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Speed and powering prediction for ships based on model testing

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Abstract

The ITTC78 method was originally designed for conventional single screw ships, but has later been modified to adapt twin screw ships and podded propulsors. Nowadays even more unconvensional propulsors are introduced, and the need for a new powering performance method is increasing. This thesis covers the load varying self propulsion method, and looks into how this corresponds to the standard ITTC78 method. The load varying method uses data from a self propusion test only, and uses a predefined increment value to change the revolutions during each run. In this way there are no need for an open water test or a resistance test, and time can be saved. In addition the vessel is tested as a unit, and not broken down into separate pieces like with the ITTC78 method.

Chapter 1

Sammendrag

ITTC78 metoden ble opprinnelig laget for konvensjonelle skip med en propell, men har senere blitt modifisert for å også takle skip med flere propeller og podder. I den senere tiden har ukonvensjonell fremdrift økt i omfang, og det er begrenset hvor lenge man kan modifisere denne metoden. Denne oppgaven tar for seg en lastvariert propulsjonsprøve, og sammenligner denne mot ITTC78 metoden. Den lastvarierte metoden bruker kun data fra en lastvariert propulsjonsprøve, det er ikke bruk for data fra friprøve og motstandsprøve. Dette kan bety betydelige tidsbesparelser.

NTNU Trondheim
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MASTER THESIS IN MARINE TECHNOLOGY

SPRING 2012

FOR

Espen Øyan

Speed and powering prediction for ships based on model testing

The final verification of the prediction for speed-power performance of a new ship is done by model testing. This type of testing has a long history. The procedure to reach a prediction for full scale is quite complicated, due to the high requirements for accuracy of the prediction (power predicted to within a few percent), and the complicated flow, where both wave making and viscous effects are important. In recent years, the introduction of podded propulsors, new antifouling paint systems, increasingly powerful CFD calculations, and increasing focus on energy-efficient ships has lead to a renewed interest in the methodology for carrying out and scaling model tests to predict speed and power for new ships.
The aim of the master thesis is to explore different ways of performing the calm water performance tests, and the powering prediction procedure following that.
On this background, it is recommended that the student shall do the following in the master thesis:

- Compare the results of load-varying tests with those of conventional resistance and propulsion tests and give your recommendations towards the use of load varying propulsion tests.
- Check how the continuously varied load-varying test compares with the constant RPM and conventional propulsion test, and give your recommendations towards the use of this methodology.
- Discuss the role and necessity of the propeller open water test in the powering prediction
- The estimation of the wake fraction is a key issue in some of the proposed load-varying propulsion procedures. Discuss the role of the wake fraction in this context, and how it might be estimated.
- Compare different varieties of the powering prediction methodology (scaling and full scale prediction process) in order to identify the most critical issues. Use the comparison with full scale trials give an indication of the correctness of the different procedures.

In the thesis the candidate shall present his personal contribution to the resolution of problem within the scope of the thesis work.

Theories and conclusions should be based on mathematical derivations and/or logic reasoning identifying the various steps in the deduction.

The candidate should utilize the existing possibilities for obtaining relevant literature.

The thesis should be organized in a rational manner to give a clear exposition of results, assessments, and conclusions. The text should be brief and to the point, with a clear language. Telegraphic language should be avoided.

The thesis shall contain the following elements: A text defining the scope, preface, list of contents, summary, main body of thesis, conclusions with recommendations for further work, list of symbols and acronyms, reference and (optional) appendices. All figures, tables and equations shall be numerated.

The supervisor may require that the candidate, in an early stage of the work, present a written plan for the completion of the work. The plan should include a budget for the use of computer and laboratory resources that will be charged to the department. Overruns shall be reported to the supervisor.

The original contribution of the candidate and material taken from other sources shall be clearly defined. Work from other sources shall be properly referenced using an acknowledged referencing system.

