# The Resistance Committee

## Final Report and Recommendations to the 26th ITTC

## 1. INTRODUCTION

### 1.1 Membership

*Chairman:*
Dr. Joseph Gorski
Naval Surface Warfare Center, Carderock Division, UNITED STATES OF AMERICA

*Secretary:*
Prof. Stephen Turnock
University of Southampton,
UNITED KINGDOM

*Members:*
Dr. Bertrand Alessandrini
Ecole Centrale Nantes (ECN), FRANCE

Dr. Ho-Hwan Chun
Pusan National University, KOREA

Dr. Uwe Hollenbach
Hamburg Ship Model Basin (HSVA), GERMANY

Dr. Tommi Mikkola
Aalto University, FINLAND

Dr. Yusuke Tahara
National Maritime Research Institute (NMRI), JAPAN

Dr. Jesús Valle
Escuela Técnica Superior de Ingenieros Navales (ETSIN), SPAIN

Dr. Liangmei Ying
China Ship Scientific Research Centre (CSSRC), CHINA

### **1.2** Meetings

The committee met 4 times:
13-15 January 2009, Madrid, Spain
15-16 September 2009, Busan, Korea
18-19 May 2010, Nantes, France
18-19 January 2011, Hamburg, Germany

### **1.3** Tasks

Below we list the tasks carried out by the 26th resistance committee (RC), based on the recommendations given by the 25th ITTC.

1. Update the state-of-the-art for predicting the resistance of different ship concepts emphasising developments since the 2008 ITTC Conference. The committee report should include sections on:
   a) the potential impact of new technological developments on the ITTC,
   b) new experimental techniques and extrapolation methods,
   c) new benchmark data,
   d) the practical applications of computational methods to resistance predictions and scaling,
   e) the need for R&D for improving methods of model experiments, numerical modelling and full-scale measurements.
2. Review ITTC Recommended Procedures relevant to resistance (including procedures for uncertainty analysis).

a) Identify any requirements for changes in the light of current practice, and, if approved by the Advisory Council, update them.
b) Identify the need for new procedures and outline the purpose and content of these.
c) With the support of the Specialist Committee on Uncertainty Analysis, review and if necessary amend, Procedures 7.5-02-02-03, 04, 05 and 06 "Uncertainty Analysis spreadsheets for measurements of resistance, speed, sinkage and trim and wave profile" and Procedure 7.5-0302-01 "Uncertainty Analysis in CFD Example for Resistance and Flow to bring them into line with the ISO approach adopted by the ITTC.

3. Identify the parameters that cause the largest uncertainties in the results of model experiments, numerical modelling and full-scale measurements related to resistance.
4. Survey and document the range of practices adopted for turbulence stimulation. Update parts of Recommended Procedure 7.5-01-01-01, Model Manufacture which deal with turbulence stimulation, paying particular attention to different kinds of bulbous bows and high-speed ships. Liaise with the Specialist Committee on High-Speed Craft.
5. Review ITTC Recommended Procedures relevant to scaling and extrapolation methods including theoretical and experimental investigations of the friction line. *Note: At the present time the ITTC does not consider introducing a new friction line without extensive validation.*
6. Make the concept of form factor consistent in all relevant ITTC procedures. Include the form factor in the formulation of the ITTC 1957 friction line as an option in ITTC Recommended Procedure 7.5-02-02-01, "Resistance Tests".
7. Review methods used for the scaling of appendage resistance, especially in relation to the problem of pod drag scaling. Ensure that the appendage drag scaling is treated consistently for resistance, propulsion and the 1978 Powering Performance Prediction Method. Liaise with the Propulsion Committee.
8. Continue the tests in the ITTC worldwide series for identifying facility biases. Prepare a common calculation sheet to analyze the results of the tests. Check and record the model dimensions regularly.

## 2. FACILITY BIAS WORLDWIDE CAMPAIGN

The 24th ITTC Resistance Committee invited all the ITTC members to participate in a worldwide series of comparative tests for identifying facility biases under the framework of ITTC procedures for uncertainty analysis. The tests were done during the 24th and 25th ITTCs and continued during the 26th ITTC period.

For these tests two geosims of the DTMB 5415 Combatant with 5.720 and 3.048 meters length, respectively, have been used.

The Committee created and distributed a technical procedure for identifying facility biases, compiling model and test procedure information, including data submission guidance to preserve the confidentiality of the data.

Facility biases have been analysed for the following most typical towing tank tests:

- Resistance
- Sinkage and trim
- Wave profile and wave elevations

### 2.1 Participants

In the 24th ITTC 20 institutions of 15 countries participated in the worldwide campaign. The number of institutions was increased during the 25th ITTC to 35 and the number of countries to 19. The number of participating institutions increased again during the 26th ITTC to the number of 41 and the number of countries to 20.

The tests for the 5.720 m length model finished at the beginning of 2011, on time for the 26th ITTC. The participant institutions, their countries and the month of reception of the model are shown in the following table.

Table 1. 5.720 m length model tests.

| Institution | Country | Reception month |
|---|---|---|
| CEHIPAR | Spain | Jun 2004 |
| INSEAN | Italy | Sep 2004 |
| Helsinki University of Technology | Finland | Nov 2004 |
| Krylov Shipbuilding Research Institute | Russia | Feb 2005 |
| ICEPRONAV S.A. | Romania | Sep 2005 |
| Vienna Model Basin | Austria | Dec 2005 |
| Huazhong University of Science and Technology | China | ------- |
| CSSRC | China | Sep 2007 |
| Samsung Ship Model Basin | Korea | Dec 2007 |
| MOERI | Korea | Feb 2008 |
| Pusan National University | Korea | Apr 2008 |
| Akashi Laboratory | Japan | Aug 2008 |
| IHI Corporation | Japan | |
| Universal Zosen | Japan | |
| Akisima Laboratory | Japan | |
| Ship Research Center (NMRI) | Japan | |
| Osaka University | Japan | |
| Naval Surface Warfare Center | USA | Mar 2009 |
| Institute for Ocean Technology | Canada | Sep 2009 |
| QinetiQ | UK | Apr 2010 |
| Bassin d'Essais des Carenes | France | Aug 2010 |
| CEHIPAR | Spain | Mar 2011 |

The tests for the 3.048 m length model did not finish on time for the 26th ITTC due to the following causes:

- The accumulated delays, the internal planning and the amount of work of the institutions increased the testing periods.
- The required time to move the models between institutions was longer than predicted.
- There was a problem sending the model from Brazil to the United Kingdom. When these proceedings were written the model was still retained in the customs and that is the reason why the remaining tests have been postponed.

The provisional schedule, indicating the month of reception of the model, is summarized in the following table.

Table 2. Schedule for the 3.048 m length model.

| Institution | Country | Reception month |
|---|---|---|
| CEHINAV | Spain | Feb 2005 |
| LSMH/NTUA | Greece | Apr 2005 |
| Inha University | Korea | Dec 2005 |
| Seoul National University | Korea | Jan 2006 |
| Pusan National University | Korea | Feb 2006 |
| Ulsan University | Korea | Mar 2006 |
| Harbin Engineering University | China | ------- |
| University Teknologi Malaysia | Malaysia | Sep 2006 |
| Australian Maritime College | Australia | Nov 2006 |
| Canal de Experiencias de Arquitectura Naval | Argentina | Feb 2007 |
| University of Iowa – IIHR | USA | Jul 2007 |
| Stevens Institute of Technology | USA | Jan 2009 |
| First Memorial University | Canada | Sep 2009 |
| Institute for Ocean Technology | Canada | Nov 2009 |
| IPT – Instituto de Pesquisas Tecnológicas do Estado | Brazil | Apr 2010 |
| University of Glasgow and Strathclyde | UK | Postponed |
| University of Liège – ANAST | Belgium | Postponed |
| Ecole Centrale de Nantes | France | Postponed |
| Istanbul Technical University | Turkey | Postponed |
| INSEAN | Italy | Postponed |
| CEHIPAR | Spain | Postponed |

## 2.2 Dimensional Control of the Model

In order to guarantee the quality of the results the dimensions of the model were controlled in some facilities. Minor scratches and paint failures were detected for both models and they were repaired when necessary. There were not important deformations of the model.

A complete comparison of the dimensions of the 5.720 m length model was done at CEHIPAR, comparing the measurements of the model after the tests with the original data used to manufacture the model. There were not important deviations which could affect the tests.

## 2.3 Testing Procedure and Data Submission

Each institution has tested the model in 4 different sessions, in order to change the test conditions and obtain better uncertainty analysis results. All the Institutions have used their standard techniques to test the models and have corrected their results taking into account the blockage effects, using their standard procedures.

During each session, 10 runs were done, with Froude numbers 0.10, 0.28 and 0.41.

The results of the tests have been submitted to the Resistance Committee using ASCII neutral files. The content of these files is specified in the technical procedure for identifying facility biases and the proceedings of the 25th ITTC.

## 2.4 Analysis Method

In January 2008, the Resistance Committee and the Uncertainty Analysis Committee had their first meetings in Madrid. That was a good opportunity to meet together and discuss the analysis method to be used for the facility bias worldwide campaign.

The Uncertainty Analysis Committee proposed a new analysis method testing two geosims of the same model in each facility. The Resistance Committee agreed with this analysis method but it was unviable at this stage of the worldwide campaign, so the decision was to continue with the analysis method proposed in the 24th and 25th ITTCs.

This analysis method, based on $M \times N$-order level testing, where $N$ repetitions of the same experiment are done in each of the $M$ different facilities participating in the experience, was detailed in the Proceedings of the 25th ITTC.

## 2.5 Analysis Program

Due to the number of calculations and the great amount of data used in the analysis method, a computer program has been created to facilitate the analysis.

A new version of the program with corrected bugs is available on the ITTC web page. The user manual, the submitted data for both models and the program installer are included in the following files:

- Manual.pdf
- DataForLargeModel.zip
- DataForSmallModel.zip
- ITTC Setup.msi

When these proceedings were written, the submitted data for both models was not complete, so only analysis of the existing data has been done. The entire data set for the analysis will be actualized in the ITTC web page by changing the following files.

- DataForLargeModel.zip
- DataForSmallModel.zip

## 2.6 Submitted Data

During the 25th ITTC only 9 data sets were received, 4 for the large model and 5 for the small one. Due to format problems only 3 data sets for the large model and 5 data sets for the small one were valid for the analysis.

During the 26th ITTC 19 data sets were received, 14 for the large model and 5 for the small one. Nevertheless, only 9 data sets were valid for analysis of the large model, because one data set was tested for erroneous velocities and had incoherent data format, three data sets were sent twice and another one was previously included in the data received for the 25th ITTC.

The distribution of the submitted data is summarized in Table 3.

Table 3. Distribution of the submitted data.

| Model | Nr. of facilities which tested the model | Nr. of facilities which sent data | Nr. of valid data sets |
|---|---|---|---|
| 5.720 m | 21 | 14 | 12 |
| 3.048 m | 15 | 10 | 10 |

The data for each model has been arranged in folders that have been numbered. The number of the folders does not correspond with the reception order or the test schedule. This procedure guarantees the confidentiality of the submitted data.

The data has been reformatted because many facilities have sent only the medium values of the tests, instead of the temporal data, and there were many errors in the data format, which were corrected. Also, 30% of the received files were in a different format than specified in the instructions for data submission and that made the analysis difficult.

Some facilities have also sent a copy of the ITTC spreadsheets for uncertainty analysis, and many of them used the values of the example instead of changing the values in correspondence with their equipments.

All data and the results of the analysis are available for all the ITTC members, so each institution can identify its own data, and consequently its folder number, comparing the submitted data with the data available in each folder.

The main results of the analysis are summarized in the following sections.

## 2.7 Resistance

For each model the following data is presented:

- The total resistance coefficients for each facility $(C_T)_i$ compared with their mean value $\overline{C_T}$ .
- The uncertainties of the resistance coefficients for each facility $(U(C_T))_i$ compared with their mean value $\overline{U(C_T)}$ , expressed in percentage of $(C_T)_i$ .
- The uncertainties of the facility bias for each facility $(U_{FB}(C_T))_i$ compared with their mean value $\overline{U_{FB}(C_T)}$ , expressed in percentage of $(C_T)_i$ .
- The total uncertainties of the resistance coefficients for each facility $(U_t(C_T))_i$ compared with their mean value $\overline{U_t(C_T)}$ , expressed in percentage of $(C_T)_i$ .

All the facilities had sent resistance data, but in some cases data was missed for a particular Froude number. Some errors were detected in the uncertainties analysis for some particular Froude numbers and in those cases the data was withdrawn and not used in the analysis.

A mistake was detected in the spreadsheet for resistance uncertainty analysis: the conversion of the total resistance coefficient to 15ºC is not done as described in procedure 7.5-02-02-02. Nevertheless, this fact has not had an important relevance in the results.

In the following figures the dotted columns represent Fr =0.1 results, the stripped columns represent Fr = 0.28 results and the solid columns represent Fr = 0.40 results for the individual facilities. The solid line with squares is the average value for Fr = 0.1, the dashed line with triangles represents the average value for Fr = 0.28 and the dotted line with diamonds represent the average value for Fr = 0.41.

5.720 meters length model.

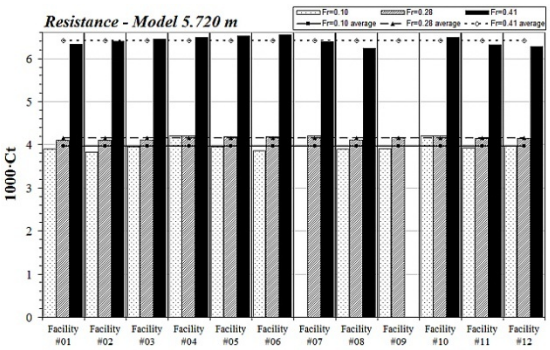


Figure 1 Large model resistance coefficient comparison

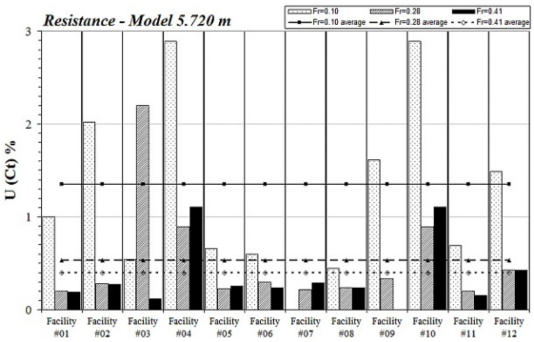


Figure 2 Large model uncertainties for resistance coefficient

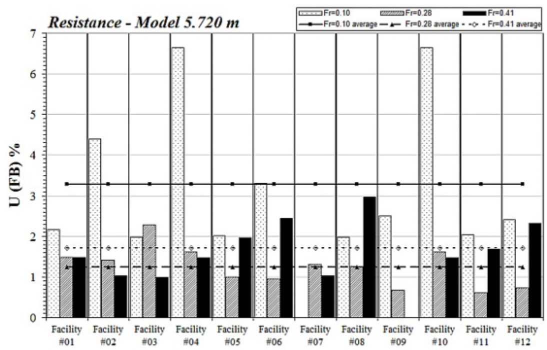


Figure 3 Large model uncertainties for resistance facility bias.

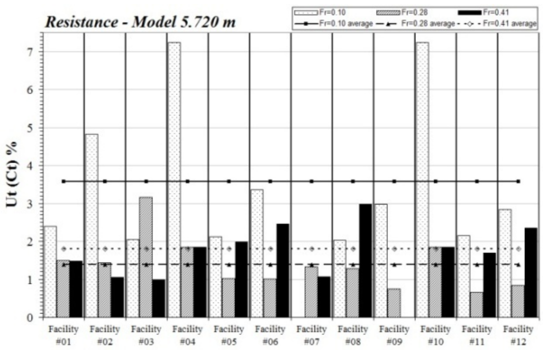


Figure 4 Large model total uncertainties for resistance coefficient

3.048 meters length model.

