
NMI 1.6m	1.6	Appended	Model	3.7-11.2	?	?	Amidships
BSHC	8	Appended	Ship	7.2	?	?	COG?
VBD/HSVA	5	Appended	Ship	7	?	?	Amidships
Davidson	1.6	?	?	?	?	?	?
DMI	5	Appended	Ship	8	Yes, sand at bow.	Free	Amidships

The first observation is that model lengths between 1.6 and 8 meters have been tested. The second observation is that all tests, except the one performed in Japan (JAMP 2.5 m), have been made using appended models. For the appended models, large variation exists as to the selection of propulsion point, i.e. ship or model self-propulsion point. Especially, in very shallow water, it is expected that the suction created by the propeller has major influence on the overall measured side force and especially the yaw moment.

The approach speed is not the same for all tests and it is known that especially at low speeds this will add some uncertainty to the measured results. The reason probably being in stationary boundary layer phenomenon. In relation to this scale effect problem, only some references mention the type of turbulence stimulation, if applied.

A classic assumption in captive model testing is that the model should be free to heave and pitch during the tests. This is usually done to obtain the same pressure distribution around the hull as in the full-scale situation and thus to include the effect of sinkage and trim on the measured forces. However, this principle has not always been used and only in a few cases it was documented what was actually the case during the experiments.

And finally, before a comparison of the yaw moments, it is necessary to know the origin around which the forces or derivatives are defined. Typically, two origins are used, either the centre of gravity or the amidships point.

Now, having seen this lack of complete information in most of the reports/papers, it becomes clear that the work made by the ITTC to develop procedures for different model tests and numerical predictions is highly relevant. Especially, it should be noted that the newly developed procedure for Captive Model tests specifically states that all of the above parameters should be documented.

Apart from the above parameters, the model test program itself is of importance. The strategy for selecting the speed in the individual runs is important and should be documented as well. At some institutions, it is common practise to reduce the model speed for increasing drift angle and yaw rate, thus implicitly including the speed loss in a manoeuvre such as a turning circle or zig-zag manoeuvre.

Measured Forces and Moments. In order to remove uncertainties due to different fairing principles and formulations of mathematical models, it was decided to try to find the directly measured forces and moments from the various test reports/papers. It was hoped that this could remove some of the variation in the test results. In reality, this is the first most basic comparison that is relevant before moving into the fairing process and mathematical model structure discussions.

Unfortunately, it was not possible to extract such information from JAMP (1985) reference. It was possible to extract pure drift results for the rest of the available sources except the missing Davidson Lab. Report. However, only few references actually document the measured results from the dynamic pure yaw tests. It was only possible to find this information for the tests performed at BSHC, NMI 3.5 m model and the DMI data.

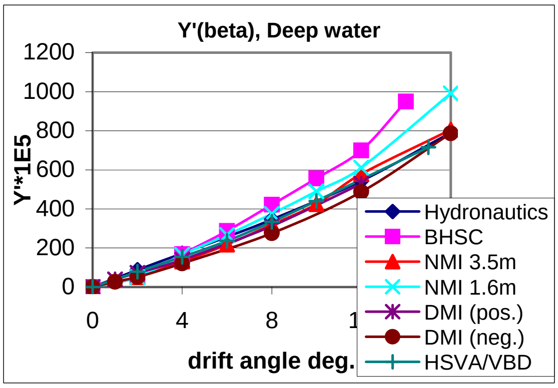


Figure 6.1- Non-dimensional side force Y' as function of drift angle beta in deep water.

The measured side force Y' is shown in the Figs. 6.1 to 6.3 for the three water depths (Deep, h/T=1.5, h/T=1.2). The following principles have been used in deriving the data points in the figures:

- Values measured at zero drift angle have been subtracted the total values.
- When possible, drift forces and moments have been used for positive drift angles.

- Results from BSHC have been “reversed” as they had focus on negative drift angles.
- The notation DMI (pos.) and DMI (neg.) corresponds to positive and negative drift angle.

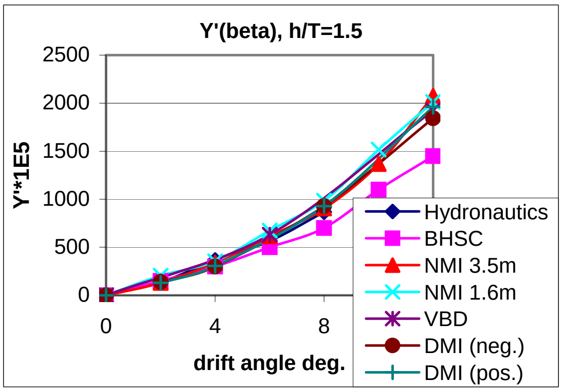


Figure- 6.2- Non-dimensional side force Y' as function of drift angle beta at h/T=1.5

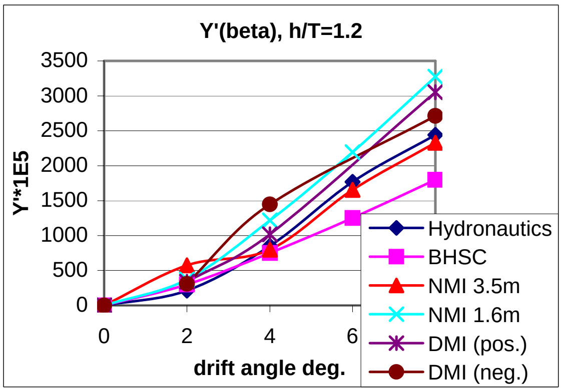


Figure 6.3- Non-dimensional side force Y' as function of drift angle beta at h/T=1.2

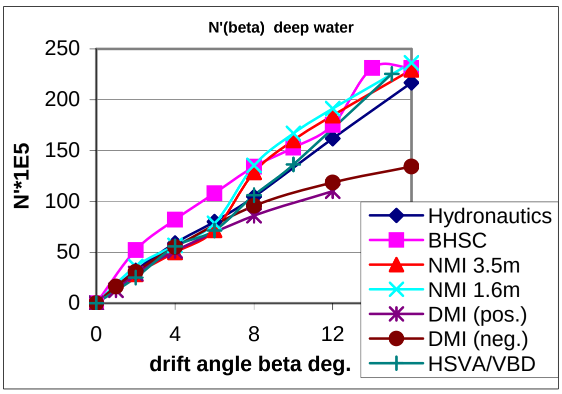


Figure 6.4- Non-dimensional yaw moment as function of drift angle in deep water.

Dr. Gronarz kindly provided a few data files he was able to find containing raw measurements from the VBD model tests.

The results for the yaw moments around midships are shown in the Figs. 6.4-6.6.

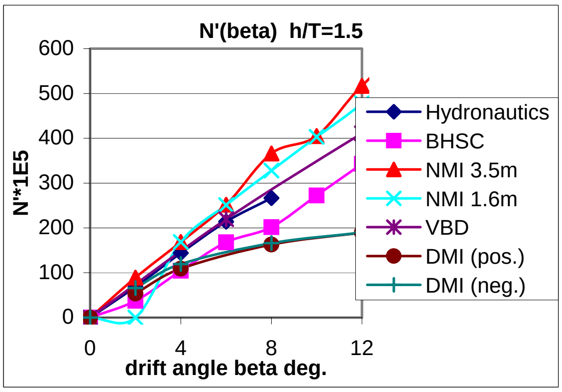


Figure 6.5- Non-dimensional yaw moment as function of drift angle in h/T=1.5.

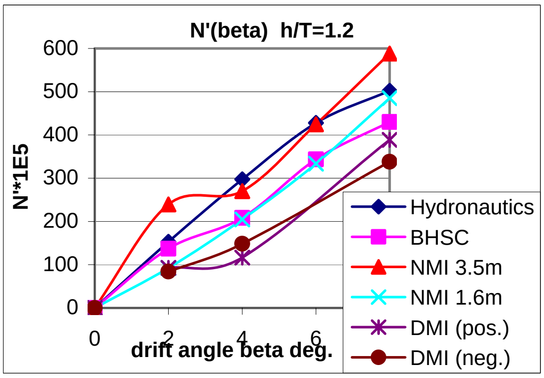


Figure 6.6- Non-dimensional yaw moment as function of drift angle in h/T=1.2.