The thesis shall be submitted in two copies:

- Signed by the candidate
- The text defining the scope included
- In bound volume(s)
- Drawings and/or computer prints that cannot be bound should be organized in a separate folder.
- The bound volume shall be accompanied by a CD or DVD containing the written thesis in Word or PDF format. In case computer programs have been made as part of the thesis work, the source code shall be included. In case of experimental work, the experimental results shall be included in a suitable electronic format.

Supervisor : Professor Sverre Steen
Start : 16.01.2012
Deadline : 15.06.2012

Trondheim, 16.01.2012

Sverre Steen
Supervisor

Preface

This report is the result of my masters thesis, written in the last semester of my Master program in Marine Hydrodynamics at the Department of Marine Technology at NTNU.

The purpose of this thesis has been to look into the newly proposed method for conducting powering predictions, the load varying self propulsion method, and to compare it to MARINTEKs standard powering performance method (a modified version of the ITTC78 method) and full scale trials. Further I have looked into some limitations and advantages of this method and, based on the experience gained during the tests, written some suggestions on how to determine the important parameters in this method.

I would like to thank my supervisor, Professor Sverre Steen, for all help and discussions during the semester and the staff at MARINTEK for valuable help regarding practical issues during the tests. I would also like to thank my fellow students for valuable discussions and tips, especially regarding the use of LaTeX.

Trondheim, 10 June 2012

Espen Øyan

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Chapter 2

Introduction

This master's thesis will look into different aspects of powering performance predictions based on the load varying self-propulsion method, a method originally proposed by Holtrop (2001). The main argument for leaving the standard ITTC78 method is the fact that it originally was designed for conventional single screw ships. Later it has been modified to also cover twin screw ships and podded propulsion. The resent inventions in ship propulsion are taking an increasingly larger part of the marked, which introduces new effects to include in the ITTC78 method. The load varying self-propulsion method seeks to solve these problems by minimizing the number of tests to be done in a performance prediction and being more adaptable to unconventional propulsors. This is done by testing the hull and propulsor as a system, and not broken into separate parts. The main argument for using the load varying method on conventional vessels as well is the time aspect. Where the ITTC78 method requires three separate tests to obtain all necessary information, the load varying method relies on a single test only: the load varying self-propulsion test. The time saved can therefore be significant, if the method delivers accurate results. This is the basis for this thesis; does the load varying method deliver reliable and accurate enough results?

To be able to say something about this method, three working hypothesis was established in the beginning of the work with the thesis:

- The results from the load varying extrapolation procedure inside the expected range, when comparing against the results from MARINTEKs standard procedure and full scale speed trials.
- Repeated tests will give the same results.
- The increment value used in the tests effects the results from the extrapolation procedure. If true, there is a certain limit where the results are starting to get severely skewed.

During the work with the thesis an additional hypothesis established itself:

- The chosen thrust range influences the results, when the thrust range of interest cannot be covered by one test.

The last hypothesis was established during the testing of M2890C, as some difficulties showed up during the tests.

Chapter 3

Ships used in the analysis

In this thesis, three different ships are used in the analysis of the load varying self-propulsion method. As it will be interesting to see whether the procedure can handle different types of ships, three quite different types was chosen: a twin screw car freighter, a single screw PANAMAX bulk carrier and a ducted single screw offshore vessel. All models was originally tested at MARINTEKs facility in Trondheim, Norway, but has also been through new resistance and propulsion tests, in addition to the load varying tests, during the work with this thesis. This has been done to exclude effects from slightly different loading conditions, differences in roughness of the hull due to storage and different water temperature, when performing the comparison between the load varying method and MARINTEKs standard method.

3.1 M2375J - Slender body car freighter

3.1.1 Basic information

MARINTEKs model M2375J, a slender body twin screw car freighter, went through its original resistance and propulsion tests in Trondheim in 2001. The model tests lead to the building of three sister ships, and all three ships went through full delivery trials upon delivery from the yard. The model has went through repeated testing the last years as a part of a course lectured at Norwegian University of Science and Technology (NTNU) every fall, hence a large amount of previously recorded data are available.