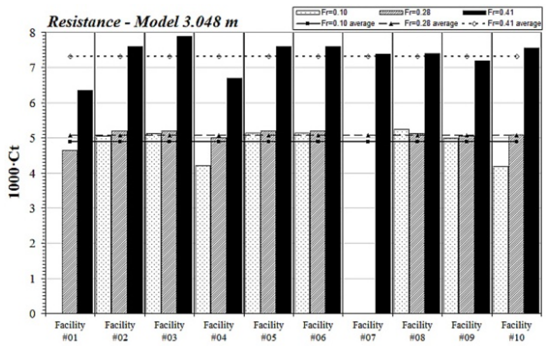


Figure 5 Small model resistance coefficient comparison

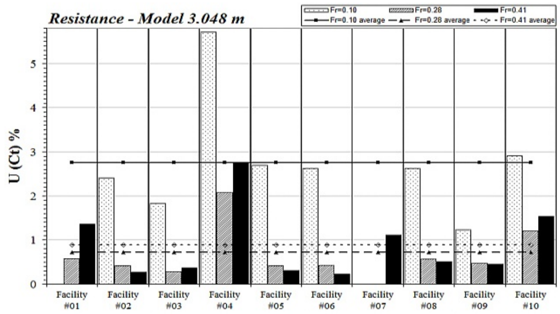


Figure 6 Small model uncertainties for resistance coefficient

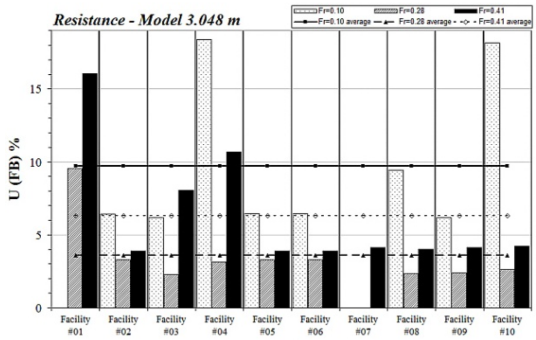


Figure 7 Small model uncertainties for resistance facility bias

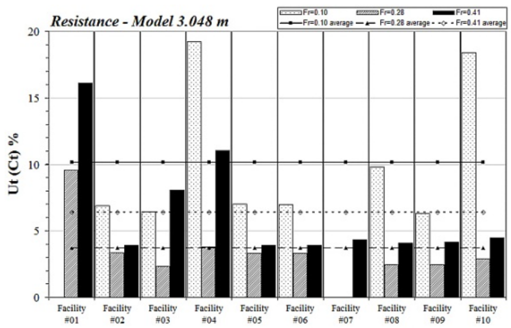


Figure 8 Small model total uncertainties for resistance coefficient

## 2.8 Sinkage

For each model the following data is presented:

- The sinkage value in mm for each facility $(z_s)_i$ compared with their mean value $\overline{z_s}$.
- The uncertainties of the sinkage for each facility $(U(z_s))_i$ compared with their mean value $\overline{U(z_s)}$, expressed in percentage of $(z_s)_i$.
- The uncertainties of the facility bias for each facility $(U_{FB}(z_s))_i$ compared with their mean value $\overline{U_{FB}(z_s)}$, expressed in percentage of $(z_s)_i$.
- The total uncertainties of the sinkage for each facility $(U_t(z_s))_i$ compared with their mean value $\overline{U_t(z_s)}$, expressed in percentage of $(z_s)_i$.

All the data sets submitted for the large model included sinkage data, but only 6 facilities for the small model did. In some cases data was missed for a particular Froude number. Some errors were detected in the uncertainty analysis for some particular Froude numbers and in these cases the data was withdrawn and not used in the analysis. There was not enough data to analyze the uncertainties for Froude number 0.1 for the small model.

<u>5.720 meters length model.</u>

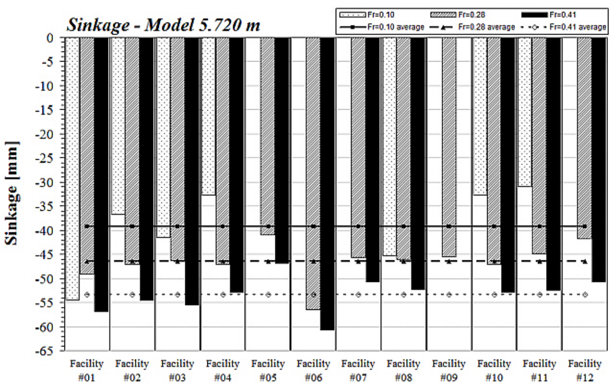


Figure 9 Large model sinkage comparison

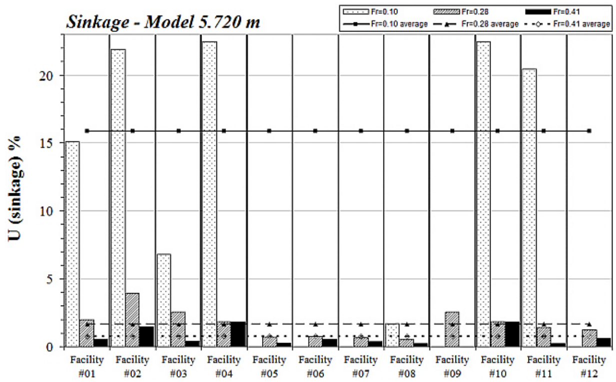


Figure 10 Large model uncertainties for sinkage

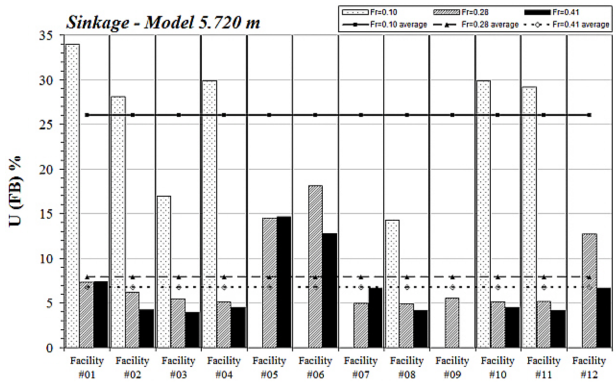


Figure 11 Large model uncertainties for sinkage facility bias

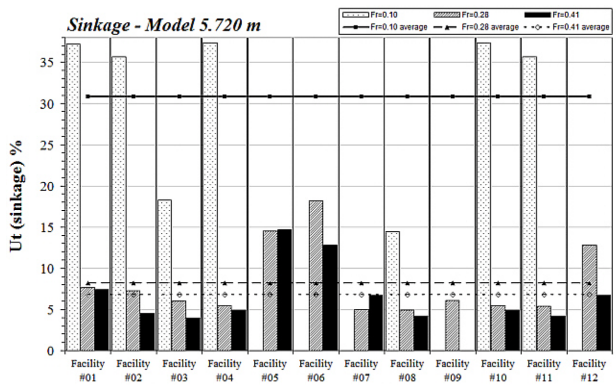


Figure 12 Large model total uncertainties for sinkage

3.048 meters length model.

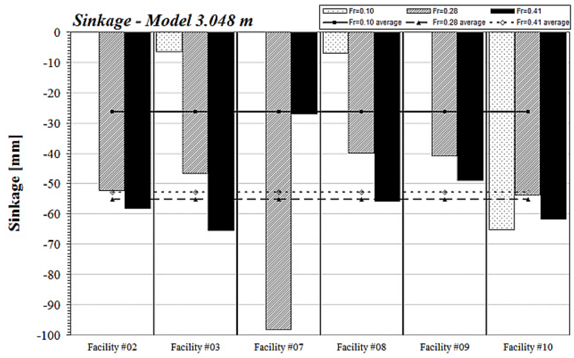


Figure 13 Small model sinkage comparison

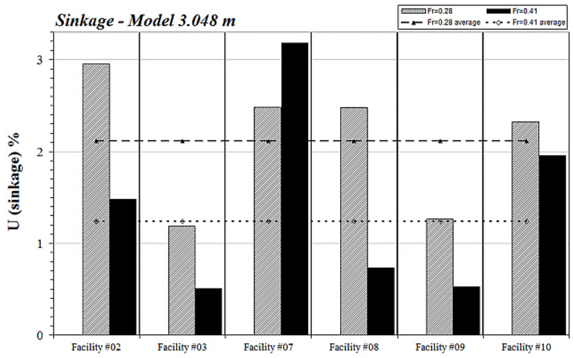


Figure 14 Small model uncertainties for sinkage

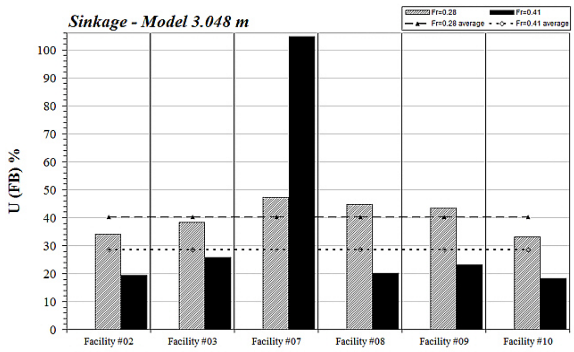


Figure 15 Small model uncertainties for sinkage facility bias

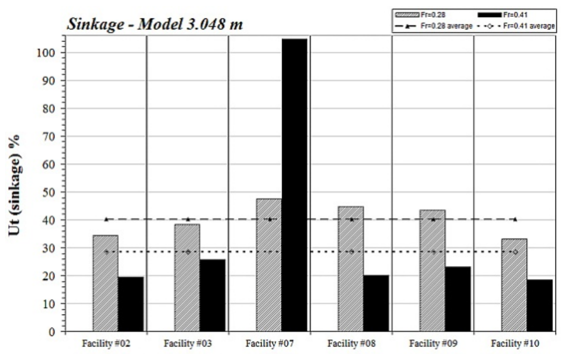


Figure 16 Small model total uncertainties for sinkage

## 2.9 Trim

For each model the following data is presented:

- The trim value in mm for each facility $(\tau)_i$ compared with their mean value $\overline{\tau}$ .
- The uncertainties of the trim for each facility $\left(U(\tau)\right)_i$ compared with their mean value $\overline{U(\tau)}$ , expressed in percentage of $(\tau)_i$ .
- The uncertainties of the facility bias for each facility $\left(U_{FB}(\tau)\right)_i$ compared with their mean value $\overline{U_{FB}(\tau)}$ , expressed in percentage of $(\tau)_i$ .
- The total uncertainties of the trim for each facility $\left(U_t(\tau)\right)_i$ compared with their mean value $\overline{U_t(\tau)}$ , expressed in percentage of $(\tau)_i$ .

All the data sets submitted for the large model included trim data but only 6 facilities for the small model did. In some cases data was missed for a particular Froude number. Some errors were detected in the uncertainty analysis for some particular Froude numbers and in these cases the data was withdrawn and not used in the analysis.

There was not enough data to analyze the uncertainties for Froude number 0.1 for the small model.

5.720 meters length model.

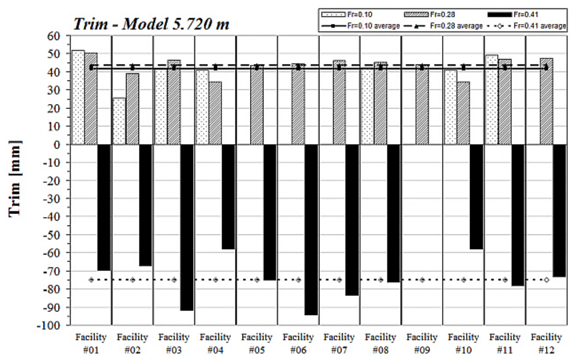


Figure 17 Large model trim comparison

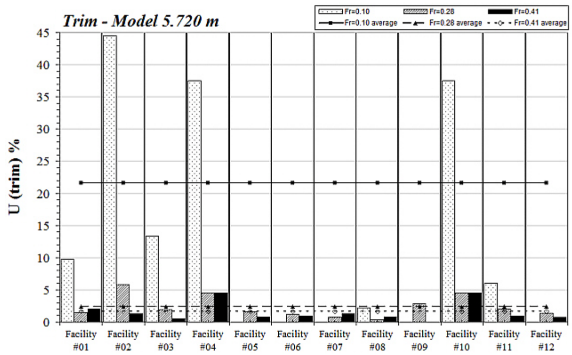


Figure 18 Large model uncertainties for trim

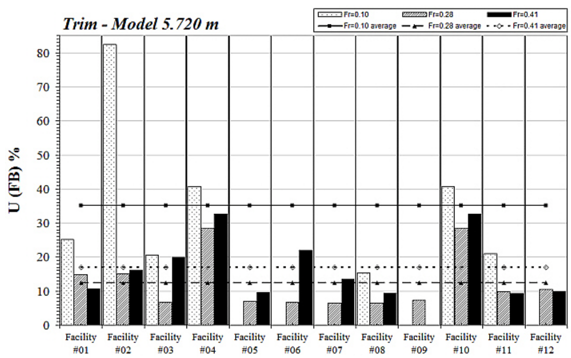


Figure 19 Large model uncertainties for trim facility bias

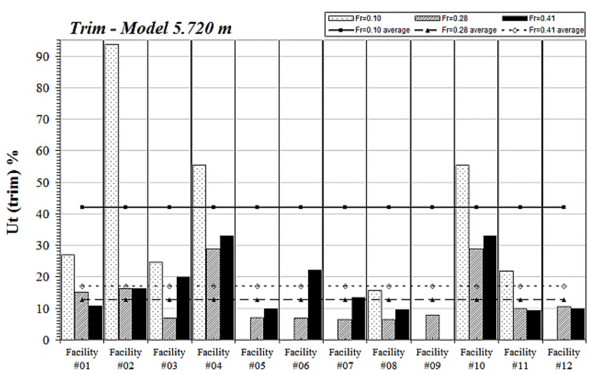


Figure 20 Large model total uncertainties for trim

3.048 meters length model.