Now, studying the six figures (Figs. 6.1-6.6) it is easily seen that the general spread in the results is large, perhaps as large as the spread observed in Barr (1993). The spread in the results can only be explained from the fact that the model tests are not identical; the models are not of the same scale and the test parameters are different. Furthermore, the tests are conducted at different facilities with different equipment and analysed by different methods. However, one comment is perhaps relevant for the results from the FORCE/DMI shallow water tank. It seems as the yaw moment becomes significantly smaller than the rest of the measurements for larger drift angles, whereas they are comparable for smaller drift angles. The reason for this might be explained by the dimensions of the relatively small shallow water basin compared to the model size. The DMI shallow water tank size is 25 X 8 meters.

The focus of this investigation was to determine the effect of shallow water on the hydrodynamic forces. The common way to express this effect, using hydrodynamic derivatives, is to model the effect of the shallow water by having an expression for the relative change of a hydrodynamic derivative as a function f_h of the water depth ratio T/h, f_h(T/h), where indices h indicates the hydrodynamic derivative. If such a function can be derived for all hydrodynamic derivatives it is possible to develop a mathematical model, which is able to perform manoeuvres in both deep and shallow water. A review of empirical methods for prediction of shallow water dependencies using this principle is given in the report of the manoeuvrability committee of the 23rd ITTC. The conclusion from the 23rd Manoeuvring Committee review of empirical shallow water correction methods is that the various methods differ considerably when going towards shallow water. Furthermore, the various methods have varying asymptotic expressions for moving into very shallow water, where the flow in principle will become a 2D flow since no cross-flow can occur. Some of the methods go towards infinity in this case.

In order to study the effect of shallow water on the hydrodynamic forces and moments, and to try to normalise the bias error for each of the model tests, it has been attempted to determine this function value for various fixed drift angles. The measured hydrodynamic force or moment f_h(h/T) at a given water depth h/T has been divided by the corresponding hydrodynamic force in deep water, f_h(deep) at a given drift angle. The results are shown in the Figs. 6.7 to 6.12 for the non-dimensional side force and yaw moment at specific drift angles.

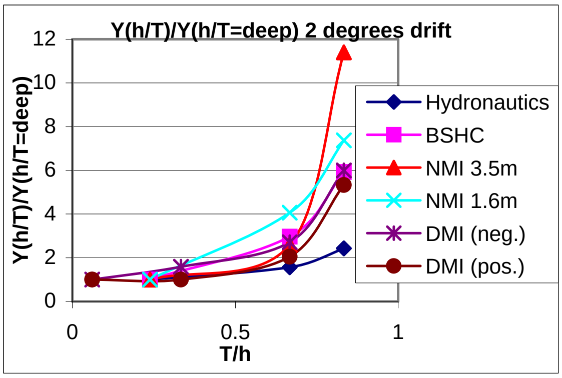


Figure 6.7- Non-dimensional side force as function of water depth, 2 degrees drift angle.

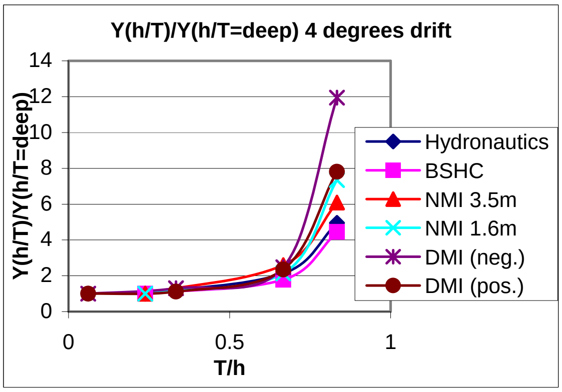


Figure 6.8- Non-dimensional side force as function of water depth, 4 degrees drift angle.

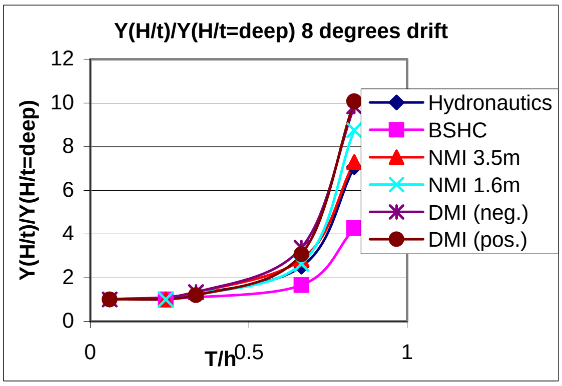


Figure 6.9- Non-dimensional side force as function of water depth, 8 degrees drift angle.

The following comment is given by not taking the results from the small DMI shallow water tank into consideration. For the side forces it is interesting to note that the dependency on water depth seems to depend on model size for h/T=1.2 In all cases, the function value increases with decreasing model length, except in one case. For the 2 degrees drift angle case, the 3.5 m model has a larger relative side force compared with the 1.6 m model. This can, however, be explained by studying the original results in Fig. 15 of the original reference, Dand and Hood (1983). Here it is seen that if the negative drift angles had been chosen instead of the positive drift angle, the smaller model would have had a much larger side force than the 3.5 m model for 2 degrees drift angle. This is in line with the conclusion given above.

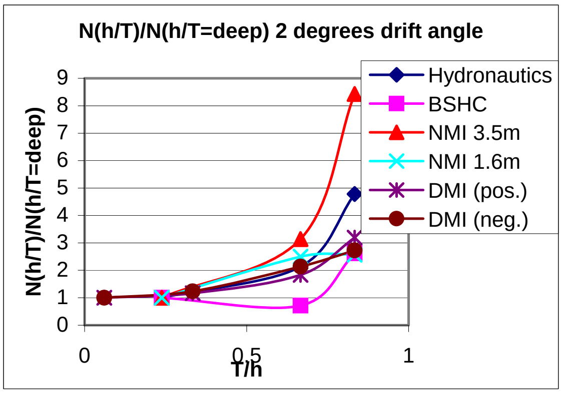


Figure 6.10- Non-dimensional yaw moment as function of water depth, 2 degrees drift angle

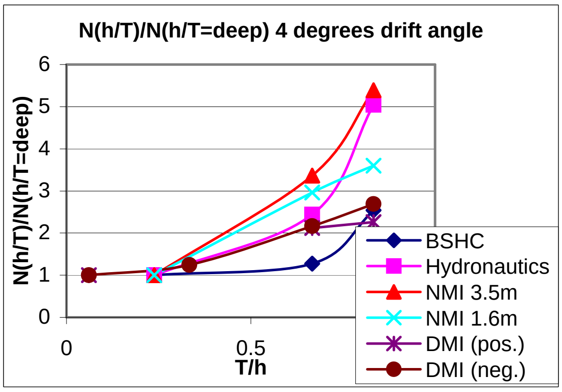


Figure 6.11- Non-dimensional yaw moment as function of water depth, 4 degrees drift angle.

The relatively larger influence of shallow water found for the FORCE/DMI data, for the two drift angles 4 and 8 degrees, might be explained by the relatively small shallow water basin, where the blockage of the model becomes larger for decreasing water depth and increasing drift angle.

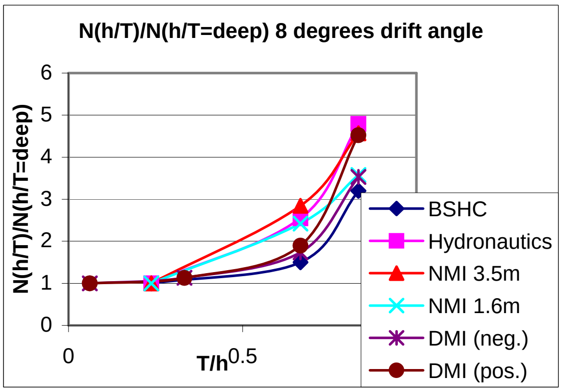


Figure 6.12- Non-dimensional yaw moment as function of water depth, 8 degrees drift angle.

The influence of water depth on the yaw moment is shown in Figs. 6.10 to 6.12.

The results from plotting the influence of water depth on the yaw moment show a pretty confused picture, the reason probably being that the yaw moment is sensitive to small changes in test parameters and scale, including sinkage and trim.

Only little data is available for raw measurements of the pure yaw motion. In fact, only the paper by Bogdanov et al. (1987) (BSHC) and the report by Dand and Hood (1983) (NMI 3.5 m, three depths, 1.6 m model, only deep water) contains Y′ and N′ as function of yaw rate. The master thesis by Dahl (1993) contains fairings of raw time series of one cycle but only for the two shallow water draughts.

The results for Y′ and N′ as function of yaw rate are given in the Figs. 6.13-6.18. Note that the results from NMI are results from a rotating arm test, not PMM results.

The yaw moments are shown in the Figs. 6.16 to 6.18.