Table 3.1: Principal Hull Data - M2375J

Scale	25.676			
Waterline	WL2		Ship	Model
Length overall	L_{OA}	m	140.019	5.453
Length on designed waterline	L_{WL}	m	139.689	5.440
Length betw. perp.	L_{PP}	m	131.300	5.114
Breadth moulded	B	m	22.700	0.884
Draught at LPP/2	T	m	5.595	0.218
Draught at FP	T_{FP}	m	4.760	0.185
Draught at AP	T_{AP}	m	6.430	0.250
Trim (pos. aft)	t	m	1.670	0.065
Volume displacement	∇	m^3	9259.1	0.547
Displacement	Δ	t	9546.1	0.546
Block coefficient	C_B	-	0.5552	0.5552
Wetted surface	S	m^2	3516.22	5.334
Wetted surf. of transom stern	A_T	m^2	9.70	0.015

3.1.2 Model condition

The original model hull was fixed and repainted in 2006, but the years in storage and the frequent change in temperature due to the repeated testing has with no doubt effected the hull condition severely the last years. At its present state, with some cracks and defects, the hull should ideally been fixed and repainted again. Thus, the state of the hull should not affect the main purpose of this report, as the reference used when extrapolating the load varying self-propulsion test results are the ordinary resistance and propulsion tests, recorded during the same session. Table 3.1 presents the principal hull data for the ship.

3.2 M2890C - Full body bulk carrier

3.2.1 Basic information

MARINTEKs model M2890C, a PANAMAX full body bulk carrier, went through its original resistance and propulsion tests in Trondheim in 2009. The ship was later built, and went through full delivery trials upon delivery from the yard in the end of 2011. When comparing the original performance prediction against the full scale trials some questions were raised, as the results did not agree as well as expected. One parameter that may have influenced the original performance prediction is that the propeller open water test was conducted at 8.5Hz. This may have led to laminar flow around the propeller blades, and to ensure avoidance of such effect when retesting the model, an additional propeller open water test at 15Hz were conducted.

Table 3.2: Principal hull data - M2860C

Scale	49.048			
Waterline	WL4		Ship	Model
Length overall	L_{OA}	m	359.900	7.338
Length on designed waterline	L_{WL}	m	340.797	6.948
Length betw. perp.	L_{PP}	m	353.000	7.197
Breadth moulded	B	m	65.000	1.325
Draught at LPP/2	T	m	11.200	0.228
Draught at FP	T_{FP}	m	10.900	0.222
Draught at AP	T_{AP}	m	11.500	0.234
Trim (pos. aft)	t	m	0.600	0.012
Volume displacement	∇	m^3	196871.3	1.668
Displacement	Δ	t	202196.7	1.667
Block coefficient	C_B	-	0.7661	0.7661
Wetted surface	S	m^2	25018.50	10.400
Wetted surf. of transom stern	A_T	m^2	0.00	0.000

3.2.2 Model condition

The model has been in storage at MARINTEKs facility the last couple of years, and is in rather good shape with some cracks and defects. Most defects found were located between DWL (fully loaded condition) and WL4 (ballast condition), and were therefore not fixed as the new tests were conducted at WL4. The defects located underneath WL4 were only minor, and were fixed prior to the testing. The repairs done to the hull are considered so small that they should not affect the results significantly. This means that the original resistance and propulsion tests conducted in 2009 can be used for verification of the results from the re-testing conducted in 2012. Principal hull data for the ship is presented in table 3.2.

3.3 M3025A - Offshore vessel

3.3.1 Basic information

Model M3025A is an offshore vessel with a ducted propeller, which has not yet been built. Due to this the rest of the information about this vessel is restricted and cannot be presented here.