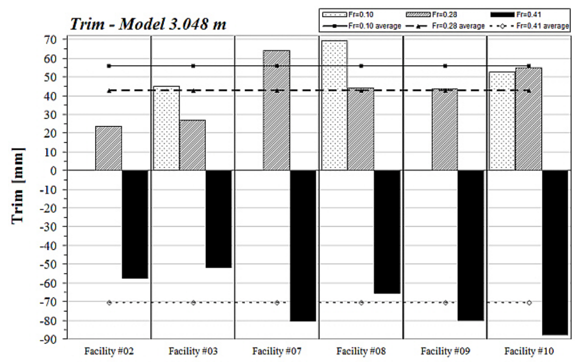


Figure 21 Small model trim comparison

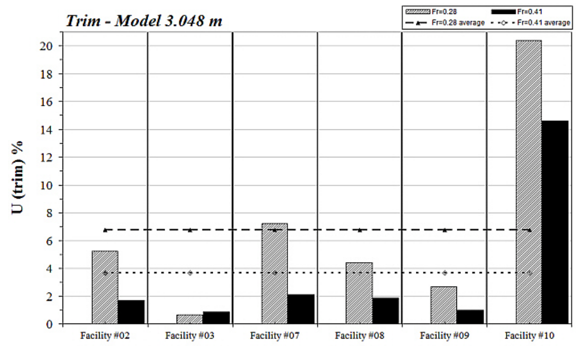


Figure 22 Small model uncertainties for trim

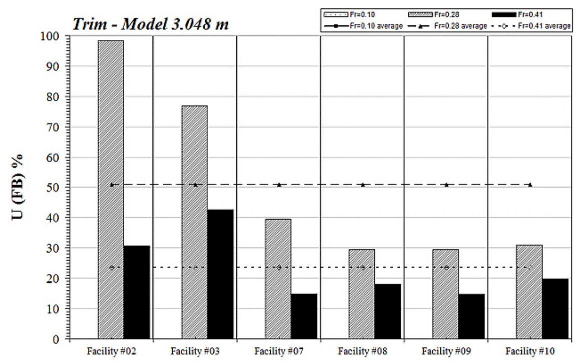


Figure 23 Small model uncertainties for trim facility bias

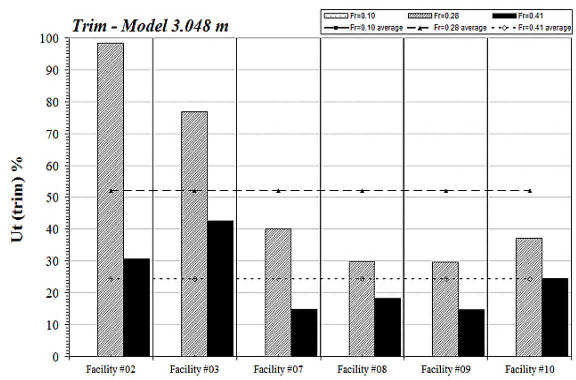


Figure 24 Small model total uncertainties for trim

## 2.10 Wave Profiles

Only 1 facility submitted wave profile data for the small model and 5 facilities for the large one, but 3 of them had almost zero data values, so they have not been taken into account.

There were measured profiles only for 2 facilities for the large model and 1 facility for the small model. The dispersion of the values was high, as it can be seen in the following figures, and the uncertainties were high and not conclusive, so they are not presented in this report. In the below figures the longitudinal position is measured along the hull from section 0 (positive to bow).

5.720 meters length model.

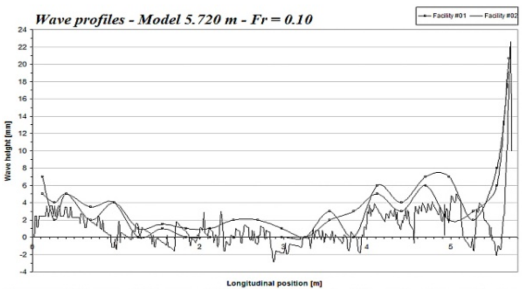


Figure 25 Large model wave profiles for $Fr = 0.10$

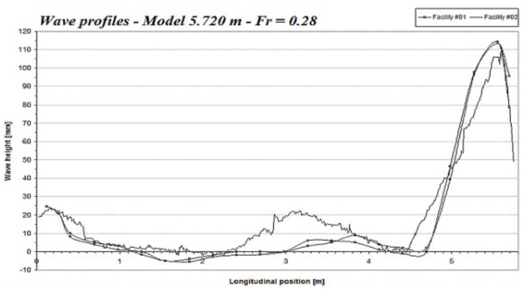


Figure 26 Large model wave profiles for $Fr = 0.28$

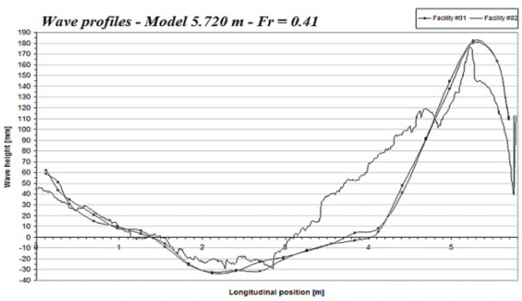


Figure 27 Large model wave profiles for $Fr = 0.41$

3.048 meters length model.

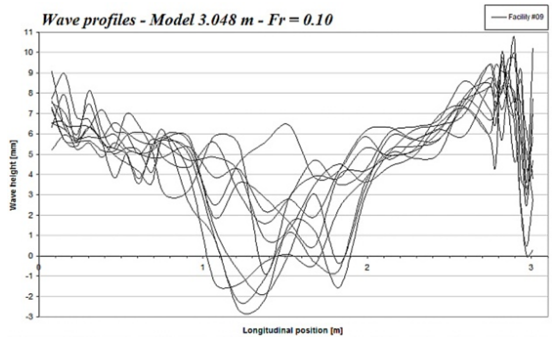


Figure 28 Small model wave profiles for $Fr = 0.10$

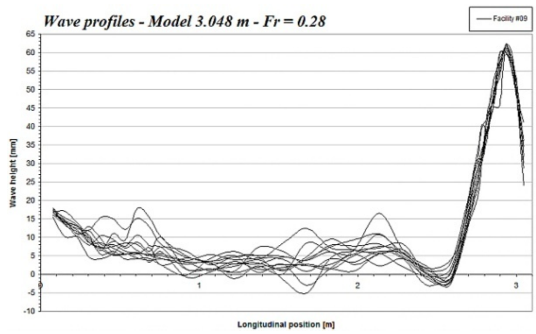


Figure 29 Small model wave profiles for $Fr = 0.28$

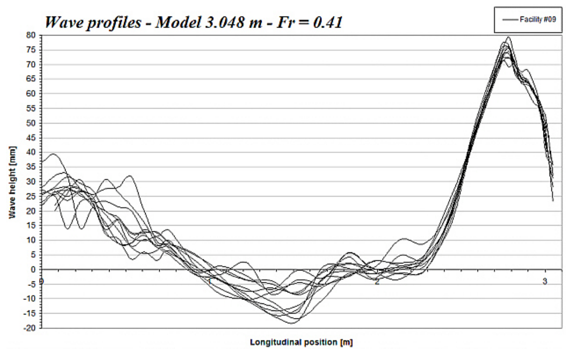


Figure 30 Small model wave profiles for $Fr = 0.41$

## 2.11 Wave Elevations

Only 4 facilities submitted wave elevation data for the large model and 3 facilities for the small one. Nevertheless in both cases there was 1 facility that sent almost zero data values, which have not been taken into account for the analysis. So, there were only 3 valid facilities for the large model and 2 valid facilities for the small one.

For the small model and for the lower velocity of the large model the dispersion of the data and the uncertainties were high and not conclusive, so they were not included in these proceedings. Some spurious values have also not been taken into account in the analysis.

For each model and each Froude number, the following data is presented:

- For each facility, the wave elevations, in mm, in 100 equidistant longitudinal positions along the hull length $(\zeta)_i$.

For the large model and for Froude numbers 0.28 and 0.41, the following data is also presented:

- For each facility, the uncertainties of the wave elevations in 100 equidistant longitudinal positions along the hull length $(U(\zeta))_i$ compared with their mean value $\overline{U(\zeta)}$, expressed in percentage of $(\zeta)_i$.
- For each facility, the uncertainties of the wave elevations in 100 equidistant longitudinal positions along the hull length $(U_{FB}(\zeta))_i$ compared with their mean value $\overline{U_{FB}(\zeta)}$, expressed in percentage of $(\zeta)_i$.
- For each facility, the total uncertainties of the wave elevations in 100 equidistant longitudinal positions along the hull length $(U_t(\zeta))_i$ compared with their mean value $\overline{U_t(\zeta)}$, expressed in percentage of $(\zeta)_i$.

In the below figures the longitudinal position is measured along the hull from section 0 (positive to bow).

<u>5.720 meters length model.</u>

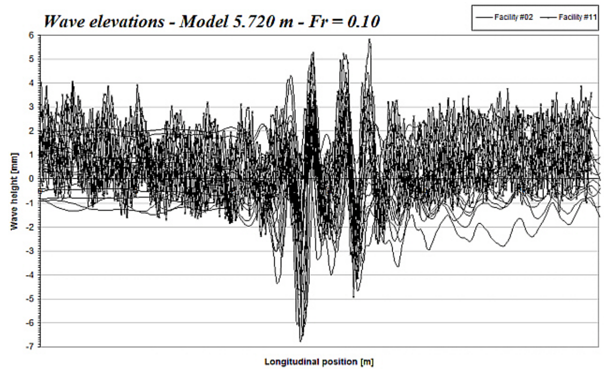


Figure 31 Large model wave elevations for *Fr* = 0.10

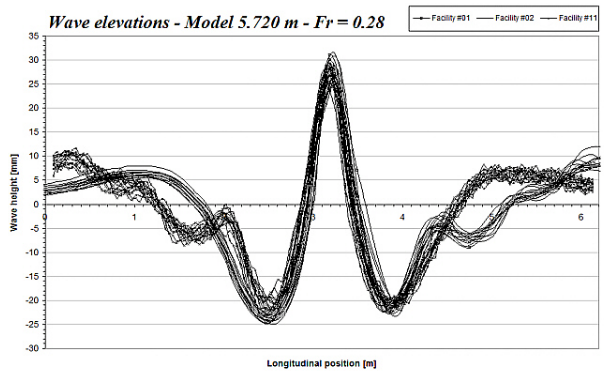


Figure 32 Large model wave elevations for *Fr* = 0.28

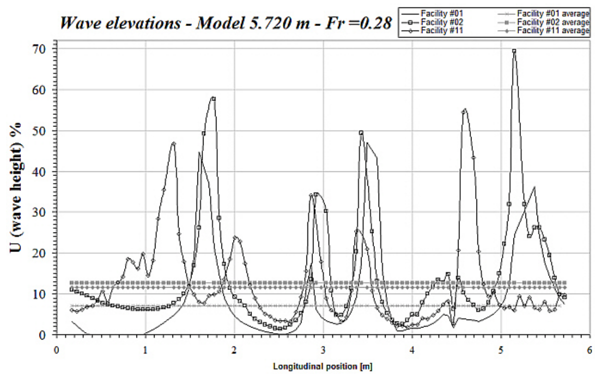


Figure 33 Large model uncertainties for wave elevation for *Fr* = 0.28

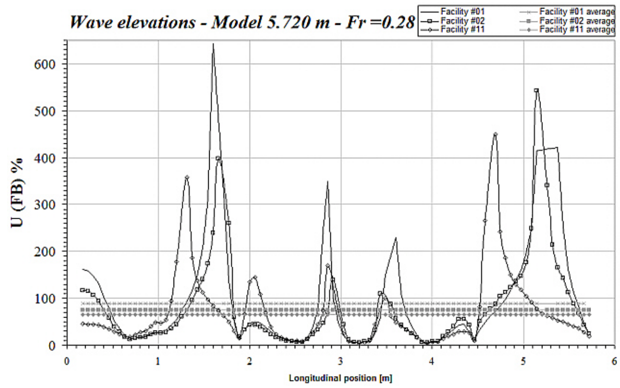


Figure 34 Large model uncertainties for wave elevation facility bias for *Fr* = 0.28

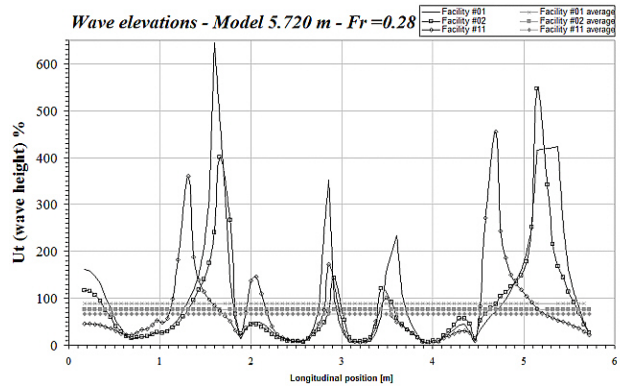


Figure 35 Large model total uncertainties for wave elevation for *Fr* = 0.28

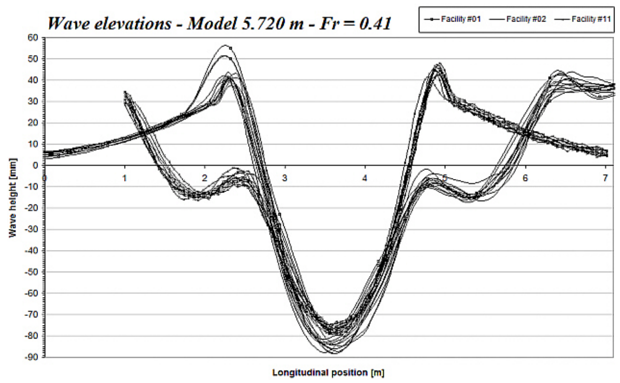


Figure 36 Large model wave elevations for *Fr* = 0.41

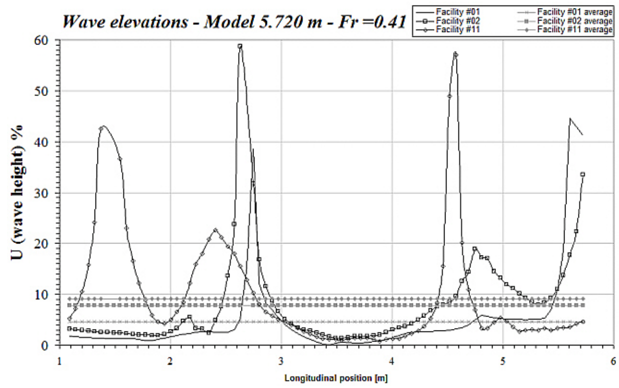


Figure 37 Large model uncertainties for wave elevation for *Fr* = 0.41

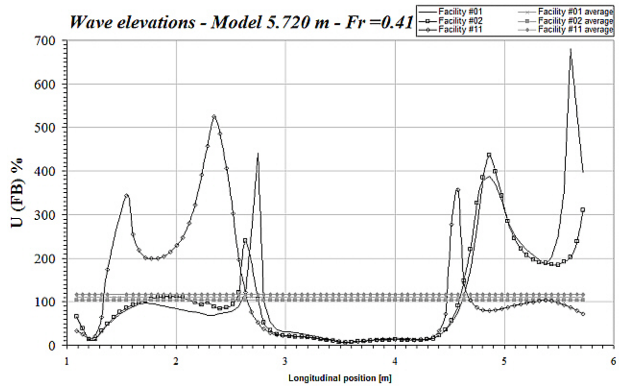


Figure 38 Large model uncertainties for wave elevation facility bias for Fr=0.41

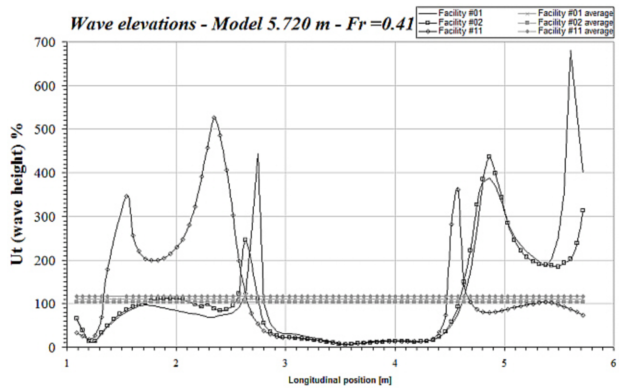


Figure 39 Large model total uncertainties for wave elevation for Fr = 0.41

3.048 meters length model.