No particular comments seem appropriate for the results for pure yaw. The results from the small DMI shallow water basin seem to produce larger yaw moment (larger damping) than the results from the two other institutions, except for large yaw rates in the intermediate depth (h/T=1.5). The result from BSHC correlates quite well with the rotating arm results in the two shallow water cases, whereas large differences occur for deep water.

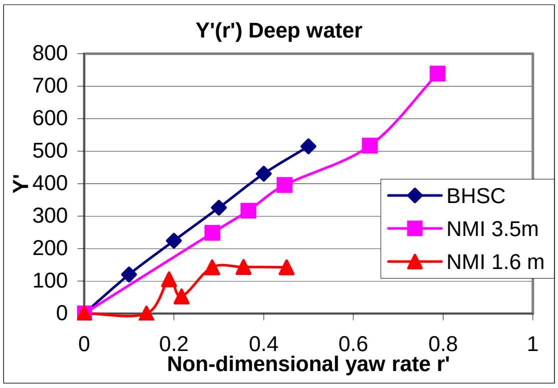


Figure 6.13- Non-dimensional side force as function of non-dim. yaw rate r′, deep water.

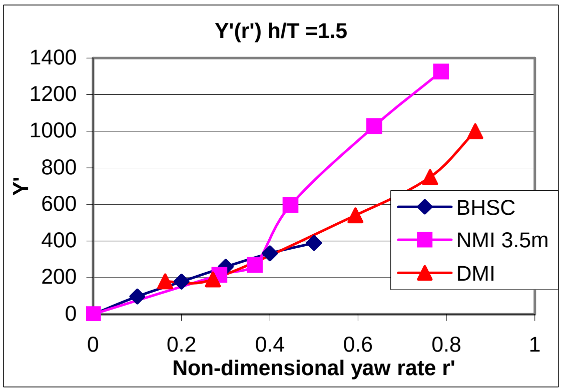


Figure 6.14- Non-dimensional side force Y′as function of non-dim. yaw rate r′, h/T=1.5.

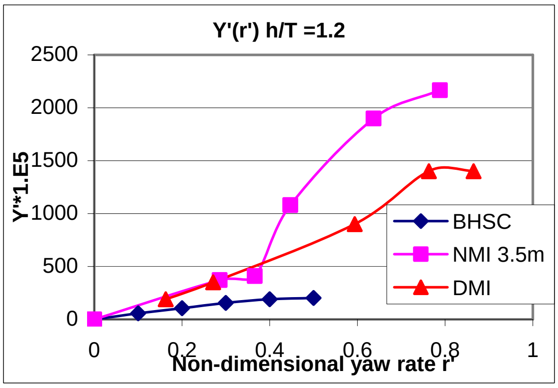


Figure 6.15- Non-dim. side force Y′as function of non-dim. yaw rate r′, h/T=1.2.

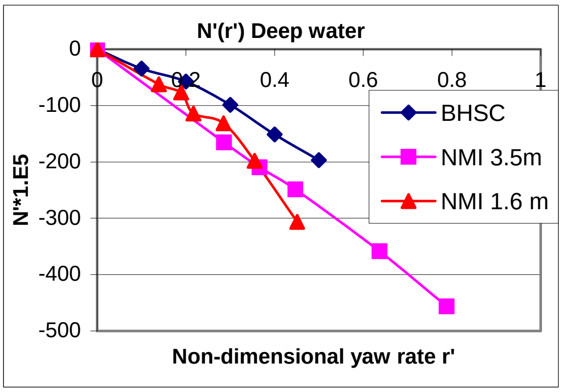


Figure 6.16- Non-dim. yaw moment N′as function of non-dim. yaw rate r′, deep water.

It is difficult to demonstrate the effect of shallow water in the same way as it was done for the pure drift case, as the three institutions have chosen different values of r′.

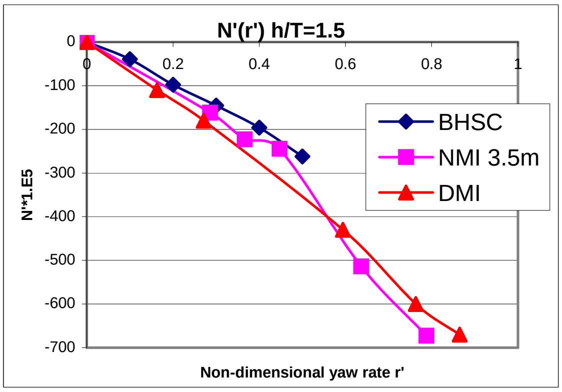


Figure 6.17- Non-dimensional yaw moment N′ as function of non-dim. yaw rate r′, h/T=1.5.

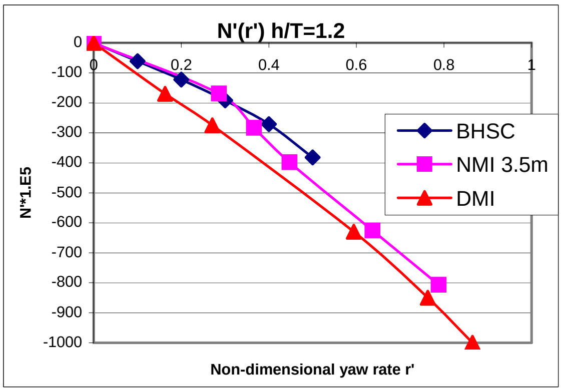


Figure 6.18- Non-dimensional yaw moment N′as function of non-dim. yaw rate r′, h/T=1.2.

Test Parameters. A comment on the choice of test parameters for the various Esso Osaka model tests is considered relevant. If the purpose of either PMM or free model tests is to predict turning circles and zig-zag manoeuvres in deep and shallow water, the changes of manoeuvring characteristics in shallow water should be included in the planning of the model test program. Studying the results of the normal ship manoeuvres from interviews with Captains and relevant literature (Gronarz, 1998, Crane, 1979, Bogdanov et al., 1987, Oltmann et al., 1986) makes it possible to suggest a "typical" table of maximum speed, drift angles, yaw rates and speed loss as function of water depth to draught ratio for large merchant ships. This Table 6.2 can serve as a guideline for selection of test parameters for manoeuvring tests in shallow water.

Table 6.2- Suggested relevant manoeuvring parameters for various depth to draught ratios for "normal" manoeuvring.

Large merchant ships (Turning circle)	Deep water h/T=4.0	Intermediate depth h/T=1.6	Shallow water h/T=1.2
Approach speed knots	Service	12	7
Max drift angle, Deg.	18	9	4.0
Max yaw rate, Deg/s	-0.95	-0.7	-0.4
Max speed loss	$0.35*U_0$	$0.5*U_0$	$0.7*U_0$

Conclusions, Esso Osaka Shallow Water Benchmark Review. A short review of benchmark tests with the Esso Osaka in shallow water has been made.

Hydrodynamic forces from the different sources have been compared in figures and tables for different water depths.

The results do not correlate well; in fact, the correlation seems to be just as bad as demonstrated by Barr (1993). The results do, however, indicate evidence of scale effects for the hydrodynamic side force as function of drift angle for h/T=1.2 where the non-dimensional hydrodynamic side force seems to increase for decreasing model size. This is an interesting conclusion, considering that predictions are supposed to be valid for full-scale.

It has been tried to define the basic test conditions for the different tests. However, not all, if any, of the reports document all important test parameters. For future purposes, both for commercial testing as well as for research purposes, it is recommended to study the two procedures from the ITTC: "Captive Model Test Procedure" and "Validation Procedure for manoeuvring simulation models" as inspiration for new reports. Due to the spread in model size, approach speed, lack of information on test parameters etc. it is perhaps not so surprising to find this large spread in the results.

And finally, this study shows that new benchmark tests are needed due to the spread in the results, combined with the poor knowledge of the actual test parameters during some of the tests. Such new benchmark tests must be properly documented and carefully executed as a first step towards the situation where methods for prediction of ship manoeuvres can be considered as properly validated.