3.3.2 Model condition

The model was subjected to its ordinary resistance and propulsion tests at MARINTEKs facility during first quarter of 2012. The hull is therefore in almost perfect

Table 3.3: Principal hull data - M3025A

Scale	17.069			
Waterline	WL1		Ship	Model
Length overall	L_{OA}	m	75.550	4.426
Length on designed waterline	L_{WL}	m	75.287	4.411
Length betw. perp.	L_{PP}	m	73.400	4.300
Breadth moulded	B	m	16.000	0.937
Draught at LPP/2	T	m	6.400	0.375
Draught at FP	T_{FP}	m	6.400	0.375
Draught at AP	T_{AP}	m	6.400	0.375
Trim (pos. aft)	t	m	0.000	0.000
Volume displacement	∇	m^3	5575.8	1.121
Displacement	Δ	t	5748.6	1.120
Block coefficient	C_B	-	0.7418	0.7418
Wetted surface	S	m^2	1845.58	6.335
Wetted surf. of transom stern	A_T	m^2	0.68	0.002

condition, as it has not been subjected to storage at varying temperatures (which has been known to make the outer layer crack). Due to this, the results from the original tests are assumed representative for the model in its present condition. The plan is therefore to only conduct one verification run at 13 kn, to check the new recordings against the previously recorded series. Principal hull data for the ship is presented in table 3.3.

Chapter 4

General scaling laws

As for all model tests there are some basic requirements and assumptions that forms the basis of which the tests are performed on. These requirements are divided into three different criteria to ensure similarity in forces between model and full scale: geometric-, kinematic- and dynamic-similarity (Steen and Aarsnes, 2010, p6-9).

4.1 Geometric similarity

Geometric similarity assures that the model and full scale ship are geometrically similar, by requiring a scale factor λ to be constant between model and full scale. λ is defined as the ratio between length in model and full scale:

$$\lambda = \frac{L_s}{L_m} \tag{4.1}$$

This requirement is also valid for the surrounding environment. This implies that waves, water depth and hull roughness are supposed to be modeled with the same scale number λ. This is often easier said than done, and in some cases there are more convenient to make a correction for the error done by not fulfilling these criteria, than model correctly according to it. The most used correction for not fulfilling the geometrical similarity criteria are the hull roughness correction ΔC_F, that will be looked further into later in this report.

4.2 Kinematic similarity

Kinematic similarity requires equal ratios between velocities in model and full scale. This criteria implies that the flow around the hull will undergo similar motions in both model and full scale, hereby for instance that the advance coefficient J is the same in both scales.

4.3 Dynamic similarity

Dynamic similarity requirement states that ratio between forces should be equal in both model and full scale. This criterion therefore gives us the requirement to scale the model by the Froude number F_N, through requiring the same ratio between inertia and gravity forces:

$$F_N = \frac{V_m}{\sqrt{gL_m}} = \frac{V_s}{\sqrt{gL_s}} \tag{4.2}$$

This similarity ensures that the wave pattern for the same Froude number will be equal in model and full scale; hence the wave making resistance follows the Froude number. This is an important requirement since it is the basis for assuming that the residual resistance coefficient C_R, which mainly consists of wave making resistance, will be equal in both scales.

Further the dynamic similarity criteria imply that model tests should be scaled by the Reynolds number Re, to obtain the correct ratio between inertia and viscous forces:

$$Re = \frac{V_m L_m}{\nu_m} = \frac{V_s L_s}{\nu_s} \tag{4.3}$$

This is easier said than done, since it implies a model tow speed that is unrealistically high to full-fill this criteria. Therefore the convenient way of dealing with this problem is to scale by the Froude number, and make a correction for the wrong Reynolds number. This is done by applying a turbulence stimulator to the model, to ensure turbulent flow around the hull. This stimulator are usually placed a certain distance from FP, and it is assumed that the added friction due to the stimulator are compensated by the lack of turbulent flow in the area in front of it. Due to this assumption it is possible to calculate around the effected frictional resistance values when scaling the results.

Another important scaling law that follows from the dynamic similarity criteria is that the cavitation number σ has to be equal in both scales, if cavitation occurs.

Chapter 5

1978 ITTC Powering Performance Method

5.1 Introduction

In the 1970's there was no standard procedure on how to perform a performance prediction test for ships, and how to scale the results afterwards. Therefore the 14th ITTC conference in Ottawa (1975) recommended the 15th conference to look into finding *"an analytical method for general acceptance"* (Lindgren et al., 1978, p359-360). The 14th committee stated that Prohaskas method for determining the form factor k was the best method present, and that it should be the recommended way to determine this value. Further they stated that Method 55 was the best of the methods they had studied, and that it should be the starting point for the next committees work.