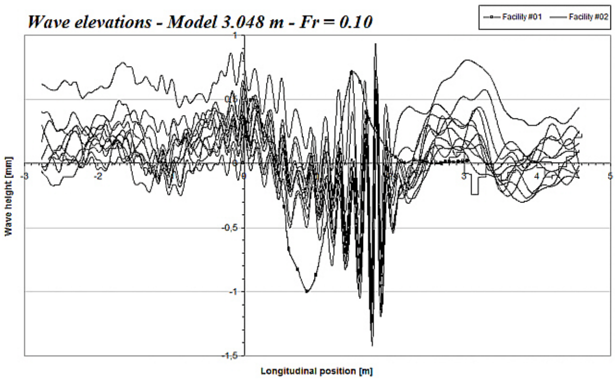


Figure 40 Small model wave elevations for Fr = 0.10

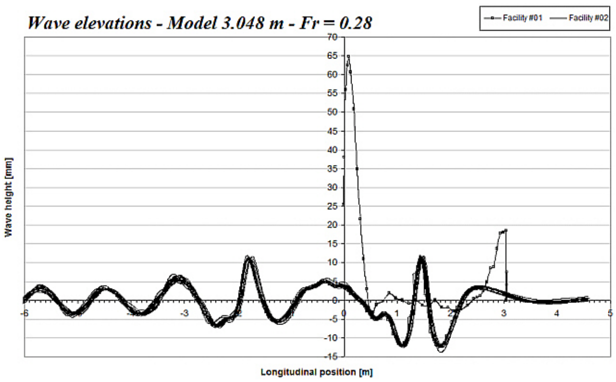


Figure 41 Small model wave elevations for Fr = 0.28

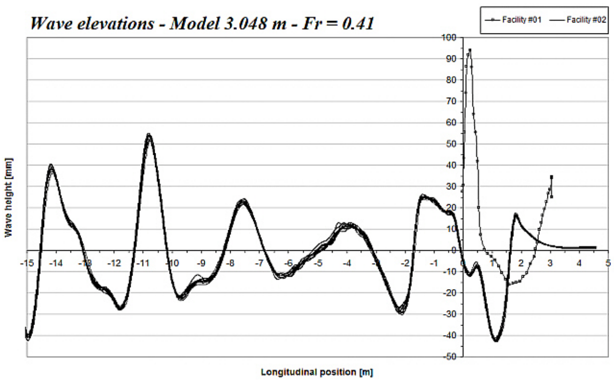


Figure 42 Small model wave elevations for Fr = 0.41

## 2.12 Summary of Results

The following is a summary of the data obtained:

- 41 facilities from 20 countries have enrolled the Facility Bias Worldwide Campaign.

- The tests for the 5.720 m length model have finished on time.

  – 21 facilities have tested the model and there is available data, in the correct format, for 12 of them.
  – All the facilities sent resistance, sinkage and trim data.
  – Only 2 facilities sent wave profile data, in the correct format.
  – Only 3 facilities sent wave elevation data, in the correct format.

- The tests for the 3.048 m length model have not finished due to facility delays, and schedule problems. The model was also lost during the transportation from Brazil to the United Kingdom.
  – 15 facilities have tested the model and there is available data, in the correct format, for 10 of them.
  – All the facilities sent resistance data.
  – Only 6 facilities sent sinkage and trim data.
  – Only 1 facility sent wave profile data, in the correct format.
  – Only 2 facilities sent wave elevation data, in the correct format.

- Some errors in the previous uncertainty analysis were detected in some submitted files, which were withdrawn and not used in the analysis.

- The median total uncertainty for the 5.720 m length model is:

  – 3.6% of the resistance coefficient for Froude number 0.10.
  – 1.4% of the resistance coefficient for Froude number 0.28.
  – 1.8% of the resistance coefficient for Froude number 0.41.
  – 30.8% of the sinkage value for Froude number 0.10.
  – 8.2% of the sinkage value for Froude

number 0.28.

- 6.8% of the sinkage value for Froude number 0.41.
- 41.9% of the trim value for Froude number 0.10.
- 12.6% of the trim value for Froude number 0.28.
- 17.0% of the trim value for Froude number 0.41.
- Lower than 89.2% of the wave elevation value for Froude number 0.28.
- 117.1% of the wave elevation value for Froude number 0.41.

- The median total uncertainty for the 3.048 m length model is:
  - 10.1% of the resistance coefficient for Froude number 0.10.
  - 3.7% of the resistance coefficient for Froude number 0.28.
  - 6.4% of the resistance coefficient for Froude number 0.41.
  - 40.2% of the sinkage value for Froude number 0.28.
  - 28.6% of the sinkage value for Froude number 0.41.
  - 52.0% of the trim value for Froude number 0.28.
  - 24.3% of the trim value for Froude number 0.41.

- There is not enough data to analyze sinkage and trim uncertainties for the 3.048 m length model at Froude number 0.10.

- There is not enough data to analyze wave profile uncertainties.

- It is only possible to analyze wave elevation uncertainties for Froude numbers 0.28 and 0.41 with the 5.720 m length model.

- The dispersion of the data is high in the submitted wave profiles and wave elevations. It was necessary to correct phases and signs in the wave elevation data submitted.

- The uncertainties are higher for the low Froude number because the measured values are quite small.

- Some facilities have sent a copy of the ITTC spreadsheets for uncertainty analysis, and many of them used the values of the example instead of changing the values in correspondence with their own equipment.

- A mistake was detected in the spreadsheet for resistance uncertainty analysis: the conversion of the total resistance coefficient to 15ºC is not done as described in procedure 7.5-02-02-02. Nevertheless, this fact has no important relevance in the results.

- 30% of the received files were in a different format than specified in the instructions for data submission and that made the analysis difficult.

- The Uncertainty Analysis Committee proposed a new analysis method for uncertainty analysis, testing two geosims of the same model in each facility. As only one model has been tested in each facility it was not possible to use that method in this worldwide campaign.

## 2.13 Conclusions

The tests for the large model are now complete whereas there are still three test programs still to be conducted with the small model. However, there still remains a proportion of tests for which results have not been recieved. As this data arrives it will be added to the existing database. A task still to be done is to make this dataset available to the wider ITTC community.

Overall, the whole process has been of considerable value if nothing more to remind the ITTC perhaps of the significant challenges still associated with conducting high quality experimentation across multiple facilities with a worldwide geographical dispersion. The results should challenge many facilities as to whether they actually are delivering experimental uncertainty to the levels they are stating in resistance test documentation. Key areas are still associated with model quality and set-up, sinkage and trim, as well as the method by which resistance data is analysed.

## 3. TRENDS IN EXPERIMENTAL FLUID DYNAMICS (EFD)

### 3.1 Introduction

This chapter summarises the state of art in EFD related research in naval hydrodynamics since the last ITTC conference in 2008, consisting of 2 parts: 1) new and advanced EFD techniques covering PIV/LDV Flow Measurement Technologies, High Speed Video and Imaging Technologies, Wave Measurement Technologies, Free Running Model/Ship Technologies, Other Model Testing Technologies, Experiments in Circulation Water Channel, Full Scale Measurements, 2) EFD in drag reduction.

### 3.2 New and Advanced EFD Techniques

The most notable and organized activity investigating advanced experimental techniques would be the HTA (Hydro-Testing Alliance) project of EU. The HTA is one of the European Commission's Sixth Framework Programmes starting in September 2006 and running for 5 years. Major participants to the HTA are MARIN, SIREHNA, HSVA, SSPA, INSEAN, MARINTEK, FORCE Technology, QinetiQ, VTT, CTO and universities. The purpose of HTA is to develop a formal and lasting structure to coordinate the definition and introduction of novel measurement observation and analysis technologies for hydrodynamic model testing. The HTA consists of nine JRPs (Joint Research Program) as follows;

- JRP1 PIV operation in hydrodynamic experimental facilities
- JRP2 Flow data analysis and visualization
- JRP3 3-D wave field measurements
- JRP4 POD/Dynamic forces
- JRP5 Wireless data transmission
- JRP6 High speed video recording and analysis
- JRP7 Intelligent materials and production methods
- JRP8 Wetted surface
- JRP9 Free running model technologies.

The activities and research results can be found at (http://hta-noe.eu/ ). As part of the HTA activities, the first conference on advanced model measurement technologies, called AMT'09 was initiated with 34 presentations on contemporary experimentation topics such as PIV, high speed video methods and flow/wave measurement techniques.

The Japanese Ship Propulsion Committee (JSPC) organized a symposium on the present status and perspective of tank testing techniques, see JSNAOE (2010). The contents include not only trends in measurement techniques including uncertainty analysis (Nishio, 2010), but also the domestic research project on Advanced Tank Testing Technique (JASNAOE AEFD Committee, 2010), and strategy in EEDI and tank tests (Sasaki, 2010). Research on tank tests in Korea (Chun et al., 2010) and China (Ma, 2010) are also presented in the symposium.

PIV/LDV Flow Measurement Technologies

For the last decade or so, remarkable progress has been made on the hydrodynamic flow measurement techniques used in water tank/tunnel/channel, wave tank and wind tunnel due to noticeable improvements of optical and computer technologies. Two laser-optical flow velocity measurement techniques are widely used in towing tanks: a highly accurate and highly time-resolved point technique called Laser Doppler Velocimetry

(LDV) and a whole field Particle Image Velocimetry (PIV) with the ability to measure instantaneous planar sections of the flow field with a moderate temporal resolution. Both LDV and PIV can be configured to measure all three components of the velocity vector simultaneously. While LDV has become a very mature technique, PIV is still rapidly evolving both in terms of the hardware (cameras and lasers) and the software (algorithms used to determine velocities and graphical user interface to make the handling of a PIV system as user friendly as possible). The detailed information about the spatial structure and dynamics of a flow obtained with PIV/LDV can be used for validation of CFD results and to improve the design of ships and propulsion systems. However, there are still other challenges left for practical use in towing tanks, for example in time-resolved PIV, measurements near the ship hull or around free-sailing, manoeuvring ships, require a low weight and rapid data feedback for high quality measurements, see various applications in Hallmann et al. (2009), Gjelstrup (2009), Liarokapis et al. (2009a), Grizzi et al. (2009), Anschau et al. (2009), Bouvy et al. (2009), Borleteau et al. (2009).

The HTA JRP-1 research activities on “PIV operation in hydrodynamic facilities” conducted by a large group is reported by Fréchou et al. (2009) covering wave run-up, wake flows in powering/manoeuvring conditions, wake flows in cavitation tunnel or shallow water basin, nuclei sizing in cavitation tunnel, and a flat plate benchmarking program.

Measuring the turbulent kinetic energy dissipation rate in an enclosed turbulence chamber that produces zero-mean flow is an experimental challenge (de Jong et al., 2009). The capability of Interfacial PIV was validated against Particle Image Distortion using synthetic image pairs generated from a DNS velocity field over a sinusoidal bed (Nguyen et al., 2010). Wake measurement data provide physical insight into the factors to be considered for the propeller operation of VLCC in damaged conditions (Yang et al., 2009). Flow characteristics of the hull wake behind a container ship model were investigated under different loading conditions by PIV (Lee et al., 2009)

High Speed Video and Imaging Technologies

The procedure for cavitation observation has basically not changed over the last hundred years, consisting in a stroboscopic freeze of the propeller motion at a specific angular position. Thus, the temporal dynamics of cavitation cannot be understood and the phenomenological correlations with erosion, pressure or noise effects cannot be assessed with sufficient insight (Pereira et al., 2009). The high speed video (HSV) recording technique has been demonstrated to be an invaluable tool to address this limitation, since it offers a time-resolved recording of the cavitation pattern during the propeller revolution, see Grekula and Bark (2009), Savio et al. (2009) and for ILIDS (Interferometric Laser Imaging Droplet Sizing) technology, see Lacagnina et al. (2009).

Since the accurate wetted area estimation is important in high speed ships such as planing craft, Allenström et al. (2009) suggested image processing and computer vision techniques to replace most of the manual work to detect the waterline, a live-wire technique being found to be effective.

Wave Measurement Technologies

A number of advanced techniques to measure the three-dimensional wave field generated by a moving ship are used: a scanning Light Detection and Ranging, (LiDAR) system, a laser sheet-optical quantitative visualization (QViz) system, a Nortek Acoustic Wave and Current (AWAC) profiler, ultrasonic range finders, finger probes, and capacitance wires. Fu and Fullerton (2009) and Drazen et al. (2010) describes recent efforts and compares results from the LiDAR, QViz, AWAC, and ultrasonic range finder systems for the measurement of waves generated by the tow tank wave makers and by a large generic transom stern model. A new generation of acoustic wave probes is developed to function well at high towing carriage speeds and high wave encounter frequencies with the support of the HTA project (Bouvy et al., 2009). Cobelli et al. (2009) introduced an optical profilometric technique that allows for single-shot global measurement of free-surface deformations.

The direct measurement of wave resistance from the measurement of ship-wave height on a patch of surface water near the hull with optical instrumentation was done by Çalişal et al. (2009).

Image measurement techniques have gained more attention as Sanada and Nagaya (2010) summarized the recent status: e.g., Takayama et al. (2008), Sanada et al. (2010), and Tanimoto et al. (2010) for ship wave measurements by using the RLD method; and Nagaya et al. (2010) for flow field measurements by using stereo PIV.

Free Running Model/Ship Technologies

La Gala and Gammaldi (2009) introduced a Wireless Inertial Motion Unit (WIMU) for Motion Analysis in Towing Tank Experiments and in general for free model tests. Kennedy et al. (2009) reported an advanced free running model technology from model to full-scale. Kimber et al. (2009) studied on real time wireless data and control communication of free running submerged scale models of submarines.

Other Model Testing Technologies

A direct measurement disk sensor system of wall shear stress on low speed boundary layers under a flat surface was developed by Harleman et al. (2010) with a capability of the disk surface being interchanged to study the influence of wall roughness or coating. Liarokapis et al. (2009b) developed a seven-hole pitot tube arrangement for measuring high quality wake flows in the towing tank.

Other experiments for new designs are: Tanigami et al. (2008) for a full ship's propulsive performance in shallow water; Miyoshi et al. (2008) for hydrodynamic characteristic of the rudder with anode protection; Arai et al. (2009) for stall delay by wavy leading edges of a 3-dimensional wing.

Works related to development of new experimental methods were reported: e.g., Wakahara et al. (2008a, 2008b) for development of an affix-type multipoint pressure sensor by use of FBG; and Kawashima et al. (2008, 2009) for accurate measurement of flat plate resistance by means of parallel towing. On the other hand, Takimoto et al. (2009) investigated a single view distance measurement system for a floating body on the free surface, and Rheem and Katsura (2010) sea surface measurement by a microwave pulse Doppler radar.

Experimental investigation of wind pressure characteristics were also continued, e.g., Momoki (2008), Fujiwara et al. (2009a), Fujiwara and Sasaki (2009), and Fujiwara et al. (2009b).

Experiments in Circulation Water Channels

Recent status of measurements in circulation water channels was summarized by Kawashima and Hashizume (2010), including future prognosis of the development. For example, Mori (2010) presented a measurement technique in the flow noise simulator equipped in a large circulation water channel. Also, Suzuki et al. (2008) and Kawashima and Miyoshi (2008) presented progresses in wave field measurements and flow field measurements, respectively, in their circulation water channels.

Full Scale Measurements

Techniques on full scale measurements have gained more attention as summarized by Tanaka and Masuda (2010). For example, Herai, et al. (2010) verified effectiveness of a newly developed energy saving appendage in full scale measurements. The full scale measurements are also focused on evaluation of ship propulsion performance in actual sea environments: e.g., Minoura et al. (2008) and Minoura (2008) for ship speed loss and other related phenomena by analyzing onboard measurement data; Sasaki et al. (2008) for speed loss of large container ships operating in a sea way; Shoji et al. (2008) for analysis method for ship performance by using an abstract logbook; Yamamoto et al. (2010) for full scale measurement data of large container ships on the effect of the hydroelastic response; and Kano et al. (2010) for measured wave spectra by coastal vessels and estimated added resistance in waves.