7. MANOEUVRING STANDARDS AND SAFETY

Recognizing that the manoeuvrability of a ship is an important factor for the safety of navigation, International Maritime Organization (IMO) started its action on ship manoeuvrability in 1968 with the adoption of resolution A.168 (IV) on Recommendation on Data concerning Manoeuvring Capabilities and Stopping Distances of Ships. And then IMO adopted the Interim Standards for Ship manoeuvrability (A.751 (18)). IMO also recommended Governments to collect data obtained by the application of the standards, and report them to the Organization (IMO, 1993). Since 1993, ship designs focused on complying these standards, and the Netherlands (Quadvlieg, 2003), Japan (Haraguichi, 2000), and Korea

(Rhee, 2003) gathered the full-scale results of about 100ships, 287ships and 112ships, respectively. Based on the collected data, active discussions on the revision of the interim standards had been made (IMO, 1999, 2000, 2001), and the Standards for Ship Manoeuvrability (Resolution MSC.137 (76)) was finally adopted at the 76th meeting of IMO Maritime Safety Committee (IMO, 2002). IMO Standards for Ship Manoeuvrability provide for a minimum performance standard which a ship would have to achieve to ensure acceptable manoeuvring properties. During the 34th (1991), 35th (1992), and 36th (1993) Sessions of the Sub-Committee, the manoeuvring performance characteristics were established: turning ability, initial turning ability, yaw checking ability, course-keeping ability, and stopping ability. While the IMO Ship Manoeuvring Standards are primarily for use by ship designers, it is necessary that crew-members understand their implications in the proper discharge of their duties aboard a ship. Doerffer (1990) pointed out that the seafarer is interested in the manoeuvring characteristics of the ship in service, in real sea conditions. He is not interested in theoretical formulae worked out by scientists; he is not interested in parameters adopted by the designer at the designing stage. But he is extremely interested in actual manoeuvring parameters, which the ship is showing in actual service. For this purpose, works to validate that the IMO manoeuvring standards and the criteria are good measures for safe navigation at sea are carried out.

7.1 Turning Abilities

Turning ability of a ship is judged by the turning circle manoeuvre. Turning circle manoeuvre is performed to both starboard and port with 35° rudder angle or the maximum rudder angle permissible at the test speed and averaged values of both port and starboard data are used to represent the turning characteristics. Solid lines on Fig. 7.1 are the new IMO resolution MSC 137 (76). In case of advance and tactical diameter, all the ships satisfy the new resolution with sufficient margin.

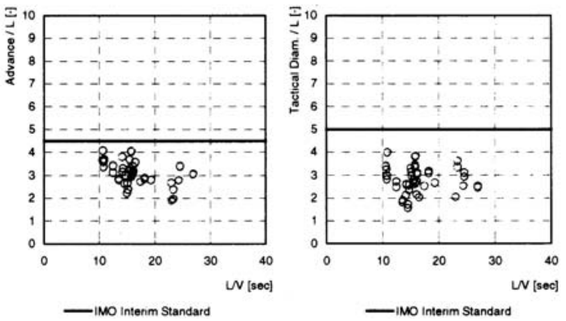
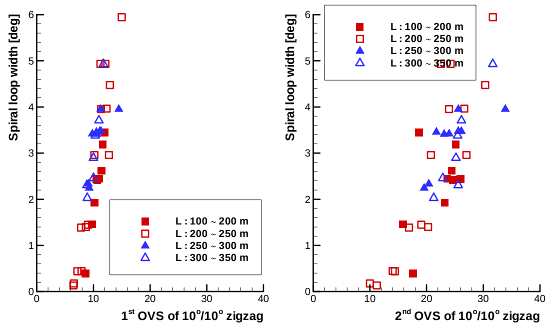


Figure 7.1- Turning ability indices from sea-trial. (Quadvlieg et al. 2003).

7.2 Yaw Checking and Course-Keeping Abilities

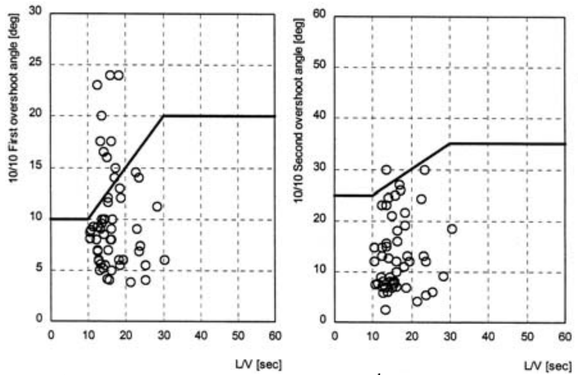
Yaw checking and course-keeping abilities are judged by zig-zag manoeuvres. Yoshimura et al. (2000) review these abilities. The criteria are evaluated based on trial data as well as theoretical and simulator studies, concluded that the 1st overshoot of 10deg. Zigzag test is one of the best index of course-keeping and yaw checking abilities. Rhee et al. (2001, 2003) investigated the correlation between spiral loop width and overshoot angle of zigzag test by simulation and Fig. 7.2 shows the relationship between the overshoot angles and the spiral loop width acquired from numerical simulation. Correlation coefficients between them are summarized in the Table 7.1. The strong correlations between the overshoot angles and the spiral loop width are shown and these reconfirm the fact that the overshoot angles are deeply correlated with the directional stability of a ship and they can be used as a good index for yaw-checking ability. Figure 7.3 shows yaw-checking and course-keeping ability indices measured from manoeuvring sea-trials of around 70 ships sailing in and around Dutch water, and Table 7.2 presents a statistical representation of the principal dimensions of the ships. A few ships having the values of L/U in between 10 seconds and 20 seconds cannot satisfy the resolution, and the vessels reported by the Korean and Japanese administrations that do not fulfil have an L/U value of around 30 seconds.



(a) Spiral loop width vs. 1st OVS of 10/10 zigzag

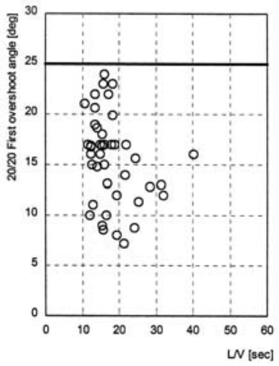
(b) Spiral loop width vs. 2nd OVS of 10/10 zigzag

Figure 7.2- Relationship between the overshoot angles and the characteristics of spiral loop from simulation (Rhee et al. 2003).



(a) 1st overshoot angles of 10°/10° zigzag test

(b) 2nd overshoot angles of 10°/10° zigzag test



(c) 1st overshoot angles of 20°/20° zigzag test

Figure 7.3- Yaw-checking and course-keeping abilities indices from sea-trial. (IMO DE document by the Netherlands 2000).

Table 7.1- Correlation coefficients between the overshoot angles and spiral loop and height (Rhee et al. 2003).

	Overshoot angles from simulation		
Spiral loop	10°/10° Z 1st OVS.	10°/10° Z 2nd OVS.	20°/20° Z 1st OVS.
Width	0.869	0.819	0.714
Height	0.571	0.411	0.735

Table 7.2 Statistics on principal dimensions (IMO DE document by the Netherlands 2000).

	UNIT	MEAN	STD	MAX	MIN
L_{PP}	m	155.4	61.5	325.0	68.5
B_M	m	24.74	9.0	53.0	10.2
T_M	m	6.8	3.3	21.8	2.7
C_b		0.73	0.10	0.87	0.47

Evaluation of Navigational Difficulty by Auto-Tracking Simulation. How can the safety of navigation of a ship be evaluated? Although, there can be many answers to this question, from the viewpoint of manoeuvrability of a ship, it is clear that a ship which is easier to navigate is safer than others. So, the question may be modified as "How can we evaluate the difficulty of navigation?" The difficulty of navigation has been usually evaluated by two ways. One is the evaluation by the pilots (Yoshimura et al. 1990) (Sohn et al. 2002), the other is the evaluation by the physical indices such as the rudder index and swept-path index. In many cases, both indices are used simultaneously to complement each other, and it has been found that a ship, which is well reputed by the pilots, gives less usage of rudder and smaller swept path for a given mission (Gong et al. 1997). Therefore, the rudder index and the swept-path index calculated from the auto-tracking simulation may be a good measure for the evaluation of the navigational difficulty.

Rudder Index and Swept Path Index. For the evaluation of navigational difficulty, Rhee et al. (2001) used two physical indices; rudder index and swept path index. Rudder index, RI is defined as:

$$RI = \frac{1}{T}\cdot\int_0^T |\,\delta(t)\,|\,dt \qquad (7.1)$$

where,

T : Overall controller operating time [sec]

$\delta(t)$: Rudder angle during T [rad]

This index reveals the averaged usage of rudder during the given mission, and it is well known that a ship devaluated by the pilots usually needs more operation of rudder to achieve the same mission. Second index is the swept-path index, SI, which is defined as:

$$SI = \frac{1}{T} \cdot \int_0^T | \beta(t) | \, dt \tag{7.2}$$

where,
$\beta(t)$: Drift angle [rad]

This index gives the averaged drift angle during the given mission. It has been reported that pilots feels difficulty when the drift angle is larger than 10 degree (Gong 1997).