By this basis, the 15th ITTC Powering Performance Committee concentrated their work into finding an analytical performance prediction method. They studied several different methods and aspects for powering predictions, before their work culminated with the well-known "1978 ITTC Performance Prediction Method for Single Screw Ships". This method is today better known as the ITTC78 method, and it follows Method 55 briefly.

When the method was introduced in 1978, the committee assumed that it would be done further work to improve the method, and in their report they stated several parts that needed to be looked further into (Lindgren et al., 1978, p 360). However to a large extent this has not been done, and with only minor changes, the original method presented in 1978 is the most used powering prediction method around the world today.

The procedure consists of three different tests; The resistance test, The open water test and The self-propulsion test. Figure 5.1 shows an overview of how the entire procedure are built up, and what that is extracted from the different parts. When the tests are done, an extrapolation procedure is used to give a full scale prediction.

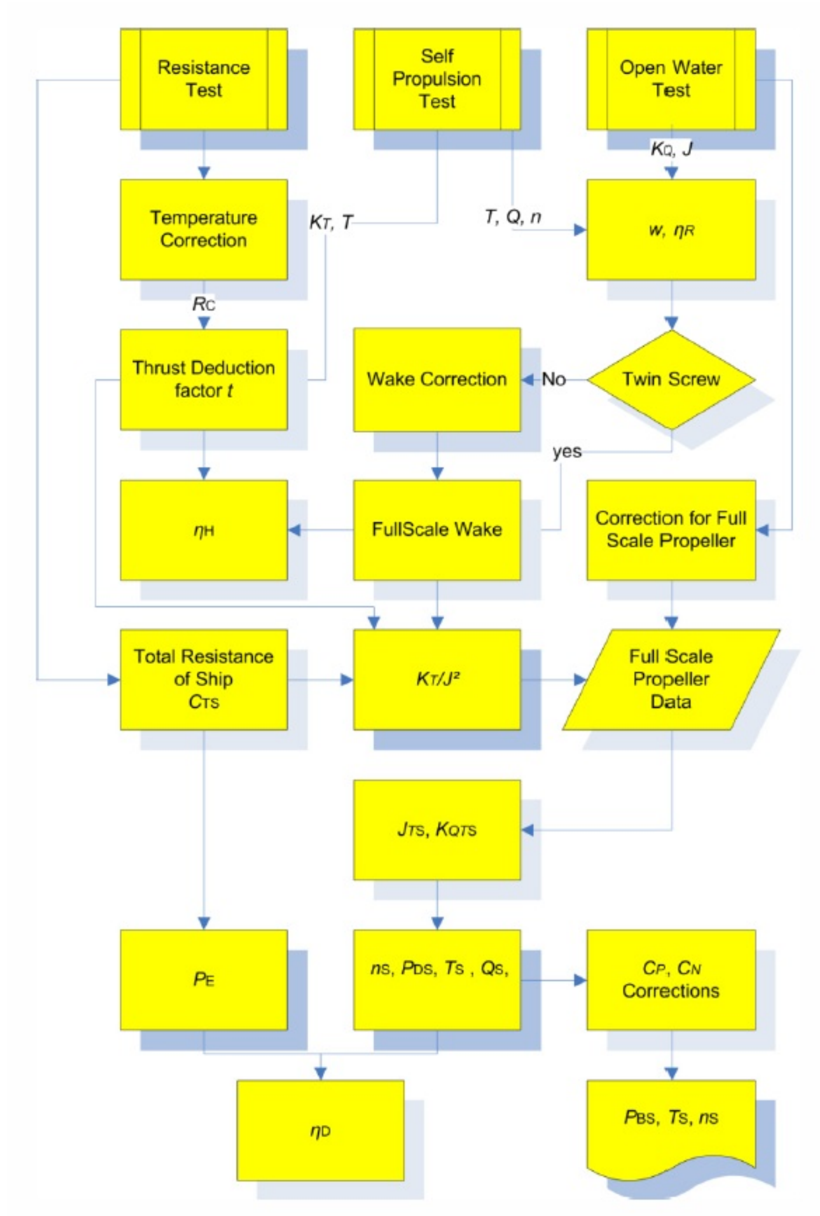


Figure 5.1: Overview of the ITTC78 procedure. (ITTC, 2008c, p.6)