## 3.3 Experimental Study on Drag Reduction

In the past three years in Japan, air lubrication to reduce ship frictional drag was reported in quite a few papers. The study was initially carried out as a domestic Japan research project, which involved development of the devices, model scale experiments to evaluate the effectiveness of the devices and to investigate the related flow physics, development of numerical models for simulation, and finally full scale measurements. For example, Makino et al. (2008) reported the effect of surface curvature on skin friction reduction by air bubbles, Hinatsu et al. (2008) and Kodama et al. (2008) presented a full-scale air lubrication experiment using a large cement carrier whose overall length is 126.6m, Murakami et al. (2008) performed numerical simulation of flow around a full scale ship equipped with bubble generators, and Kawanami and Hinatsu (2010) performed bubble flow visualization around a ship equipped with an air lubrication system. The full scale test results summary is as follows: a maximum of 11% reduction in ship resistance, power savings of 4 % and 6% for full and ballast loads, respectively, and 40% reduction in local skin friction on the hull bottom.

The activity to develop an air lubrication system in Japan was taken over by a private shipyard. Recently, Mitsubishi Heavy Industries built a module carrier *YAMATAI* and *YAMATO* whose overall length is 162m, for which an air lubrication system was equipped. Some reports to describe the work were presented: e.g., Takano et al. (2010) presented the overall system and evaluation through the full scale ship test, Kawakita (2010) investigated a related study, i.e., prediction accuracy of full scale wake distribution required by propeller design, and Takano et al. (2010) presented the behaviour of injected air on the ship bottom and its influence on the propeller. The full scale test results summary is that a maximum of 12% power saving was achieved for 7mm thickness in air layer and more effective benefits can be achieved with more air flux.

The air layer/cavity drag reduction activities in Europe during the SMOOTH (Sustainable Methods for Optimal design and Operation of ships with air-lubricaTed Hulls) project have been compiled and reported in the international conference on ship drag reduction (SMOOTH-Ships) on May 20-21, 2010 in Istanbul, Turkey. The SMOOTH project, a sequel to the PELS projects, was carried out by many organizations including classification bodies, shipyards, coating and shipping industry, universities and research institutes in Europe from 2006 to 2010. In his keynote speech, Thill (2010) reported a contradicting view compared with the Japanese's promising views, showing that the micro bubble lubricated test ship Till Deymann showed hardly any improvement in terms of power saving and even a negative net energy saving by considering the air compressor power input. Ceccio et al. (2010) gave a cost-benefit analysis for air layer drag reduction, emphasizing the importance of persistence length of the air layer and the draft concerns in order to accomplish a net energy saving in consideration of air pumping cost. A few studies were directed toward the investigation on the relation between the condition of air layer and the surface condition, i.e., the type of coating (Allenström and Leer-Andersen 2010; Foeth et al., 2010; Insel et al., 2010).

In Russia, research activities on drag reduction by air lubrication and air cavity have been widely known with many full scale ship applications. The state of the art of the air lubrication technologies together with current research activities in Russia can be seen in Sverchkov(2010).

Reduction of frictional drag and related subjects were also studied in various approaches. A notable compilation of the most recent experimental as well as numerical progress can be found in the proceedings of the EDRFCM 2010 (European Drag Reduction and Flow Control Meeting). Among various strategies are plasma control and using a plasma actuator caught the attention for aerodynamic applications (Whalley and Choi, 2010; Berendt et al., 2010). Total drag reduction capabilities of outer-layer vertical blades, a vertical LEBU (Largy-Eddy BreakUp device), was clarified by Lee et al. (2010).

Kulik et al. (2010) reported that a new type of "stiff" compliant coating with enough endurance for real application led to drag reduction in a fully turbulent boundary layer. Recent trends in journal publications in the field of frictional drag reduction are mainly focused upon the use of polymer/surfactant injection and surface morphology using super-hydrophobic surfaces. For bubble and gas injection, Ceccio (2010) gave an extensive review. Notable progress in the identification of drag reduction mechanisms in the case of polymer injection are found in Somandepalli et al. (2010) with quantification of concentration flux using PIV/PLIF investigation and Cai et al. (2009) to combine POD (Proper Orthogonal Decomposition) with PIV measurements. The consideration of super-hydrophobic coating is associated with the recent development of microfabrication technologies. There are frequently found controversies regarding the drag reduction efficiency of superhydrophobic surfaces in submerged condition; Daniello et al. (2009) stated effectiveness of such coatings for even the totally turbulent regime, whilst Su et al. (2009) demonstrated that the microbubbles at the superhydrophobic surface increased the friction. The assessment of frictional drag reduction could be affected significantly by the choice of a specific experimental method. It is worth mentioning that the comparative evaluation of skin friction reduction capability, which was performed by the Ceccio group at the University of Michigan, gave no evidence that any of the currently reported skin-friction reducing superhydrophobic coatings really leads to skin friction reduction.

In Japan, Kawashima et al. (2010) reported an experimental investigation of the frictional drag reduction by using polymer released from the painted surfaces. Summary of the recent studies on fictional drag reduction of the ships along with theoretical review of model scale to full scale friction lines are given by Katsui and Kawakita (2010).

### 3.4 Conclusions

Recent advanced technologies in EFD in naval hydrodynamics are well reported in AMT'09 that is the product of the HTA (Hydro-Testing Alliance) project of EU that is the most notable and organized activity related to advanced experimental techniques. The HTA project covers PIV/LDV, high speed video and imaging technologies, wave measurement, free running model/ship measurements etc. For the last decade or so, remarkable progress has been made on hydrodynamic flow measurement techniques due to noticeable improvements of optical and computer technologies. The detailed information about the spatial structure and dynamics of a flow obtained with PIV/LDA can be used for validation of CFD results and to improve the design of ships and propulsion systems. However, there are still other challenges left for practical use in towing tanks, for example time-resolved PIV, measuring near the ship hull or around free-sailing models.

For the last three years, drag reduction technology based on an air lubrication system using micro bubbles showed promising results from full scale sea trial tests in Japan: a maximum of 11% reduction in ship resistance, power savings of 4 % and 6% for full and ballast loads with a large cement carrier(LOA=126. 6m), and also a maximum of 12% power saving with a module carrier *YAMATAI* and *YAMATO* whose overall length is 162m. This result encourages the commercial use of micro bubble lubrication technologies for drag reduction and powering savings. However, full scale tests in Europe indicated a contradicting view, showing that the micro bubble lubricated test ship Till Deymann showed hardly any improvement in terms of power saving and even a negative net energy savings by considering the air compressor power input. This means that there seems to be a need for a good design combination between the ship and the air lubricant system in order to achieve drag reduction benefits. The air layer/cavity drag reduction activities in Europe during the SMOOTH project have been reported in the international conference on ship drag reduction (SMOOTH-Ships).

Reduction of frictional drag and related subjects were also studied in the various literature. A notable compilation of the most recent experimental as well as numerical progress can be found in the proceedings of the EDRFCM 2010 (European Drag Reduction and Flow Control Meeting).

The RC suggests the following tasks:

(i) Continue to review trends and new developments in experimental techniques for towing tanks, especially on unsteady flows and dynamic free surface phenomena.

(ii) Monitor new developments in test facilities and model manufacturing devices.

(iii) Continue to monitor new developments in Verification and Validation methodology and procedures.

## 4. TRENDS IN COMPUTATIONAL FLUID DYNAMICS

### 4.1 Overview

The major trend in the use of computational fluid dynamics in ship hydrodynamics is the wider adoption of the computational methods outside the community of method developers and the application of the methods for a wider spectrum of problems. This section reviews the recent activities in the field since the 25th ITTC conference. The primary focus of the review is on the prediction and scaling of resistance according to the terms of reference of the committee.

The dominating approach for modelling turbulence is Reynolds Averaged Navier-Stokes (RANS), which is also reflected in the recent papers reviewed for this report. However, a few papers presenting studies based on large eddy simulation (LES) or the variety of detached eddy simulation (DES) modelling have appeared as well (see e.g. Alin et al., 2008; Carrica et al., 2010; Ismail et al., 2010; Yang and Stern, 2009). Furthermore, Fureby (2008) has presented a review of the use of LES in ship hydrodynamics. In the majority of the reviewed papers the free surface is modelled using a surface capturing approach, either a volume of fluid (VOF) or level-set approach. Surface tracking based studies are now clearly in a minority. The grids used are mostly of unstructured or structured multi-block or overset type. A quite recent development which seems to be gaining popularity is the application of Cartesian grids and immersed boundary method for ship hydrodynamics problems (see e.g. O'Shea et al., 2008; Yang and Stern, 2009). However, the focus in these studies has been on the modelling of the wave field rather than capturing the viscous boundary layer and wake.

## 4.2 New CFD Applications

The increasing computational power particularly in terms of available parallel computing resources has made it possible to perform more detailed studies, but also to deal with more complex problems. As the experience on the use of CFD methods is increasing and as the simulation methods are becoming more mature and flexible the range of CFD applications is expanding further. Nakashima et al. (2009) have presented an analysis on wind drag reduction of accommodation house with square corner cut and step shaped geometries. Kimura et al. (2010) have analysed the hydrodynamic forces acting on a ship in shallow water. Kodama et al. (2010) have used CFD for the analysis of wall effect reduction using flow liners in a cavitation tunnel. Xing et al. (2008) have studied the simulation of resistance and propulsion curves with a single run using CFD and taking into account sinkage and trim. The effectiveness of various energy saving devices has been studied with the aid of CFD in several papers (see e.g. Thornhill et al., 2008; Hafermann et al., 2010; Chen et al., 2010; Hooijmans et al., 2010; Makino and Masuda, 2008).

The prediction of added resistance is traditionally based on potential flow methods. Further development of the potential flow methods is discussed in several recent papers (see e.g. Joncquez et al., 2008; Kim et al., 2010; Liu et al., 2011) However, the application of viscous field methods has also been demonstrated recently. Carrica et al. (2010) have presented results for the added resistance with DES, whereas Deng et al. (2010), Orihara (2010), and Simonsen et al. (2008) have used RANS for the prediction of added resistance. On a related topic, Ratcliffe et al. (2008) have performed a combined experimental and numerical study on wave induced forces and wave diffraction.

The developments have also made it possible to study ship flows with extremely high resolutions. Carrica et al. (2010) have studied the forward diffraction and motions in head waves with 60-115x$10^6$ grid points. Drazen et al. (2010) present a computational study on wave generation of a transom model with a grid resolution of up to $10^9$ cells. The simulations with such resolutions are able to capture very small flow and free surface details. Carrica et al. (2010) show that the increased resolution results in a dramatically improved prediction in terms of these features compared to resolution of a few million points. However, the integral quantities are only slightly improved.

## 4.3 Full Scale Predictions and Scaling

There is a clear trend of using CFD increasingly to make full scale predictions and to study the process and the assumptions related to the scaling of the results from the model to full scale results. Huang et al. (2010) have used CFD to predict full-scale hull resistance. Also, Kaneko et al. (2008) presented effect of turbulence models on flow simulation around a full-scale ship in a view of ship design. The availability of new full scale data has also made it possible to assess the validity of the computational predictions. For example, Wyatt et al. (2008) have compared numerical predictions of the topology of a transom stern wave with full scale measurements. Moraga et al. (2008) have developed an air entrainment model and have compared the prediction of air entrainment with full scale measurement data. From the point of view of full scale resistance the most interesting recent activities are related to scaling and the modelling of the flow close to the hull in full scale. These are discussed in the following sub sections.

Scaling. Computational simulations are not restricted to a specific scale and the same methods can be used with certain limitations to obtain predictions in model and full scale. Compared to model tests CFD provides more detailed information of the division of the resistance into various components. Furthermore, with CFD it is straightforward to neglect specific physical effects, such as wave making, and in this fashion isolate particular resistance components. In several of the recent papers CFD results have been used to study the fundamental principles and assumptions of the empirical extrapolation methods.

A fundamental aspect in the scaling of the resistance from model to full scale is the friction line used for the extrapolation and this is discussed in Section 5.2.

Tzabiras and Kontogiannis (2010) have studied three different bulbous bows for a low $c_b$ hull form with computations in model and full scale. Simulations have been complemented with dedicated model tests. The results have been used to test the assumptions of extrapolation methods, such as the assumption of constant form factor, and to study the differences between the computational and model test based full scale predictions. It has been concluded that the choice of the friction line and the wetted surface can have a significant influence on the predictions.

van der Ploeg et al. (2008) compare the predictions of two computational methods in terms of resistance, wave, wake, trim and sinkage. The scale effects on the wave and wake field are considered and the computed scale effects of the resistance are compared with extrapolation methods. The scale effects of the viscous resistance component are studied by using double body simulations and by comparing the form factors in model and full scale with various friction lines. Again the choice of the friction line has a significant influence on the variability of the form factor. The scale effects of the wave making resistance are studied by comparing predictions with and without free surface. Both methods predict a scale effect on the wave making resistance, which is in contrast with the usual assumption made in extrapolation methods.

Raven et al. (2008) have presented an extensive study on the scaling and scale effects of viscous and wave making resistance using CFD. These are discussed in light of the assumptions made in the traditional extrapolation of full-scale resistance. The study demonstrates that model test based predictions can be supported with CFD methods in order to improve their reliability. The study has also revealed some scale effects on the ship wave making and on the associated resistance component. The paper also presents an interesting comparison between a directly computed full-scale resistance prediction and a prediction based on traditional extrapolation of the computed model-scale result. The comparison reveals that the correlation allowance in the extrapolation compensates for the difference between the computed and the extrapolated full-scale resistance.

Choi et al. (2009) and Choi et al. (2010) have presented a ship-speed prediction approach which relies on computational simulations and model tests. The approach is based on resistance and self-propulsion simulations in model-scale. The full-scale prediction is based on the ITTC-78 method. A CFD-model test correlation coefficient is used in the scaling to take into account the difference between the model test and computational resistance. The prediction of the ship-speed performance is demonstrated for a range of ships and dedicated model tests have been used to assess the reliability of the predictions.

Near-wall treatment. The numerical modelling of the flow close to a ship hull in full scale differs from the modelling in the model scale in two respects. Accurate modelling of the full scale flow requires that the influence of the surface roughness is taken into account. At the same time, the smaller relative thickness of the boundary layer in full scale may lead to some numerical issues, which are not present in model scale. Near-wall turbulence models require grid densities which may still be unaffordable in full-scale simulations. Furthermore, the resulting cell aspect ratios close to the hull may lead to convergence problems.

To avoid the issues related to near-wall models Bhushan et al. (2009) have studied the applicability of wall-functions for ship flows. Two and multilayer wall-functions with roughness effects have been used. The results are compared to available experimental data (resistance, sinkage and trim in model-scale; boundary layer and wake in full-scale) and to previous simulation data with near-wall modelling. In the full-scale simulations the frictional resistance predictions are shown to be in good agreement with the ITTC 57 line. Furthermore, the change of frictional resistance with rough-wall simulations agrees with the ITTC correlation allowance. However, some issues related to the applied wall-functions, such as possible sensitivity to the distance of the first grid point from the wall, are revealed. Similar issues have been identified by Eça and Hoekstra (2011). They have studied numerical aspects of including wall roughness effects in k-ω SST model and have compared two near-wall approaches and a wall-function approach with a flat plate flow for Reynolds numbers ranging from $10^7$ to $10^9$. They conclude that all three approaches are able to simulate sand-grain roughness effects up to very high Reynolds numbers. However, the numerical behaviour of the approaches is very different, and the distance of the first grid point from the wall may have a significant influence on the results both with smooth as well as with rough walls.