<u>Evaluated Rudder Index and Swept-Path Index.</u> Auto-tracking simulations in two artificial seaways are executed for the rating of navigational difficulties. Fig. 7.4 shows the two artificial seaways; 10° bent seaway and 30° bent zigzag seaway. Fig. 7.5 and Fig. 7.6 respectively show the scatter plot of the measured overshoot angles versus calculated rudder indices and calculated swept-path indices for forty ships in two artificial seaways.

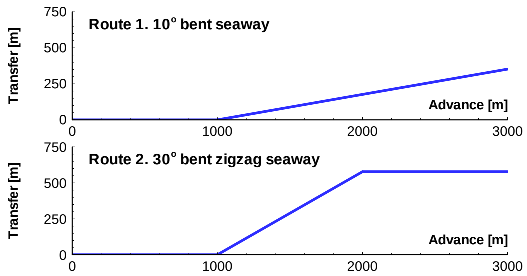
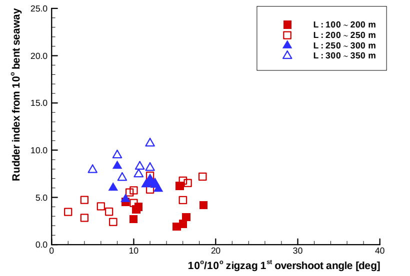
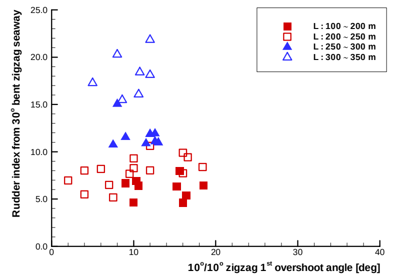


Figure 7.4- Two artificial seaways for auto-tracking simulation (Rhee et al. 2003).

Two clear tendencies can be seen from these figures that as the given mission become complex, both RI and SI are increased, and, as a ship becomes large, both indices are increased. Furthermore, it can be seen that the difference in L/U does not play a decisive roll in the determination of RI and SI. Many ships in group 2 (200m<L< 250m) and group 3 (250m<L<300m) are overlapped between 30 seconds and 40 seconds of L/U, but ships in group 3 clearly have the larger SI and RI value.

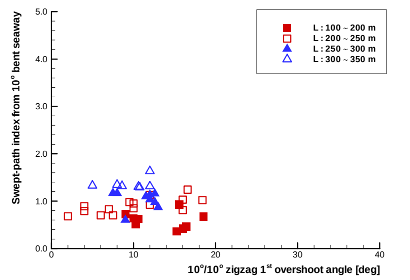


(a) 10°/10° zigzag 1st OVS vs. RI from 10° bent seaway

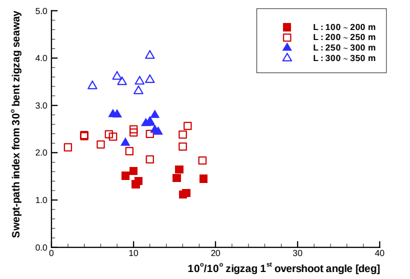


(b)10°/10° zigzag 1st OVS vs. RI from 30° bent zigzag seaway

Figure 7.5- Scatter plot between the measured overshoot angles and the calculated rudder indices. (Rhee et al. 2003).



(a) 10°/10° zigzag 1st OVS vs. SI from 10° bent seaway



(b) 10°/10° zigzag 1st OVS vs. SI from 30° bent zigzag seaway

Figure 7.6- Scatter plot between the measured overshoot angles and the calculated swept-path indices (Rhee et al. 2003).

Furthermore, a ship within group 3 has exceptionally large L/U value, but the calculated indices are almost similar or even smaller than other ships within the same group. This is somewhat contradictory to the IMO criteria for overshoot angles. According the IMO resolution, a ship with the smaller L/U should have the smaller overshoot angles, and this means that the smaller ships should satisfy the severer standard. In some sense this resolution may have a meaning that the smaller ships usually navigate the coastal area and the possibility of marine casualty in this area is higher than others due to heavy traffic. However, considering the navigational difficulty, current criteria for the small ships seems to be too severe.

7.3 Stopping Ability

The IMO standard requires evaluating the ship's stopping ability by full astern stopping test. Full astern stopping is decelerating a ship from full-ahead-sea speed until the ship comes to rest. Although, in real situation, "emergency full astern" is almost never ordered from the full-ahead-sea speed, it is a customary machinery acceptance trials and the results have been useful as a relative measure of stopping ability.

The sea trial data of the stopping test obtained by Denmark, Japan, and Korea are shown in Fig. 7.7. A number of ships do not comply with the criterion by which the track reach in full astern stopping test should not exceed 15 ship lengths as shown in the figure. This result shows that the criterion is impracticable for large ships without any problems of manoeuvring performance. So it is stated in the IMO Standards for ship manoeuvrability that the track reach in the full astern stopping test may be modified from 15 ship lengths to 20 at the discretion of the Administration, where the ship size and form make the criterion impracticable.

For the prediction of stopping distance for those ships, the Administration is recommended to use an expression in the Appendix 3 of MSC/Circ 1053. However, this expression does not seem to be recommendable, because it clearly underestimates the stopping distance of recently built ships. Fig. 7.8 shows the application results of the expression for the seventy-four ships. This underestimation is mainly caused by the assumption that the propeller is reversed as rapidly as possible after the astern order is given. The Appendix 3 on Stopping Ability of Very Large Ships (MSC/Circ.1053) suggests only 60 seconds for the time whilst the engine is reversed and full astern thrust is developed (IMO, 1993).

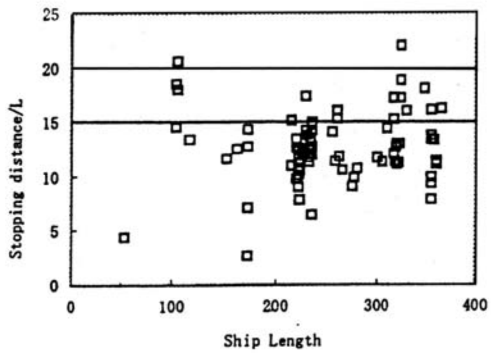


Figure 7.7- Track reach of ships at full load condition (IMO DE 44/4 2000).

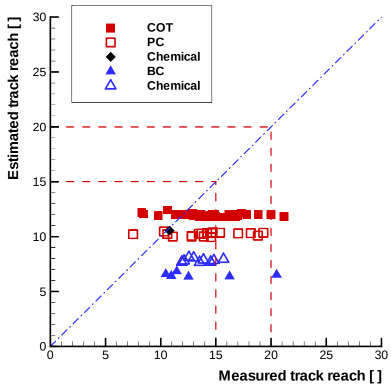
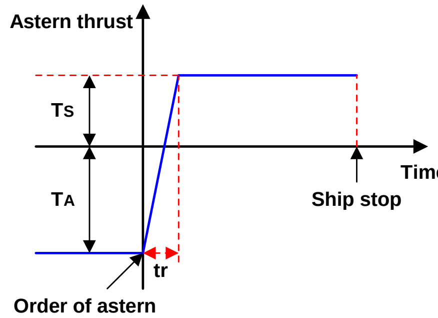


Figure 7.8- Prediction of stopping distance by the expression in the MSC/Circ.1053 (IMO DE 47 2004).

This criterion has been already applied to the ships constructed on or after 1st January 2004, and naval architects are required to predict the stopping ability of a ship from the initial design stage. There have been many studies on this problem; Chase and Ruiz (1951), Hooft (1970), Clarke and Wellman (1971), Yoshimura and Nomoto (1978), Fujino and Kirita (1978, 1979), Yoshimura (1994), Rhee et.al. (2001), Haraguchi (2002), Kim (2002), and Takagi (2003). Especially, Fujino and Kirita made it possible to numerically simulate the three dimensional motions (surge, sway and yaw) of the stopping manoeuvre. This simulation, however, requires reasonable estimates on the asymmetric hydrodynamic forces acting on the stern due to the reverse rotation of propeller. Such estimates can be obtained through the laboriously many runs of captive model test, but this is not the usual situations at design stage. Hence, the simple prediction method, which considers only the surge motion, still retains its power in the prediction of stopping ability.