5.2 The resistance test

The resistance test aims to determine the ship pure hull resistance at different speeds, and hence the test is performed without a propeller present in the aft of the ship. Further the test is usually done without appendices present, but it is possible to perform the test with appendices as well, using the approach presented in the next section.

5.2.1 Set up and test procedure

The resistance test is performed on the basis of geometric-, kinematic- and dynamic-similarity criteria. This implies that the geometrical similar scaled model are towed at Froude scaled speeds to ensure correctly scaled wave resistance. As presented in under section 4.3 it is impossible to full-fill the dynamic similarity criteria totally, as it implies to have Reynolds scaled speeds in addition to the Froude scaled speeds. To solve this, the ITTC'78 method uses the ITTC'57 correlation line (Lindgren et al., 1978, p. 375), and calculates the frictional resistance according to it. The friction line approach makes it necessary to separate the appendix resistance, if appendices are present on the hull during the test. This is done by first towing the model without the appendices, and then one tow with the appendices present. The difference in resistance is the appendix resistance, and this will be scaled separately from the hull resistance in the following scaling procedure.

The set up for a resistance test is presented in figure 5.2. As seen from the figure the model is attached to a towing carriage by a dynamometer that measures the resistance R_{Tm} during the test. Further the vessel speed V_m are recorded, and these two values are the input to the scaling procedure from the resistance test. These values will be used to determine the residual resistance coefficient C_R, which is assumed equal in model and full scale, and in the end to give an estimation of required power for the ship.

5.2.2 Analysis of Model scale results

The analysis procedure presented in this subsection is the procedure currently in use, as described by ITTC Recommended Practice (ITTC, 2008a). When the resis-

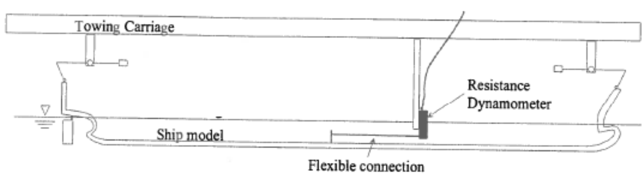


Figure 5.2: Set up for a resistance test (Steen and Aarsnes, 2010, p.61)

tance towing test has been performed, a data series including V_m and R_{Tm} will be used to determine necessary propulsive power for the full scale ship, through this extrapolation procedure. The procedure benefits from using dimensionless coefficients to express the different values. The first aim in this procedure is to determine the non-dimensional residual resistance coefficient C_R, through calculating all the other parts of the total resistance formula (equation 5.1):

$$C_{Tm} = (1+k)C_{Fm} + C_R + C_{AAm} \tag{5.1}$$

The first calculation to be done is to convert the discrete measured model resistance values to a non-dimensional form using formula 5.2:

$$C_{Tm} = \frac{R_{Tm}}{\frac{1}{2}\rho_m V_m S_m} \tag{5.2}$$

Here S_m is the wetted surface of the model, and ρ_m is water density in the model basin.

Next step is to determine the frictional resistance of the model. This follows the Reynolds number Re, and hence it cannot be scaled correctly due to the speed limitations in the towing tank (see section 4 for further discussion). Therefore this factor needs to be calculated analytically, and this is done based on the assumption that the hull friction can be represented as a flat plate with the same wetted surface S_m as the model hull, by using the ITTC'57 formula given in equation 5.3:

$$C_{Fm} = \frac{0.075}{(\log_{10} Re_m - 2)^2} \tag{5.3}$$

To obtain the actual resistance it is necessary to apply a correction for the fullness of the ship hull. This is done utilizing the form factor, by stating that $(1+k) * C_{Fm}$ corresponds to the ship models actual frictional resistance. Here k is determined by Prohaska's method, which implies a relatively low speed tow. According to ITTC (2008a) a low speed tow should be done in the range of $0.1 < F_N < 0.2$, to ensure low enough wave making resistance to neglect it.