Eça and Hoekstra (2010) have studied the influence of surface roughness on four different flow quantities: the friction, pressure and viscous resistance coefficients and the mean wake fraction. The simulation cases are a tanker, a container ship and a car carrier. The cases cover model and full scale flows and sand-grain roughness heights from 0 up to 300 μm. In addition to analysing the influence of the roughness height on the selected flow quantities, the roughness effect on the resistance is compared with four empirical correlations. The obtained results indicate that the hull geometry has an influence on the roughness effects and that a single parameter, such as the roughness height or the related Reynolds number, is not sufficient to parameterise the roughness effects. The best overall comparison between the computed results and the empirical correlations has been obtained with the Townsin et al. (1984) formula.

### 4.4 Uncertainties in CFD

The error in the solutions obtained with computational fluid dynamics, i.e. the difference between the numerical and the true value, can be considered to consist of two contributions: the modelling error related to the continuous mathematical model and the numerical error related to the discretised solution of the mathematical model. Depending on the interpretation of the concept of the model an additional contribution related to the input parameters of the computational model may be defined. Because the actual error is generally not known, it has to be taken into account in the simulation results as an uncertainty. Estimation of the modelling and possible parameter uncertainties is a validation activity, whereas estimation of the numerical uncertainty is a verification activity.

The main source of modelling uncertainty in practical simulations is the turbulence modelling. Ultimately the magnitude of this uncertainty depends on the suitability of the turbulence model for the case at hand. The choice of the turbulence model used is primarily done by the code user. Different turbulence models may give largely varying predictions for the same case, and the difference between the various models may be strongly influenced by the Reynolds number (see e.g. Eça and Hoekstra, 2008; 2009a). The quantification of this uncertainty is non-trivial and has to be based on prior validation studies of similar cases.

Another significant source contributing either to the modelling or parameter uncertainty depending on the model concept is the choice of the boundary conditions and the domain size. Inexact boundary conditions or boundaries too close to the ship may lead to a serious error in the predictions particularly in terms of the pressure resistance (see e.g. Eça and Hoekstra, 2009a).

The uncertainty related to the numerical solution includes contributions from the spatial and temporal discretisations, incomplete convergence and computer round-off. Research on the estimation of the numerical uncertainty has mainly focused on the discretisation related components, as it is mostly considered to be the dominating component. The round-off and convergence related components can in most cases be made negligible by choosing a sufficient precision for floating point numbers and an adequate number of iterations respectively. For example, recent studies on the iterative error have shown that its contribution to the overall uncertainty is negligible, if the iterative error is two to three orders of magnitude below the discretisation uncertainty (see e.g. Eça and Hoekstra, 2009b).

The spatial and temporal discretisation related uncertainties depend on the selected discretisation schemes, discretisation resolutions, discretisation quality and variation of the solution in space and time. The uncertainty may also be affected by the approach used for the evaluation of the functionals from the solutions such as resistance (see e.g. Salas and Atkins, 2009). The spatial component of the uncertainty is always present, whereas the temporal component disappears in steady state problems. The methods used in practice are mainly based on nominally second order accurate discretisations. These lead to a compact computational stencil and sufficient accuracy with practical discretisation resolutions. VOF methods are usually based on special compressive schemes in order to accurately capture and maintain the discontinuity of the volume fraction function at the free surface.

With unstructured grids adoption of higher than second order schemes is problematic, whereas with structured or Cartesian grids higher order schemes are more common. Di Mascio et al. (2009) have compared the accuracy of the second-order ENO scheme and the third and fourth order Godunov schemes with a practical free surface ship flow. They state that the level of uncertainty obtained with the second-order scheme can be obtained with the higher-order schemes already with twice as large cell size in every direction. Yang and Stern (2009) have compared the wave field predictions for the Wigley hull obtained with two different methods: an overset RANS solver with the level-set equation discretised with a second-order scheme and a cartesian grid immersed boundary method with the level-set equation discretised using a fifth-order scheme. The cartesian grid based method resulted in a significantly better prediction of the wave field which was attributed among others to the higher-order of the discretisation.

The grid resolution has a significant influence on the numerical uncertainty. Different resistance components exhibit different behaviour with changing grid resolution and have often different levels of uncertainty (see e.g. Eça and Hoekstra, 2009a). Traditionally the choice of mesh resolution at different locations is made by the code user in the pre-processing stage based on experience or on data from previous simulation of the same case. The computational load can be reduced by refining the grid only in specific locations. Orych et al. (2010) have presented an adaptive overlapping grid technique to increase the surface sharpness and numerical accuracy of surface capturing methods. Wackers et al. (2010) have demonstrated the effectiveness of a developed grid adaption technique by a wake and a wave field study.

It is well known that quality of the grid may have a significant influence on the discretisation error. Systematic studies of the influence are, however, scarce. Ismail et al. (2010) have studied the influence of the grid quality on numerical error and the associated uncertainty by assessing the performance of various linear and non-linear convection schemes on orthogonal and non-orthogonal grids. They have performed a systematic study on the influence of three grid distortion metrics on four error metrics. Based on the results recommendations for grid distortion metrics are provided.

An additional uncertainty is related to the estimation of the numerical uncertainty. Xing and Stern (2010) have discussed the development of an improved uncertainty estimation method. The presented comparison shows large variation between the results obtained with different uncertainty estimation methods based on exactly the same simulation data.

As CFD is being more widely adopted in design and optimisation, the treatment of uncertainty in optimisation is going to have an increasing importance. In optimisation studies one should have confidence that the trends are correctly predicted and that the process does not result in a non-optimal solution because of the associated uncertainties. As an example, Tahara et al. (2008) have applied a systematic verification and validation methodology to demonstrate the validity of the simulation-based design framework.

As a final note on uncertainity, at the recent Gothenburg workshop on CFD, Larsson and Zou (2010) examined the 89 resistance predictions for all the test cases. Overall, the mean of errors when compared to the experimental data was similar to that found in the preceding workshop in Tokyo in 2005, eg -0.1% but that the standard deviation had significantly reduced. This database of information will provide an invaluable source of information on uncertainity. Figure 43 indicates that it is not now solely the mesh size that is driving the reduction in predictive uncertainty.

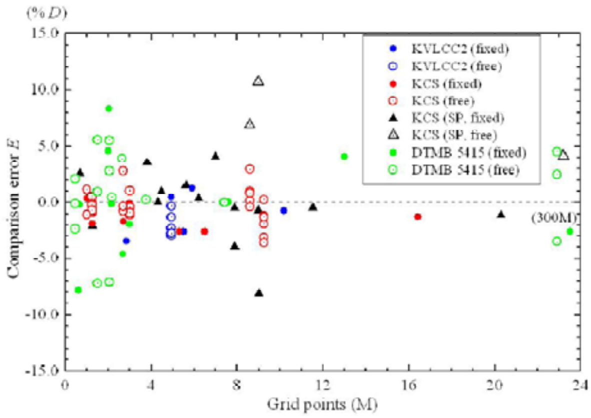


Figure 43 Comparison error for all resistance submissions versus grid size (Larsson and Zou(2010))

## 4.5 Conclusions

The computational simulations of ship hull flows are becoming everyday practice also outside the community developing the simulation methods. Surface capturing methods have become the dominating approach for free surface modelling. The increase in computational power and developments related to parallel computing have enabled grid resolutions, with which it is possible to consistently obtain calm water resistance predictions in model and full scale with a numerical uncertainty of a few percent. Advances in adaptive methods reduce the numerical uncertainty even further and are making it easier to obtain consistent predictions with reduced computational and user effort. The decreased numerical uncertainty makes it possible to reach more reliable validation and further improvement of the modelling approaches. The increased resources and accuracy has resulted in the capturing of even finer details of the flow field around ships, such as spilling of breaking bow and transom waves and primary and secondary vortex structures in the wake field, or to the extension of RANS based computations from calm water cases to prediction of added resistance in waves. However, in the latter case the accuracy of the predictions is quite variable.

Modelling of full scale flows has also advanced from the point of view of resistance with studies on the modelling of surface roughness effects and application of wall functions for the modelling of the full scale boundary layer. On the other hand, the lack of extensive validation data makes it difficult to estimate the reliability of the modelling approaches in full scale. Despite this uncertainty it seems that CFD has great potential to be used to improve the reliability of full scale predictions by providing insight into the validity of the various assumptions made in the extrapolation procedures and by being used together with model tests.

The largest uncertainties are still attributable to what they have been in the past, different turbulence models and how they are implemented in a particular RANS code, grid resolution and user experience with a particular code for a particular problem. However, the role of grid resolution is changing thanks to advances in computer power. Although one does not typically reach the asymptotic range with a grid, particularly at full-scale for practical problems, the advent of parallel processing and adaptive grid refinement methods is allowing grid uncertainty to be reduced to acceptable levels.

The RC suggests the following items for the future:

(i) Continue to review the developments and identify the need for research in the computation at full scale, free surface treatment, unsteady flows, and accurate modelling of turbulence. Full scale computations should consider issues of surface roughness and ability to correlate the computation with real ship data. A particular emphasis should be placed on the ability to compute a resistance curve with appropriate sinkage and trim as well as the application to realistic fully appended configurations. Validation by reliable data from experiments should also be considered.

(ii) Review and identify developments in the design of new ship concepts, improvements in design methods and diffusion of numerical optimization applications.

(iii) Continue to monitor the development and the use of Simulation Based Design environments, with special emphasis on geometry manipulation and parameterization, surrogate models and variable fidelity schemes applications.

(iv) A related area is that associated with the need to maximise energy efficiency of shipping through improved understanding of all sources of resistance. The fundamental assumption that an optimal hull shape is one that minimises calm water resistance may no longer be appropriate given the developments in CFD that give the designer the ability to make assessment of both wave and viscous effects for added resistance in waves as well as the interaction between hull-propulsor and appendages. A possible task is to evaluate the current capabilities in these areas.

## 5. SCALING AND EXTRAPOLATION

### 5.1 Introduction

One of the purposes of model testing is to acquire information on the resistance of a ship. The forces measured at model scale are “extrapolated” to full-scale values by a procedure originating from William Froude, but improved later by several others. The essence of Froude's approach is to split the drag of a ship into two independent components, one being a function of the Reynolds number, Re, and the other of the Froude number, Fr:

$$C_T(\mathrm{Re}, Fr) = C_V(\mathrm{Re}) + C_R(Fr)$$

The viscous resistance component, $C_V$, is assumed to be proportional to the resistance given by a ship-model correlation line:

$$C_V(\mathrm{Re}) = (1+k)C_F(\mathrm{Re})$$

In which k is the form factor and $C_F$ is a correlation line. Because model tests are carried out at the same Froude number as applies to the ship, the scale effects are concentrated in $C_V$.

It is typically assumed the $C_R$ and k do not vary with Reynolds number and by replacing the viscous or frictional resistance at model scale with the frictional resistance at full-scale an estimate of the full-scale resistance is obtained. This approach is consistent throughout the various ITTC procedures where the frictional resistance is determined by the ITTC-57 correlation line. A survey conducted by the 25th ITTC Specialist Committee on Powering Performance Prediction (2008) shows that many member organizations, although not all, use the ITTC-57 line and form factor concept in scaling model test data to full-scale.

## 5.2 ITTC-57 Correlation Line

One cause of confusion in the application of the above scaling or extrapolation method is that the original proposal of Froude based the viscous resistance on that of an equivalent flat plate. However, strictly speaking, the ITTC-57 line is not a flat plate correlation line, but a ship-model correlation line that already contains a form factor correction.

The ITTC-57 correlation line was based on the Hughes version of a flat plate friction line. The Hughes flat plate line is given by

$$C_F = \frac{0.067}{(\log_{10}\mathrm{Re}-2)^2}$$

and the ITTC-1957 line is given by:

$$C_F = \frac{0.075}{(\log_{10}\mathrm{Re}-2)^2} = (1+0.1194)\frac{0.067}{(\log_{10}\mathrm{Re}-2)^2}$$

Various other flat plate friction lines have been and are being proposed as better approximations to a flat plate than the ITTC-57 line. In addition to the analytical studies conducted by the 25th ITTC RC aiming at a possible recommendation for a new formula for the friction line there is ongoing research on this topic. Within the European VIRTUE project a study on the numerical calculation of the friction resistance coefficient of an infinitely thin plate as a function of Reynolds number have been performed using different turbulence models, Eca and Hoekstra (2008).

The numerical results of this study have been compared with four analytical equations proposed for the frictional resistance coefficient of a flat plate: the Schoenherr Line, the ITTC-57 correlation line, the proposal of Grigson and the line derived by Katsui et al. (2003).

Eca and Hoekstra concluded from their numerical study that at the lowest Reynolds numbers, the differences between the friction lines obtained numerically are similar to the ones in the four lines proposed in the open literature. Also, at the highest Reynolds numbers the seven numerical lines are much closer to each other than the Schoenherr, ITTC-57, Grigson and Katsui et al. (2003) lines.

**5.3** Form Factor

The form factor, k, introduced earlier is a basic idea for improving Froude's Method proposed by Prohaska to take into account the viscous pressure resistance. The frictional resistance is still estimated using a correlation line and the viscous pressure resistance is taken into account by the form factor, which has to be determined for each hull form individually using resistance tests at low Froude Numbers. Obviously the form factor method must be more precise in theory, simply because the breakdown of the resistance is physically more correct.

The survey conducted by the 25th ITTC Specialist Committee on Powering Performance Prediction (2008) shows many organizations use the form factor method and many of those obtain the form factor from a low speed or Prohaska test. However, there is considerable difficulty in determining a reliable form factor for the individual ship. Problems, for example, arise from modern bulbous bows (almost all ship types), from flat sterns with large overhang (especially RoRo ships and cruise liners), from additional resistance due to stable separations and induced vortices (for example at full block vessels like tankers and bulk carriers) and from effects of large immersed transom sterns.

In a recent publication on the form factor using Prohaska's method, (Hollenbach, 2009) concluded:

- The form factor method increases the accuracy of predictions in cases where no bulbous bow and no immersed transom stern are present.

- The form factor method can increase the accuracy of predictions in cases where a reliable form factor can be determined from model tests.

- The determination of a suitable form factor using Prohaska's method is much more sensitive to modern hull form features than has been expected.

    - Modern bulbous bows hamper the determination of a reliable form factor even when fully submerged.

    - Modern bulbous bows in "off-design" conditions (e.g. smaller draughts than the design draught) make it nearly impossible to determine a reliable form factor from model tests.

    - Wetted transom sterns distort the determination of a reliable form factor in general, not only in the case of High Speed Vessels.

    - Typical RoRo stern shapes with flat, long overhang influence the determination of a reliable form factor when being immersed (see above), but also when the transom is emerged from the design waterline level.

    - For full block vessels the choice of a reasonable form factor has a significant influence on the predicted resistance. Following the past recommendations of ITTC of not using a form factor derived from the individual tests in cases where separations occur, but instead using a form factor derived from full forms, which do not suffer from severe separation problems, may cause considerably different predictions.

As discussed, obtaining a reliable form factor can be problematic. Compounding the situation is that at low speeds, where the form factor is determined, some of the largest measurement uncertainties occur and there is often large scatter in the experimental data (e.g. Toki (2008)).

An additional concern is the Reynolds number dependence on form factor. Although the original intent is that the form factor is independent of Reynolds number this does not appear to be the case. A variety of authors have shown Reynolds number dependence on form factor. Somewhat recently this has been shown using CFD by van der Ploeg et al. (2008) who demonstrated that different correlation lines lead to different dependencies on form factor. Similarly, the 25th ITTC Specialist Committee on Powering Performance Prediction (2008) compared various model data where they showed form factor was dependent on Reynolds number when using the ITTC-57 line, but less so when using the Grigson correlation line. As discussed, this does not necessarily mean one line is better than the other, but that one line may already have the scale effect built into it.