The first simple prediction method was introduced by Chase and Ruiz (1951). They modelled the stopping manoeuvre as the first order surge differential equation, and derived integral form of solutions for stopping distance and time. Closed form of analytic solution was derived by Clarke and Wellman (1971). They assumed that the thrust changes linearly from the ahead value to the astern value during the short time taken to reverse the shaft and then remains constant throughout the whole stopping manoeuvre (Fig. 7.9). This assumption was quite valid one for those days' steam turbine ships, and IMO adopted their solution for the prediction of stopping distance at design stage (Appendix 3 of explanatory note (MSC/Circ.1053)). It still gives reasonable estimates for the ships equipped with steam turbine as their prime mover. However it's time as pass, most of the nowadays' ships adopt the more economical diesel or heavy fuel engines. This makes many researchers raise a question on the validity of the IMO's prediction method.

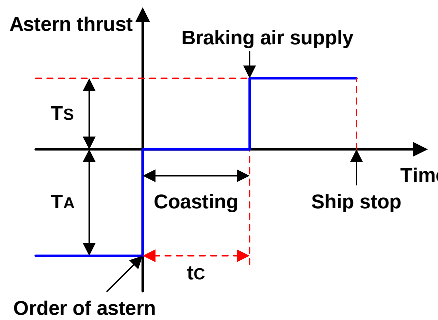
Japan (IMO DE 44/4, 2000) considered that the impracticability was caused by the constant value of the criterion on stopping for L. The stopping distance of a ship is mostly estimated by the displacement, the horsepower and the initial velocity, and thus a new criterion was proposed considering above physical parameters and the manoeuvring performance database of ships that have no problem.



$$T \sim \begin{cases} -T_A + (T_A + T_S)\cdot(t/t_r), & (0 < t < t_r) \\ T_S, & (t_r < t) \end{cases}$$

Figure 7.9- Conventional astern thrust model.

Rhee et al. (2003) proposed a method for the prediction of stopping distance and time at design stage. Coasting with propeller wind milling, which continues about 250~900 seconds from the order of astern to the development of astern thrust, is considered. The conventional astern thrust model (Fig. 7.9) that is appropriate to turbine ship is modified to a new one (Fig. 7.10) for diesel ship, and introduce some empirical formulae for the prediction of stopping ability of a ship fitted with diesel engine.



$$T \sim \begin{cases} 0, & (0 < t < t_c) \\ T_S, & (t_c < t) \end{cases}$$

Figure 7.10- Astern thrust model for diesel ship.

7.4 Questionnaire on the Application and Impact of IMO Resolution MSC.137(76)

Since the adoption of the *Interim Standards for Ship Manoeuvrability* (IMO Resolution A.751(18) in 1993 and the latest *Standards for*

Ship Manoeuvrability (IMO Resolution MSC.137(76) adopted on 4 December 2002, there have been some questions about how to apply them for different ships. ITTC member organisations are involved in the prediction of ships manoeuvrability and therefore interested in a correct and widely accepted interpretation of the currently adopted standards, both with regard to prediction methods, trials and ship types, including un-conventional vessels. In addition, some doubts still remain regarding the binding character (or not) of these standards.

For these reasons, the 24th ITTC Manoeuvring Committee has decided to make this questionnaire in order to investigate the views of the ITTC member organisations and others. The objective is to clarify how the standards are implemented today at different locations and hopefully contribute to improve the situation.

7.5 Questionnaire

A Questionnaire was circulated among 187 organizations including 129 ITTC member organizations, and receives the answers from 25 organizations (Table 7.3). The respondents are consisted of 13 manoeuvrability experts/ advisors (MA), 10 ship builders (SB), 1 classification society, and 1 flag state.

Table 7.3 shows that more than 300 cases regarding IMO Resolution MSC.137(76) and the interim standards Res. A751 has been dealt with in 2002 at the respondents and the number of cases are increased in 2003 by around 100 cases.

7.6 Questions and Answers to Questionnaire

In Tables 7.4-7.13 the questions and the corresponding answers are summarized. The answers have been split between Manoeuvrability Expert/Advisor (MA) and Ship builders (SB) as well as all together.

Table 7.3- Number of cases regarding IMO Resolution MSC.137(76) and the interim standards Res. A751 has your organization dealt with in 2003 and in 2002 .

	Organization	2002	2003
1	Akashi Ship Model Basin Co., LTD., Japan	0	5
2	DaeWoo Shipbuilding, Korea	10	15
3	Danish Maritime Authority, Denmark		15
4	FORCE Technology, Denmark	6	6
5	FSG, Flennsburger Schiffbau Gesellschaft, Germany	4	4
6	Germanischer Lloyd AG, Germany	130	150
7	Hanjin Heavy Industries, Korea	many	many
8	HDW, Germany	2	1
9	HSVA, Germany		
10	HWFSW, Germany	0	10
11	Hyundai Heavy Industries, Korea	35	40
12	IHI, Japan	0	2
13	IOT, St John’s, Canada	2	3
14	Krylov Shipbuilding Research Institute, Russia	2	
15	MARINTEK, Norway	5	5
16	Mitsubishi Heavy Industires, Inc., Japan	12	18
17	NMRI, Japan		
18	Samsung H. I., Korea	50	50
19	Ship Model Basin Duisburg VBD, Germany	0	0
20	SSPA, Sweden	30	30
21	SVA-Potsdam, Germany	3	4
22	Univ. of Michigan, U.S.A.	0	0
23	Universial Shipbuilding Co., Japan	15	20
24	MARIN, The Netherlands	20	20
25	INSEAN, Italy	3	4

Table 7.4- Does your organization consider IMO Resolution MSC.137(76) as mandatory?

	MA	SB	
Yes	5(38%)	4(40%)	9(39%)
No	4(31%)	5(50%)	9(39%)
No but recommend	3(23%)	0(0%)	3(13%)
No answer	1(8%)	1(10%)	2(9%)
	13	10	23

Table 7.5- What is the general approach of the **local Maritime Administration** regarding compliance with the Standards?

	MA	SB	
I don't know	6(46%)	2(20%)	8(35%)
No Requirements	1(8%)	2(20%)	3(13%)
Recommended as a guideline	4(31%)	0(0%)	4(17%)
No answer	2(15%)	6(60%)	8(35%)
	13	10	23

Table 7.6- What is the general approach of **ship owners**?

T5	MA	SB	
I don't know	3(23%)	0(0%)	3(13%)
Contract Term the violation of which issues penalty	4(31%)	3(30%)	7(30%)
Reporting required but the standard doesn't need to be satisfied	4(31%)	0(0%)	4(17%)
Depends on the ship	0(0%)	0(0%)	0(0%)
Have no terms on the manoeuvring performance	0(0%)	0(0%)	0(0%)
No answer	2(15%)	7(70%)	9(39%)
	13	10	23

Table 7.7- What is the general approach of the **classification society**?

	MA	SB	
I don't know	6(46%)	0(0%)	6(26%)
Mandatory for all kinds of ships	2(15%)	1(10%)	3(13%)
Recommended but not necessary	2(15%)	2(20%)	4(17%)
No answer	3(23%)	7(70%)	10(43%)
	13	10	23

Table 7.8- Does the contract between the ship owner and the ship builder normally contain clauses for the IMO Resolution MSC.137(76) standards (or other manoeuvrability standards)? (yes/no, which).

	MA	SB	
Yes	7(54%)	5(50%)	12(52%)
No	2(15%)	5(50%)	7(30%)
No answer	4(31%)	0(0%)	4(17%)
	13	10	23

Table 7.9 The standards were developed for ships with conventional propulsion and steering systems. How should they be applied for **other propulsion/steering** arrangements according to your organisations opinion/experience? (pods, flap rudders, azimuthing thrusters, water jets etc.)

	MA	SB	
I don't know or no special consideration	2(15%)	3(30%)	5(22%)
The same	3(23%)	3(30%)	6(26%)
Special considerations are being made	8(62%)	1(10%)	9(39%)
Owner requirement	0(0%)	1(10%)	1(4%)
No answer	0(0%)	2(20%)	2(9%)
	13	10	23

Table 7.10 How does your organization deal with the problem of documenting trial results in the full load condition for ships where trials can only be performed in a ballast condition? (Container ships, dry cargo, bulk carriers etc.)

	MA	SB	
No consideration or experience	3(23%)	2(20%)	5(22%)
Ballast sea trial	0(0%)	1(10%)	1(4%)
Full load model test	3(23%)	3(30%)	6(26%)
Simulation for full load condition	5(38%)	4(40%)	9(39%)
No answer	2(15%)	0(0%)	2(9%)
	13	10	23

Table 7.11- What manoeuvring standards does your organization apply for **ships shorter than 100 m, high-speed vessels, yachts and navy vessels**?