If the model is equipped with a superstructure during the model test, the air resistance needs to be taken into account. The formula uses the transverse area of the superstructure A_{Tm} over the wetted surface S_m to give an empirical expression, presented in equation 5.4:

$$C_{AAm} = 0.001\frac{A_{Tm}}{S_m} \tag{5.4}$$

The only unknown part of equation 5.1 at this point are the residual resistance, which represents the wave making resistance for the hull. Equation 5.1 is rewritten to solve for C_R:

$$C_R = C_{Tm} - (1+k)C_{Fm} - C_{AAm} \tag{5.5}$$

This coefficient is, as stated earlier, assumed equal in model and full scale due to the dynamic similarity criteria from section 4. The full scale prediction will be handled in subsection 5.5.1, together with the rest of the extrapolation procedure.

5.3 The open water test

The purpose of the open water test is to determine the propeller performance in an undisturbed inflow situation. To ensure this, the test is performed without a ship hull present, and with the propeller attached in front of the towing equipment. The set-up for an open water test is showed in figure 5.3.

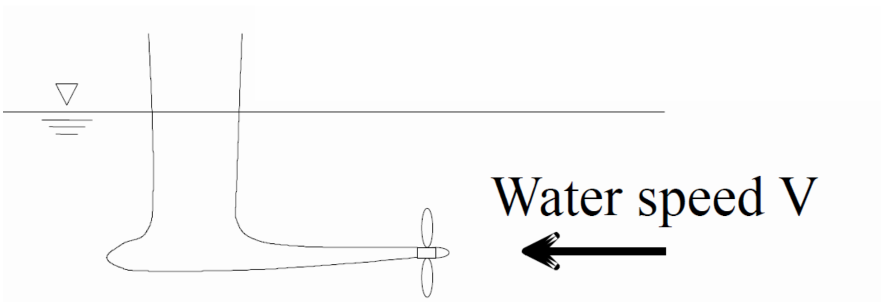


Figure 5.3: Set-up for an open water test (Steen and Aarsnes, 2010, p.62)

The acquired propeller open water characteristics are used to analyze the Propulsion test, and hereby the results will be used to estimate the required power (ITTC, 2008d). See also figure 5.1 for an overview of the entire procedure, and how the different tests interact with each other.

5.3.1 Measurements

There are several different ways to conduct an open water test. This part is based on the regular towing tank measurement, but it is also possible to conduct the test in a cavitation tunnel and to perform the test with ducted propellers if additional care is taken. This will not be mentioned any further in this section, as the underlying principals are the same, and hence the theoretical background is equal. When performing an open water test, tests for at least two different Reynolds numbers should be performed (ITTC, 2008d, p.8). One test should be done with the same Reynolds number used in the resistance test (but not lower than $Re = 2x10^5$), and one test at the highest Re possible.

The data measured during the test are:

- Speed (V_A)
- Thrust (T_M)
- Torque (Q_M)
- Propeller revolutions (n)
- Water temperature (T_{Water})

5.3.2 Analyses of the results

The recorded values for each velocity should be plotted against the speed of advance (ITTC, 2008d). This requires dimensionless values, and the thrust and torque values are made dimensionless by equation 5.6 and 5.7. The speed of advance is also expressed as a dimensionless value J, by equation 5.8.

$$K_{TM} = \frac{T_M}{\rho_M n_M^2 D_M^4} \tag{5.6}$$

$$K_{QM} = \frac{Q_M}{\rho_M n_M^2 D_M^5} \tag{5.7}$$

$$J = \frac{V_A}{n_M D_M} \tag{5.8}$$

The results from the open water test are usually presented as an open water diagram (Minsaas and Steen, 2008), a diagram that represents all the relevant information in a convenient way. An example of such diagram is presented in figure 5.4.