When scaling from model to full scale there is also the surface roughness of the full-scale ship to be considered. Recently, Shen and Hess (2011) showed that for large roughness protrusions that extend outside of the viscous sublayer into the log region of the boundary layer that the resistance to flow is largely the result of form drag and the use of the shear velocity method to account for the roughness effect is not appropriate. They have developed a new method based on boundary layer momentum thickness to scale the roughness.

### 5.4 Appendage Scaling

Many organizations also make a correction for scale effects on appendages. Generally, it is recommended to perform bare hull tests without any appendages and then a second series of tests with appendages to identify the additional resistance due to the appendages. These appended tests should be at a higher speed to avoid laminar flow on the appendages at model scale.

Recently in Germany the research project “scale effect on appendages” has been finalised. Two of the German model basins HSVA (Hamburg Ship Model Basin) and SVA Potsdam (Potsdam Model Basin) have performed this research project. The aim of this project was to investigate the uncertainties when scaling the results of resistance tests from model to full scale, especially for twin-screw ships. The state of the art at the time the project was initiated was performing resistance tests with and without appendages and extrapolating only a part of the appendage resistance to full scale (60/40-method).

For the development and the validation of a new extrapolation method viscous flow calculations around a twin-screw passenger vessel model for three different model scales and for full scale with and without appendages have been performed. In addition, model tests with and without appendages were carried out at HSVA and at SVA Potsdam.

The viscous flow calculations have been performed for three different model scales $\lambda$ = 36.25 (SVA Potsdam model), $\lambda$ = 23.2 (HSVA model), $\lambda$ = 11.2 and for full-scale $\lambda$ = 1. The calculations have been performed at model speeds equivalent to the full-scale ship speed of 24 knots.

The calculated resistance coefficients for the individual appendages in model scale are presented in the following Table 4.

Table 4 Resistance coefficients and components for appendages

| RN | 6.49E+07 | | | 2.29E+07 | | | 1.17E+07 | | |
|---|---|---|---|---|---|---|---|---|---|
| Scale | 11.6 | | | 23.2 | | | 36.25 | | |
| | $C_V$ *1000 | $C_F$ *1000 | $C_R$ *1000 | $C_V$ *1000 | $C_F$ *1000 | $C_R$ *1000 | $C_V$ *1000 | $C_F$ *1000 | $C_R$ *1000 |
| Rudders | 8.051 | 2.067 | 5.984 | 8.508 | 2.395 | 6.113 | 8.927 | 2.692 | 6.234 |
| Shafts, hull bossings | 5.966 | 2.398 | 3.569 | 6.215 | 2.747 | 3.469 | 6.418 | 3.095 | 3.323 |
| V-brackets | 30.73 | 3.089 | 27.65 | 30.42 | 3.697 | 26.72 | 30.09 | 4.206 | 25.88 |
| I-brackets | 7.576 | 2.025 | 5.551 | 7.201 | 2.400 | 4.801 | 7.097 | 2.289 | 4.807 |
| Stabilizer pockets | 1.599 | 0.960 | 0.640 | 1.855 | 1.088 | 0.768 | 2.147 | 1.171 | 0.976 |
| Forward thrusters | 13.16 | 0.011 | 13.15 | 11.72 | 0.000 | 11.72 | 10.74 | 0.035 | 10.71 |
| Aft thrusters | 12.11 | 0.041 | 12.07 | 11.93 | 0.047 | 11.88 | 11.64 | 0.054 | 11.58 |

Based on the results of the viscous flow calculations it is proposed to introduce a correlation factor, $f_C$, for extrapolating the appendage resistance respective the form factor $(1+k_2)$ of the appendage from model scale to full scale. For the twin-screw passenger vessel the following factors apply as given in Table 5.

The correlations factors $f_C$ indicate that the resistance of all openings has to be increased when extrapolating to full scale. For the appendages this differs for each individual appendage component. Obviously, much more than the usual 60 percent of the appendage resistance has to be used for the extrapolation to full scale.

Table 5 Appendage form factors

| | $(1+k_2)_{Model}$ | $f_C$ | $(1+k_2)_{Model}$ * $f_C = (1+k_2)_{Ship}$ |
|---|---|---|---|
| **Appendages** | | | |
| Rudders | 2.8 | 1.0 | 2.8 |
| Shafts, hull bossing | 2.2 | 0.7 | 1.8 |
| V-brackets | 11.0 | 1.0 | 11.0 |
| I-brackets | 3.0 | 1.3 | 3.5 |
| **Openings** | | | |
| Stabilizer pockets | 2.0 | 1.6 | 3.7 |
| Forward thrusters | 4.8 | 1.4 | 5.6 |
| Aft thrusters | 4.6 | 1.2 | 5.8 |

As it is not possible for practical reasons to investigate the effect of the individual appendage components within industrial projects in the detail as has been done here, it is proposed using a correlation factor of $f_C$ = 0.9…1.0 for appendages and for $f_C$ = 1.2…1.4 for openings. It should be noted here that the form factor of the V-brackets is much higher than could be expected. The results of the viscous flow calculations show that this is due to misalignment of the V-brackets.

Computational fluid dynamics may yet be the best way to estimate appendage drag for complicated stern appendage configurations. As discussed by Jiang (2009) the appendage drag of the surface ship, Joint High Speed Sealift (JHSS), was shown to be inaccurate when estimated with various empirical methods for the separate appendage components. These empirical methods have been shown to be inaccurate when the interaction between appendage components is considered. However, using a RANS solver provided very reasonable predictions as compared to model test data.

## 5.5 Conclusions

There is evidence to indicate there may be more suitable correlation lines than the ITTC-57 line for particular hull forms. However, one must remember that the original intent of the ITTC-57 line was to produce on average a better correlation among Geosim models of a variety of forms at different scale than the Schoenherr line. The ITTC-57 line seems to have fulfilled this role. Different correlation lines will have different behaviours and lead to correspondingly different effects on form factor. However, as pointed out by Toki (2008) and the 25th ITTC Specialist Committee on Powering Performance Prediction (2008) changing the correlation line did little to improve full scale predictions due to the scatter of model test data.

The scaling of the model resistance and developments in this field are continuously observed by ITTC. The applicability of the various proposals for a new friction line should be tested during the next ITTC period on the basis of the available results in the ITTC trials database. More importantly than a correlation line that better predicts flat plate friction data is the combination of correlation line and form factor for various hull forms. The scatter in experimental data at low speeds for determining form factor from model data makes this especially problematic.

Numerical methods may help to better understand the whole scaling procedure. The RC committee recommends the following combination of numerical and experimental investigations to be performed in future research:

- Select a set of modern hull forms for numerical and experimental investigation: one modern ultra-large container vessel, one modern high speed ferry with small transom immersion, one state of the art navy vessel with high transom immersion, one full block vessel with low form factor and one full block vessel with high form factor.
- Perform numerical free surface viscous flow calculations with the aim of determining the drag of each hull form for sea trial draught and design draught for various ship speeds for model scale. Compare with model test results.
- Perform numerical double-body viscous flow calculations for full scale Reynolds numbers, for model test (resistance test) Reynolds numbers and for Reynolds numbers, which can be achieved in circulation tanks in model scale (no free surface, higher flow speed possible). Perform double body model tests for the above model test Reynolds numbers. Determine form factors based on numerical and experimental results.
- Identify possible improvements of the ITTC extrapolation method.

An issue that also must be considered as part of scaling is surface roughness. CFD methods typically do not assume any surface roughness even for full-scale predictions. The various correlation lines in the literature have different levels of roughness that may not be well quantified and this may also account for some of the differences between them. Full-scale ships have their own roughness, much of which is commonly accounted for through correlation allowances. Trying to determine what the best approach for predicting full-scale behaviour via correlation lines or CFD should also consider roughness aspects to the procedure and how they are best accounted for.

In regards to appendage scaling from model to full-scale it is seen that it can be problematic and may be more of an art than a science as it often relies on empirical constants that are not well known or defined. Even with a stripping test at model scale it is seen from the above example that different scaling coefficients are needed for different appendage components. Consequently, it may ultimately be necessary to rely on numerical calculations for more definitive answers for specific appendages and hull forms.

In addition it is recommended that the RC continue to review scaling and extrapolation methods along with theoretical and experimental investigations on frictional correlation lines and their interdependence on form factor. It is clearly being shown that different correlation lines will require different form factors and there are correspondingly different scale effects related to these form factors. Especially significant is the determination of form factor as this can lead to some of the largest uncertainties in predicting full scale performance. Related to this is one of the findings from the world wide campaign that at low speeds the largest uncertainty may be due to the dynamometer measurements. Since the recommended means of obtaining the form factor is from the Prohaska test or similar low speed test it follows that this can be an area of significant uncertainty and it is recommended that the 27th RC look into this further.

Appendage scaling appears to be particularly problematic and arguably *ad hoc*. It is recommended to continue following trends related to appendage scaling and in particular determine if CFD based methods are able to provide a reliable role in this regards.

It is evident that model scale testing can benefit from the use of CFD to aid the process of deconstructing the results found from model scale tests and making an improved estimate of full scale behaviour. Examples are in the areas of propeller wake and appendage drag. A possible area for future work would be to examine the whole process of scaling and how CFD can be used as a complementary tool to improve full scale resistance prediction.

Related to scaling, correlation lines and CFD is the surface roughness. This is largely ignored in full-scale CFD calculations. The various friction correlation lines proposed over the decades have various levels of surface roughness. Full-scale ships have their own roughness. A potential area for future work is to examine the influence of roughness effects and what impact it has, or should have on: CFD, correlation lines and ability to predict full scale behaviour.

## 6. TURBULENCE STIMULATION

### 6.1 Introduction

The development of new and more innovative ship types, as described in the previous section places a greater requirement to understand the influence of the application of turbulence stimulators (TS) both on the bare hull as well as appendages. In approaching this task a review was conducted of the historical development of the ITTC work in this area, the developments of physical understanding of laminar-turbulent transition and the influence of various passive flow control devices. An example approach for evaluating the appropriate model scale resistance correction is given and suggestions made as to how the existing procedures that incorporate the application of turbulence stimulation can be used more effectively.

The main ITTC procedure that identifies how turbulence stimulation should be applied is 7.5-01-01-01 ‘Model manufacture ship models’. Primarily this states that a recognised type of TS should be applied and identified in the associated documentation. These are identified as sand grain strips, studs and wires. Typical model locations and device sizes are identified. Figure 44 from this procedure identifies the influence of angle of entry and model size on the desired location of standard trip studs back from the stem.

One of the concerns from the continued use of such a process is that with the passage of time the physical reasoning as to why specific sizes and locations are used becomes obscured. One of the aims of the following section is to provide an overview of these reasons and hence enable a clearer application of the existing procedures. In particular the RC’s task identifies smaller models and appendages in close proximity to the free surface such as bulbous bows. In particular the flow regime close to the bow is particularly sensitive to the interaction between viscous and free surface behaviour. Landweber and Patel (1979) provide an excellent review of the physical interactions of a ship boundary layer and the likely challenges of adopting a standard solution to ensuring laminar-turbulent transition at a fixed location.

It is worthwhile restating the reasons that TS is applied to a ship model. These are in decreasing order of importance to: (1) ensure that the flow regime at model scale is equivalent to that at full scale, (2) that the model scale flow is consistent and hence repeatable across the desired Froude number and between repeat tests, and (3) that a known scaling approach can be applied. From a practical model scale tank testing perspective TS has to be straightforward to apply. The current procedure is deemed to require greater clarification to ensure a consistent and known behaviour which allows appropriate stripping. What is required are simple, consistent methods with known influence.

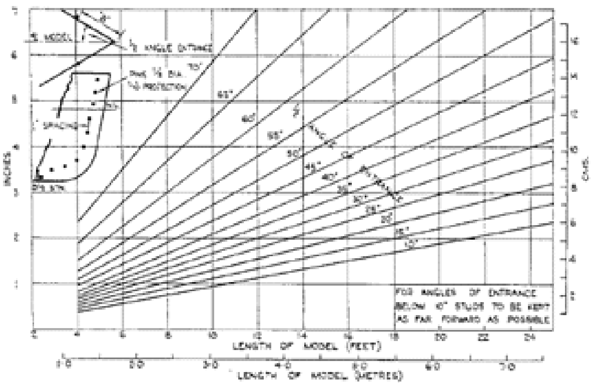


Figure 44 Location of studs as turbulence stimulators included in procedure 7.5-01-01-01

## 6.2 History of ITTC Investigation of TS

Table 7 collates an investigation of the discussion of Turbulence stimulation available through the historic on-line database of ITTC conference proceedings (http://ittc.sname.org). The various discussions are classified by the level of detail associated with the topic. What is evident is that only limited additional consideration has been made to the actual selection and application of the three classes of TS since the 1950s. This is reflected in the content of the procedures associated with TS.

Table 7 Location of references to TS in previous ITTC conference proceedings.

| Proc. | Comments | Page No |
|---|---|---|
| 9th | Incl. in name of a committee – many and varied incl. triangles/rotating upstream cylinders, thermistor for detection, effect of tank turbulence | 33-37, 51, 81-83,152-158, 165-168,172-174,179-187,194-199,204,211 |
| 10th a, | Three specific papers incl. details of many diff. types of stimulator | 10a(13,17), 10b(4-7,48-55,56-78) |
| 17th ,16th , 15th | One page section | 17(112-113, 122, 131, 136,138), 16(22-23,33), 15(25) |
| 18th, 14th, 13th, 11th a, | Limited Reference, 13a recommends studs, notes that smaller wake maybe good for self propulsion. | 18(49,63,95), 14(15,102,103), 11a(10,11,15) |
| 21st , 20th , 19th, 12th, | No mention | 20(38),19(56) |
| 24th ,23rd , 22nd | Request further work | 24(18,23,38), 23(54), 22(7,48,56) |

## 6.3 Fundamental Physics

The control of transition location such that it occurs at a fixed and known position is the function of a TS device. This type of passive flow control function is one of the sub-classes of the taxonomy identified by Gad-el-Hak (2000). Without the use of a TS device natural laminar-turbulent transition will occur at a location that is dependent on a number of parameters that include the levels of background turbulence, the surface roughness and associated boundary layer profile development upstream, the presence or otherwise of unsteady flow features, surface temperature and the local streamwise pressure gradient. Changes to any of these can quite radically move the location. For full scale ships, however, as the length based Reynolds number is typically $10^8$ or higher the location of transition will practically be at the stem.

At model scale the length based Reynolds number is often well within the range of that where transition can occur at a significant distance along the hull or for appendages that could always remain laminar. Table 8 identifies typical combinations of model/appendage length and Froude number for which most of the model would remain in the laminar flow regime.

Table 8 Length based Reynolds number with numbers in bold with a high proportion or a completely laminar flow regime

| | Model/appendage length (m) | | | | | |
|---|---|---|---|---|---|---|
| Fr | 0.01 | 0.1 | 1 | 2 | 5 | 10 |
| 0.1 | ***2.72 E+02*** | ***8.61 E+03*** | ***2.72E +05*** | ***7.70E +05*** | 3.05E +06 | 6.16E +06 |
| 0.2 | ***5.45 E+02*** | ***1.72 E+04*** | ***5.45E +05*** | 1.54E +06 | 6.09E +06 | 1.23E +07 |
| 0.3 | ***8.17 E+02*** | ***2.58 E+04*** | ***8.17E +05*** | 2.31E +06 | 9.14E +06 | 1.85E +07 |
| 0.5 | ***1.36 E+03*** | ***4.31 E+04*** | 1.36E +06 | 3.85E +06 | 1.52E +07 | 3.08E +07 |
| 0.7 | ***1.91 E+03*** | ***6.03 E+04*** | 1.91E +06 | 5.39E +06 | 2.13E +07 | 4.31E +07 |
| 0.9 | ***2.45 E+03*** | ***7.75 E+04*** | 2.45E +06 | 6.93E +06 | 2.74E +07 | 5.55E +07 |

The detailed review of Gad-el-Hak (2000) provides a comprehensive review of the extensive literature in the field of flow control and in the behaviour and types of devices used to control transition. The process of transition and how it is stimulated (or suppressed) are described in detail.