	MA	SB	
No consideration or experience	1(8%)	5(50%)	6(26%)
The same standards	3(23%)	0(0%)	3(13%)
Special standards of their own	5(38%)	2(20%)	7(30%)
Owner requirement	2(15%)	0(0%)	2(9%)
No answer	2(15%)	3(30%)	5(22%)
	13	10	23

Table 7.12- How does your organization **demonstrate compliance with IMO Resolution** MSC.137(76) (①Captive model tests, ②free model tests, ③prediction method, ④sea trials)?

	MA	SB	
④	0(0%)	2(20%)	2(9%)
①+②	3(23%)	0(0%)	3(13%)
①+③	0(0%)	2(20%)	2(9%)
①+④	0(0%)	2(20%)	2(9%)
②+③	3(23%)	1(10%)	4(17%)
③+④	1(8%)	1(10%)	2(9%)
①+②+③	3(23%)	1(10%)	4(17%)
②+③+④	0(0%)	1(10%)	1(4%)
No answer	3(23%)	0(0%)	3(13%)
	13	10	23

Table 7.13- Did you notice an increased awareness for manoeuvrability issues due to the adoption of the new IMO Standards in December 2002?

	MA	SB	
Yes	6(46%)	8(80%)	14(61%)
No	7(54%)	1(10%)	8(35%)
No answer	0(0%)	1(10%)	1(4%)
	13	10	23

7.7 Summary

The MC finds the results of the questionnaire relatively mixed which is in line with what was expected. There are no clear trends as to how the IMO criteria should be applied in the design process of new ships, both for conventional as well as un-conventional ships. This is exemplified by the fact that for conventional ships approximately half the respondents consider the IMO resolution as mandatory whereas the other half either considers them as recommendations or not mandatory. For ships equipped with unconventional propulsion/ steering systems, 60% of shipbuilders (SB) apply the same standards as conventional one, but special considerations are being made for those ships by 58% of manoeuvrability experts/Advisors (MA).

Clearly, it is recommended that the steering and manoeuvring capabilities of unconventional ships should be properly investigated before the ship is actually built.

It can be also concluded that due to the adoption of IMO Manoeuvring Standards, the number of cases regarding ship manoeuvrability is increasing rapidly, and 80% among shipbuilders (SB) notice an increased awareness for manoeuvring issues.

8. QM PROCEDURES

8.1 Status Current MC QM Procedures

Currently MC is responsible for 5 QM procedures:

7.5-02-05-05 Manoeuvrability Evaluation and Documentation HSMV
7.5-02-06-01 Free Model Test
7.5-02-06-02 Captive Model Test
7.5-02-06-03 Validation Manoeuvring Simulation Models
7.5-04-02-01 Full Scale Manoeuvring Trials

7.5-02-05-05, 7.5-02-06-03, 7.5-04-02-01 were corrected for minor errors. 7.5-02-06-01 was improved and updated. 7.5-02-06-02 was improved for addition of circular motion tests. The MC found that the current UA for captive model tests included in 7.5-02-06-02 while discussing some of the important error sources, does not follow procedure 7.5-02-01-01 Uncertainty Analysis in EFD" and thereby provide quantitative uncertainty estimates. Therefore, the MC initiated an effort of developing UA procedures for captive model tests, as described in Section 8.3. Progress on UA for free model tests is described in Section 8.4. The MC also cooperated with the Resistance Committee on preparation of a procedure for estimating facility biases, as described in Section 8.2.

8.2 Facility Biases

UA for physical model predictions of hydrodynamic performance of ships and marine installations is central to the mission of the ITTC. For individual facility UA, the 22nd ITTC adopted QM Procedure 7.5-02-01-01 Uncertainty Analysis in EFD, as recommended by the Resistance Committee. Procedure follows most recent international standards and enables quantitative estimates, but does not address issues related to facility or measurement system biases or certification of facilities.

Most work on facility or measurement system (MS) biases is for small-scale flow meter calibration facilities with focus on validation of accuracy, comparison of international flow standards, and establishing domestic flow traceability (Mattingly, 2001). Proficiency testing programs are used to establish flow measurement traceability, which are largely based on Youden plots (Youden, 1959) requiring two (e.g., tandem and/or upstream and downstream) MS at each facility. This approach not easily extended to large-scale multipurpose facilities with complex MS, including consideration individual facility and measurement systems bias and precision limits.

Individual facility and measurement systems bias and precision limits are required for use of such data as well as helpful in MS improvements.

For large-scale facilities such as wind tunnels and towing tanks with complex MS, only limited work done and facility or MS biases not yet considered. The NATO, AGARD, Propulsion and Energetics Panel, Uniform Engine Testing Program, was a remarkable early exercise in large-scale testing in which the same jet engines were tested in a number of jet engine test stands in various NATO countries and uncertainties were estimated to explain whether data scatter was within the data uncertainty and conclusions were drawn (Vleghert, 1989). Gooden et al. (1997) compares results from wind tunnel tests for same geometry and conditions at two different institutes, model scales, and using a number of different measurement techniques and extensive error-analysis. The Cooperative Experimental Program of the Resistance Committees of the 17-19 International Towing Tank Conferences (ITTC) compare results from towing tank tests at 22 institutes. Comparisons are made of global (resistance, sinkage and trim, wave profile, wave cut, wake survey, form factor, and blockage) and local (surface pressure and boundary layer traverses) data for a standard geometry (Series 60) of different sizes (1.2-9.6 m). However, uncertainty assessment not considered. The cooperative uncertainty assessment example for resistance test of the Resistance Committee of the 22nd ITTC compare results from towing tank tests at 7 institutes of resistance test bias and precision limits and total uncertainties following standard uncertainty assessment procedures, but for different model geometries and sizes (Series 60, container ships, and 5415).

Recently, Stern et al. (2004) proposed a statistical approach for estimating intervals of certification or biases of facilities or MS including uncertainties. N-order level testing reviewed followed by definitions for MxN-order level testing, which defined as M repetitions of the same N-order level experiment in M different facilities or in the same facility with M different measurement systems. If reference values known, approach used at either the N-order or MxN-order levels. However, unlike CFD where EFD provides reference values, for EFD reference values are seldom known, e.g., from a standard facility or MS. In absence of reference values, mean facility or MS used for assessing intervals of certification or biases. Certification or biases of facilities or measurement systems are defined as processes for assessing probabilistic confidence intervals for facilities or measurement systems for specific tests, data reduction equations, conditions, procedures, and uncertainty analysis. Similarly, subgroup analysis performed for isolating and assessing levels of differences due to use of different model sizes (scale effects) or measurement systems. An example provided for towing tank facilities for resistance tests using standard uncertainty analysis procedures based on an international collaboration between three facilities.

The 24th ITTC Resistance Committee will follow the Stern et al. (2004) approach in conjunction with round robin testing using 3 and 5.7 m 5415 models to identify facility biases for resistance tests.

8.3 UA for Captive Model Tests

UA examples following QM Procedure 7.5-02-01-01 Uncertainty Analysis in EFD are needed for all physical model tests of interest to the ITTC, as done previously for resistance test and provided by QM Procedure 7.5-02-02-02 UA Example for Resistance Test. In particular UA example is needed for static and dynamic PMM tests, although considerably more difficult than relatively more simple resistance test. The MC made considerable progress on UA example for static and dynamic PMM tests with results thus far completed from one institute (Simonsen, 2004) and in progress for two institutes.

Summary UA Example for PMM Test. The purpose of the procedure is to provide an example of UA for static (pure drift) and dynamic (pure sway and yaw and combined yaw and drift) PMM tests. UA estimates provided for hydrodynamic axial and side forces and yaw moment: mean values for static tests and eight harmonic Fourier series faired values for dynamic tests. Model is free in pitch and heave, but constrained in roll at zero heel angles. Errors due to non-harmonic motions and carriage accelerations are not yet considered. The procedure only covers the measured hydrodynamics forces and moment. This means that uncertainties related to the traditional manoeuvring coefficients and their application in conjunction with manoeuvring simulations not dealt with.

The data reduction equations for longitudinal and transverse forces and yaw moment are:

$$X' = \frac{F_{X_{measured}} + M(\dot{u} - rv - X_G r^2 - Y_G \dot{r})}{0.5\rho U^2 T_m L_{pp}} \tag{8.1}$$

$$Y' = \frac{F_{Y_{measured}} + M(\dot{v} + ru - Y_G r^2 + X_G \dot{r})}{0.5\rho U^2 T_m L_{pp}} \tag{8.2}$$

$$N' = \frac{M_{Z_{measured}} + I_Z \dot{r} + M(X_G(\dot{v} + ru) - Y_G(\dot{u} - rv))}{0.5\rho U^2 T_m L_{pp}^2} \tag{8.3}$$

Bias and precision limits estimated using the multiple test approach. Individual bias limits estimated for water density, carriage speed, model mass, moment of inertia, draft, length between perpendiculars, centre of gravity, heading angle, sway velocity, sway acceleration, yaw rate, yaw acceleration, surge velocity, surge acceleration, and measured F_X and F_Y forces and yaw moment M_Z. For the measured forces and moment, biases considered for drift angle setting, alignment of the model, calibration of the force gauges, data acquisition, surge velocity, sway velocity, yaw rate, surge acceleration, sway acceleration, yaw acceleration, and time. Precision limits were estimated end-to-end method based on 12 repeat tests.