The stability of the laminar boundary layer velocity profile to disturbance is critical as to whether and how quickly transition will occur. The purpose of a TS device is to fix this location at a known position and so the number and size of devices has to be sufficient to ensure transition occurs rapidly and without significantly altering the form drag. Below a certain Reynolds number transition simply may not be possible and, in which case the TS simply acts as an additional source of viscous drag that requires appropriate scaling. An example of such a study is that of Smits (1982). The TS acts to mix high momentum flow down into the lower less energetic regions of the boundary layer. The generated disturbances can be associated with various unsteady vortex structures behind a trip stud as shown in Figure 45 (Pattenden et al., 2005, Pattenden et al., 2007), those on the flow 2D flow separation behind a trip wire or the more general generation of disturbances behind multiple elements on a roughness strip.

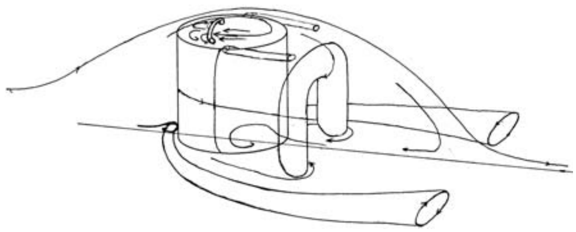

Figure 45 A schematic of typical vortical flow features around an aspect ratio 1 cylinder.

A critical parameter is the non-dimensional height (y+) of the device relative to the local boundary layer thickness. Typically a y+ value of at least 300 is required for 3D stud like devices or 600 for the less effective trip wire. The cost of the additional mixing that is promoted to initiate transition is an increase in momentum thickness, measured by for example Knobel (1978).

Murphy and Hearn (2007) report on a recent flat plate towing tank study of the influence of device size on transition location as well as the use of flow visualisation to identify transition. Figure 46 shows the variation in height required to initiate transition at a given location. Practical reasons will often dictate the actual device size chosen, in which case the momentum based resistance correction described later should be applied. Figure 47 from the same study shows the influence of a wire trip location on the relative drag of the flat plate.

There is a wide body of continuing publication in the area of passive flow control, see as examples on: effects of roughness (Ausone et al., 2007, Mathies et al., 2004, Piot et al., 2008), the use of a square rib (Bernitsas et al., 2008), 3D boundary layer transition (Saric et al 2003, Kohama, 2000), and transition initiation (Glezer et al., 1989). A collected series of articles on progress in modelling of transition is given in Borhani (2009).

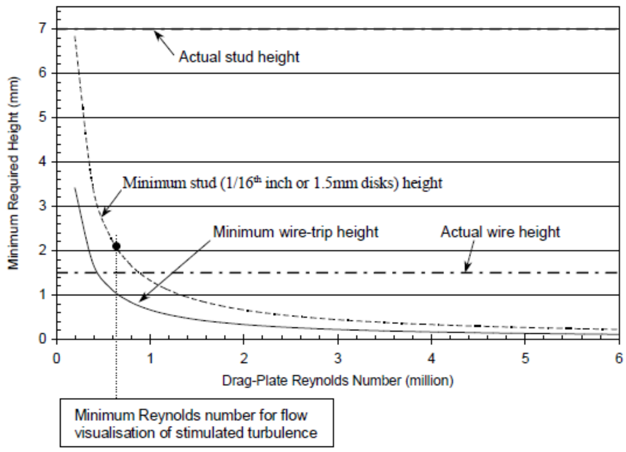


Figure 46 The minimum required stud or wire height to promote rapid transition for varying flat plate length Reynolds number (Murphy and Hearn, 2007)

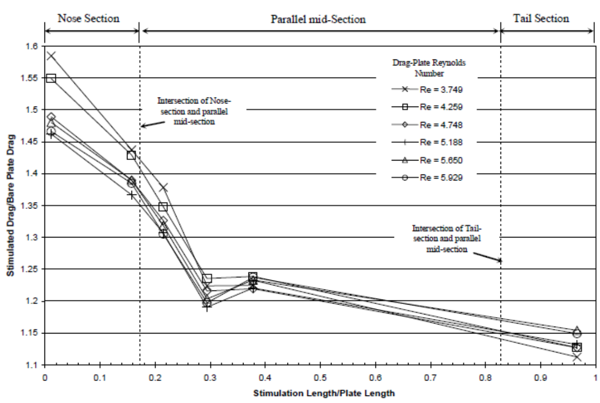


Figure 47 Variation in relative drag levels with wire turbulence stimulator location

Turbulence stimulation devices themselves can now be many and varied. The advent of new materials since the 1950s now allows much more discretion as to what device can be used. It is no longer necessary for example to use a standard aspect ratio 1 trip stud machined from brass. Bespoke shapes can be manufactured and applied using suitable glues to even the smallest of appendages. It is worth re-iterating that generally trip wires are less effective at promoting transition. That it is better to have a few roughness elements only rather than a high density attached to tape, often the tape thickness itself is sufficient to ensure transition and that use of a serrated tape edge is effective at promoting transition. It is worth checking that the spacing of 3D devices is sufficient otherwise large areas of laminar flow can be created downstream before the turbulent disturbance has spread to the full width.

The following section examines how for a device of known drag – in this case a trip stud- the trade-off in resistance between the laminar skin friction drag reduction and the increase in drag due to the form drag of the TS device itself can be accounted for.

### 6.4 Example Method for TS Model Scale Resistance Correction

As an example of the method used to evaluate the appropriate sizing and positioning of, in this case a unit aspect ratio trip stud an example calculation is reviewed from Molland et al. (1994) from which full details can be found. Figure 48 schematically represents the increase in momentum thickness $\delta_2$ at stud position $l_{laminar}$. This can be estimated for a given stud type and spacing as follows

$$\delta_{2stud} = \frac{hwnC_D}{4T_{stem}}\left(\frac{\bar{U}}{U_o}\right)^2$$

where $h$ is height, $w$ width of $n$ studs with a drag coefficient $C_D$ distributed over two sides of draught $T_{stem\_}$ in a freestream $U_o$ and average velocity $\bar{U}$ across the height of the stud.

The resultant boundary layer momentum thickness is now higher than that of an unstimulated but fully turbulent boundary layer. The actual momentum thickness at the stud position is the sum of the laminar momentum thickness and stud drag effects. An equivalent length can be calculated for which a turbulent boundary layer would have this same momentum thickness. This typically requires an iterative approach using an assumed length either based on the ITTC 1957 *Cf* line or equivalent theoretical approach until the same momentum thickness is found. An effective model length can then be found.

$$l_{effective} = l \quad - l_{laminar} + l_e$$

This length can be used to find $C_{Feffective}$ at the trailing edge and equivalent momentum thickness $\delta_2$. The overall skin friction aft of the trip studs can then be found as

$$C_{F_{turbulent}} = \frac{l_{effective} C_{F_{effective}} - 2\delta_{2_{total\,at\,stud}}}{(l - l_{laminar})}$$

The skin friction drag aft of the hull $D_{turbulent}$ can then be calculated as can the $D_{laminar}$ forward of the trip studs, the drag of the studs (already calculated) and the drag of the unstimulated turbulent flow to evaluate the model scale drag correction. Table 10 illustrates for a particular hull model (L=1.6m) the typical values found and the magnitude of the correction.

$$\Delta D = D_{unstimulated} - D_{turbulent} - D_{laminar} - D_{stud}$$

Table 10 Stud correction for model 6b at two speeds (Molland et al., 1994) All forces in N.

| U [m/s] | R | $D_{stud}$ | $D_{turb}$ | $D_{lam}$ | $D_{unstim}$ | Correct | % |
|---|---|---|---|---|---|---|---|
| 2.0 | 3.5 | -0.140 | 1.642 | 0.047 | 1.767 | -0.062 | 1.8 |
| 4.0 | 8.9 | 0.610 | 5.713 | 0.134 | 6.199 | -0.260 | 2.9 |

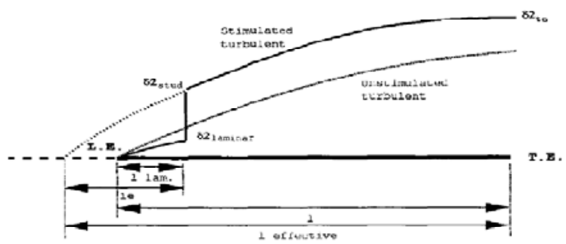


Figure 48 Development of boundary layer momentum thickness

## 6.5 Application to Appendage Scaling

In discussing the use of turbulence stimulation it is worth considering what options are available when the local appendage chord based Reynolds number is too small for transition. This is an important component of the process of appendage scaling and contributes to the general level of uncertainty associated with aspects such as the appropriate form factor (see Section 5.4, Table 5). In some circumstances the only option, as mentioned previously, is to carry out systematic stripping experiments to ensure the relative influences of the appendages on the local field are correctly captured and to use suitable empirical/theoretical methods to estimate the appendage contribution, see for example Molland and Turnock (2007).

In the case where the wake fraction based Reynolds number of the appendage is lower than $1x10^5$ it may be more appropriate to either use a rough surface appendage or add sufficient trip devices to give the equivalent momentum loss as if it were turbulent. This assessment will be complicated by appendages that act as lifting bodies, those whose streamlining at low Reynolds number gives rises to laminar separation bubbles or for whatever reason are effectively bluff. The rudder mounted behind the propeller lies in the accelerated race of the propeller giving a 20-50% rise in Reynolds number compared to the appended resistance case (Molland and Turnock, 2007). Many of these types of complexities need to be treated on a case by case basis. Although the application of CFD methodologies has the potential to aid the analysis process care still needs to be exercised when working at model scale as again low Reynolds number turbulence closure still has difficulties with large areas of laminar flow and transition (Borhani, 2009).

Often for structural reasons various appendages effectively behave as bluff bodies that introduce significant uncertainty into the viscous scaling process and into the application of flow control devices, in this case to fix separation. Again at model scale classic flow trips are effective at reducing variability with Froude number not seen at full scale. A review is given by Choi et al. (2008) with Figure 48 clearly illustrating quite how complex the response to different flow speeds is for flow around a sphere with different devices installed. In this case the appropriate length scale is the transverse width with a large form factor dependence on Reynolds number.

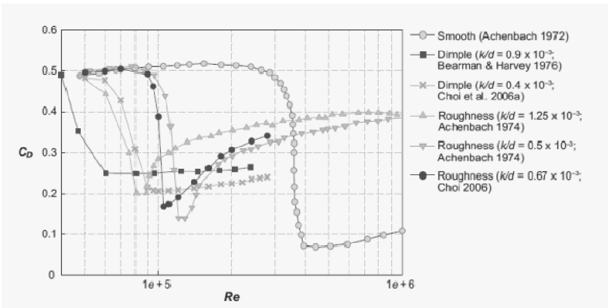


Figure 48 Variations of the drag coefficients for smooth, dimpled and roughened spheres, where k is the roughness or dimple depth (Choi et al., 2008).

As discussed in Section 5 for faired non-lifting devices a method similar to Section 4 can be applied to estimate the relative contributions of laminar and turbulent flow at model scale. However, this will become a far greater challenge if the section profile chosen is appropriate for full scale Reynolds number but at model scale has issues with laminar leading edge separation. It then, maybe more effective to select an alternative section to that used on the actual vessel.

For appendages that create sideforce or lift the presences or otherwise of TS devices can have a significant effect on the amount of lift generated for a given flow incidence and the maximum value of lift. This will have a bearing on the appropriate selection of TS for appendages for manoeuvring tests. Ideally for resistance tests appendages should be aligned to generate zero lift and thus eliminate what can be a significant component of induced drag that is much less dependent on Reynolds number.

A suggested approach for dealing with appendage resistance and selection of appropriate TS devices is as follows:

1. For each appendage component classify the device as bluff, faired or lifting.

2. Evaluate for the range of Fr to be tested what the expected Reynolds number range is

3. Select a suitable TS device that is of a appropriate size and spacing to force transition without adding a significant amount of extra momentum loss that will alter the wake downstream.

4. Examine whether for those devices where there is a risk of flow separation or which will remain mostly laminar what is the chosen strategy to mitigate the altered flow regime, either for example by using a rough surfaced appendage or altering its section profile.

5. Conduct progressive component stripping process, progressively adding appendages starting with the furthest forward.

6. Evaluate the resistance increase due to each component.

7. Scale the correct resistance contribution.

The procedures in which turbulence stimulation is identified all require an understanding of the expected flow regime at model scale test speeds. The correct process is already identified within the existing procedures to a greater or lesser extent. As a result only minor typographical corrections and a direct reference to this section are made to provide supporting background in order to ensure correct decisions are made about appropriate turbulence stimulation as and when it is of benefit.

## 6.6 Summary

An overview is given of the activities of the ITTC over the last six decades with regard to specifying appropriate turbulence stimulation devices. A much deeper theoretical understanding of the mechanisms that are associated with passive flow control of transition is now available and this includes both experimental and theoretical approaches. For those appendages away from the free surface reasonable estimations can be made of how appendage drag measurements made at model scale can be scaled. However, for devices such as bulbous bows great care has to be taken that the size and number of TS devices used does not fundamentally alter the bow wave and thus the progressive accumulation of pressure and skin friction resistance along the hull.

Overall, the procedures for which the use of TS is specified are still deemed appropriate, however, what is required especially for newer ship types is a fundamental assessment of the likely flow regime experienced by all appendages. From this an appropriate scaling process that corrects for the drag change due to the use of the TS device should be selected. An example of such a calculation is given for a small semi-displacement hull form with the maximum change in model resistance being 3%.

## 6.7 Conclusions

The major advance in the development of turbulence stimulation, TS, for ship model testing was based on work conducted over fifty years ago. A review has been made of the various developments as reported in the various committee reports of the Resistance committee over the ensuing period. The basic principles developed then of applying some form of passive flow control device that initiates boundary layer transition at a known location still applies. It is worth re-iterating that appropriate scaling of model scale resistance requires an estimate of the drag penalty associated with the TS device chosen. An example is given of how such 'striping' techniques should be applied. The theoretical understanding of fluid dynamic behaviour and in particular the mechanisms for transition have made considerable progress during the last fifty years. This knowledge allows the selection of less intrusive TS devices that can be tailored to a particular model or appendage.

The process for selecting a TS device should assess where boundary layer transition will occur, what magnitude of disturbance is required to effect transition and finally what is the most appropriate device. In some cases, notably on smaller model appendages and features such as bulbous bows the local length base Reynolds Number may be too small for transition to occur. Use of TS in such circumstances may not be appropriate as the TS device provides only a drag augment without recreating the equivalent turbulent boundary layer. Other analysis based techniques may then be appropriate, using for example CFD or analysis to estimate the unrepresentative flow behaviour at model scale. Such problems are particularly acute for high speed vessel tests were models tend to be smaller and appendages are a greater proportion of the resistance. Care should be taken in applying TS at or near the freesurface – it is likely that use of roughness strips will have less of an influence on the bow wave system than three dimensional devices such as trip studs.

# 7. RECOMMENDATIONS

Adopt the updated procedure No. 7.5-01-01-01 Model Manufacture Ship Models.

Adopt the updated procedure No. 7.5-02-02-01 Testing and Data Analysis Resistance Test.

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