Table 8.1- Uncertainties from the PMM test.

Fr	Test	X'			Y'			N'		
		$B_{X'}$ (%)	$P_{X'}$ (%)	$U_{X'}$ (% X')	$B_{Y'}$ (%)	$P_{Y'}$ (%)	$U_{Y'}$ (% Y')	$B_{N'}$ (%)	$P_{N'}$ (%)	$U_{N'}$ (% N')
0.138	Static drift ($\beta = 10^\circ$)	97.2	2.8	11.3	79.0	21.0	3.5	69.2	30.8	2.2
	Static drift (IIHR, $\beta = 10^\circ$)	100.0	0.0	22.5	88.6	11.4	4.9	96.3	3.7	9.5
	Pure yaw ($r_{max} = 0.57$)	97.6	2.4	11.3	90.3	9.7	15.8	98.8	1.2	7.3
0.280	Static drift ($\beta = 10^\circ$)	77.8	22.2	3.4	74.2	25.8	2.1	21.0	79.0	2.4
	Pure yaw ($r_{max} = 0.49$)	98.8	1.2	3.4	93.4	6.6	5.5	93.9	6.1	3.3
	Pure sway ($v_{max} = 0.29$)	98.1	1.9	3.1	98.3	1.7	1.8	92.6	7.4	1.5
	Yaw & drift ($\beta = 10^\circ$)	99.2	0.8	5.8	89.0	11.0	2.1	98.0	2.0	2.7
0.410	Static drift ($\beta = 10^\circ$)	89.6	10.4	1.6	69.6	30.4	1.8	43.4	56.6	1.4
	Pure yaw ($r_{max} = 0.60$)	98.0	2.0	2.9	90.6	9.4	3.5	87.7	12.3	1.4

Table 8.1 summarizes the results, which are reasonable and acceptable in comparison previous results for resistance tests. Repeatability is good such that in general bias limits are larger than precision limits, except for static drift yaw moment and medium and high speed. Calibration of the force gauges, drift angle setting, surge and sway velocities, and carriage speed identified as the largest error sources and candidates for improvements. Uncertainties are larger for lower than higher speeds due to use of load cells covering both small and large amplitudes. Uncertainties are larger for smaller model (IIHR) due to influence of increased sensitivity coefficients for larger than smaller model size, at least for static drift test.

Future work will focus on combined results and conclusions from at least three facilities, effects of roll, non-harmonic motion, carriage acceleration, and biases due to sinkage. The current UA for captive model tests included in 7.5-02-06-02 may be useful for estimating biases due to non-harmonic motion and carriage acceleration.

8.4 UA for Free Model Tests

Progress on including UA for free model test in 7.5-02-06-01 was slower than anticipated; however, it is expected that UA for free-model tests greatly facilitated by completion UA for captive model tests such that the 25th MC can compete both. The 24th MC received a written contribution from the Bulgarian Ship Hydrodynamics Centre (Chotukova and Milanov, 2005), which includes preliminary UA for free model tests following 7.5-02-01-01 Uncertainty Analysis in EFD."

Section 2.4, Summary of Methodology of 7.5-02-01-01 provides overview of UA procedure. (1) Determine data reduction equations for all measured data. (2) Estimate bias limits using block and data stream diagrams usually at the elemental and individual measurement system levels and propagation of error equations. (3) Estimate precision limits usually using multiple test method with end-to-end data acquisition and reduction. (4) Provide elemental and measurement system contributions to bias limits along with total bias and precision limits and total uncertainty.

Results (measure data) for the free model tests listed in Section 2.3.1 of 7.5-02-06-01. Thus, first step is to derive data reduction equations for each result r usually in non-dimensional form,

$$r = r(X_1, ... X_J) \qquad (8.4)$$

where,
X_i are the individual measured variables. As already mentioned, bias limits estimated at the Xi and sub level, whereas precision limits estimated at r level. Chotukova and Milanov (2005) perform a preliminary UA with consideration to many of the elemental bias errors, but do not derive data reduction equations or estimate precision limits.

9. CONCLUSIONS

9.1 Section 2: Progress in Systems-Based Manoeuvring Simulations

1. At early design stage regression methods used, whereas at later design stages captive and free model tests are used, often followed by system identification. CFD expected to play larger role in future.
2. Wide variety of methods is used and there is a crucial need for validation at both model (i.e., using free model tests results) and full scale to distinguish their strengths and weaknesses.
3. In many cases, it is difficult to evaluate the approach due to insufficient documentation.
4. Weakness of systems-based methods is a proper understanding of propeller/hull/ rudder interaction and scale effects, for which CFD will help.

9.2 Section 3: Progress in CFD-Based Manoeuvring Simulations

1. There is an impressive and rapid development URANS from resistance and propulsion to seakeeping and manoeuvring (static and dynamic), but validation still lacking.
2. CFD methods used to predict hydrodynamic forces/derivatives for use by system based simulation methods as a supplement to captive model tests and data base regression methods.
3. CFD should improve systems-based methods especially for propeller/hull/rudder interaction and scale effects as well as physical understanding assuming CFD sufficiently verified and validated.
4. After CFD validated against forced motions tests, focus will move to prediction of trajectories and validation using free model tests data.
5. Required resources, lack of trained users, user-friendly codes, and need for V&V are pace setting issues for more widespread use of CFD in practice.

9.3 Section 4: Benchmark Data

1. Global data available but in some cases lack of documentation conditions, inconsistencies between data for same hulls, and old hull forms are serious problems.
2. Local flow data is missing, as required for understanding of rudder-propeller-hull interaction and CFD validation.
3. Need for modern hull form data, including effects stern shape variation on manoeuvring such as Japanese A&B and KRISO VLCC1 and VLCC2.
4. Need for new full-scale data.
5. VLCC1 and VLCC2, KCS and 5415 along with other ships listed in our report should be included as manoeuvring benchmarks.
6. We recommend that PMM free model and local flow data procured including UA for the manoeuvring benchmarks.

9.4 Section 5: High speed and Unconventional Vessels

1. Studies of high speed and unconventional vessels more common indicating increased activity.

9.5 Section 6: Confined Waters

1. There are papers on shallow water, bank, especially ship/ship interactions, and squat.
2. CFD used for shallow water, banks and ship-ship interaction.
3. Model tests performed for shallow water, banks and ship-ship interaction.
4. Increased focus on confined waters combined with ship-ship interaction due to an increase in ship sizes and cargo import/export.
5. Global benchmark data are available but in some cases lack of documentation conditions, inconsistencies between data for the same hull and old hull forms are a serious problem.

9.6 Section 7: Manoeuvring Standards and Safety

1. As a conclusion to the manoeuvring standards, it is recommended to continue the works to make sure that the IMO standards are satisfactory and provide sufficient constraints to results in reasonable handling qualities for safe operation at sea at service speed.
2. Future MC may actively participate IMO, e.g., revise the existing criteria and develop criteria for safety on low speed/shallow water/harbour.
3. As a conclusion to the questionnaire the MC recommends that the 25th Manoeuvring committee continues to clarify the implications of the IMO manoeuvring standards to obtain a more consistent interpretation of how the members of the ITTC should work with these criteria.

9.7 Section 8: QM Procedures

1. Progress on UA for facility biases and captive and free model tests is promising but application by more ITTC members is required for finalizing quality manual procedures.

10. RECOMMENDATIONS

Adopt the improved procedure 7.5-02-06-01, "Testing and Extrapolation Methods, Manoeuvrability, Free Running Model Test Procedure"

Adopt the improved procedure 7.5-02-06-02, "Testing and Extrapolation Methods, Manoeuvrability, Captive Model Test Procedure"

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11.2 Nomenclature

FAST	International Conference of Fast Sea Transportation
HIPER	International Euro Conference on High-Performance Marine Vehicles
IMDC	International Marine Design Conference
MARSIM	International Conference on Marine Simulation and Ship Manoeuvrability
T-POD	International Conference on Technological Advances in Pod Propulsion