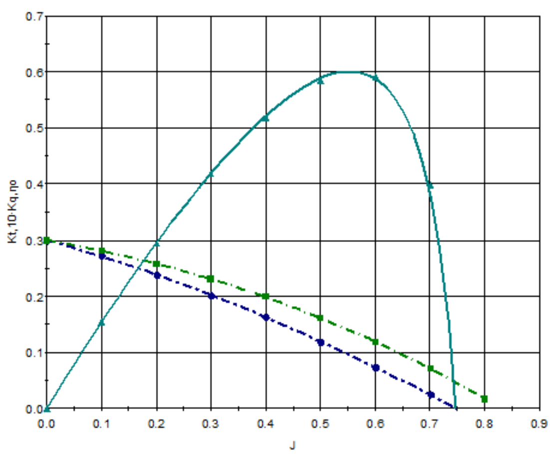


Figure 5.4: Example of an open water diagram from ShipX

As seen from the diagram in figure 5.4, the propeller efficiency η is also plotted against the advance coefficient. This parameter is calculated according to equation 5.9. Further note that the torque coefficient is plotted with a factor of 10, to make the results easier visible.

$$\eta = \frac{T_M V_A}{2\pi Q_M} = \frac{J_A}{2\pi} \frac{K_{TM}}{K_{QM}} \tag{5.9}$$

As seen from the overview in figure 5.1, the main parameters extracted from the open water test are the torque coefficient K_{QM} and the advance coefficient J. These are, together with data from the self-propulsion test, used to calculate wake fraction w_T and propeller efficiency η_R.

5.4 The self propulsion test

The purpose of the self-propulsion test is to perform a model test that takes the propeller-hull interaction effects into account. To ensure this, the test is performed with a scaled hull model fitted with a scaled propulsion system. To ensure consistency in the scaling method, the hull model has to be tested with the same ballast and trim configuration as the model was tested with in the resistance test (ITTC, 2008b, p.3). If appendices are present, they should also be in the same condition.

During the test runs continuous recordings of tow force, thrust, torque, rate of revolutions, trim and model speed are done (ITTC, 2008b, p.10), and together with the results from the open water test this makes it possible to investigate wake and thrust deduction effects.

5.4.1 Different applicable propulsion systems

The procedure is in general applicable for a great variety of propulsion systems, as the main purpose is to find out how the propulsion system interacts with the hull. Therefore ordinary one- or twin-screw propellers, ducted propellers, contra-rotating propellers, podded propulsion systems and similar propulsion systems based on propellers to accelerate the water, are treated the same way in this procedure (ITTC, 2008b). Thus the great variety, the procedure does not handle self-propulsion tests with water jet systems. Note that the extrapolation procedure from the tests are valid only for single screw and symmetric twin screw ships (ITTC, 2008b, p. 12).

5.4.2 Test procedure

When preforming a self-propulsion test there are two different approaches on how the test can be performed, the Constant speed (load varying) method and the Constant Loading method. Both methods are currently in use in model tanks today, and have their advantages and disadvantages. The ITTC recommended procedure (ITTC, 2008b, p.9) also addresses a third possible method, by combining the two already mentioned methods. This will however not be treated any further in this section.

5.4.2.1 Constant speed method (the British method)

The constant speed method is a load varying method, often better known as the British method. The set up for this test is the same as for a resistance tow, which means that the model is connected to a resistance dynamometer (as indicated in figure 5.5) that measures the actual resistance during the test.

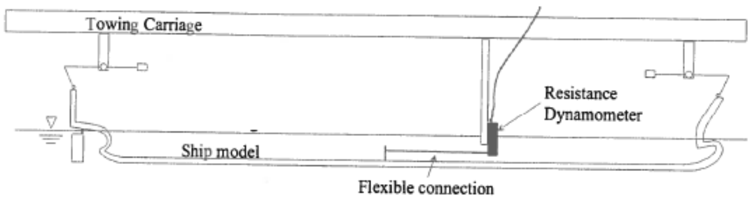


Figure 5.5: Set up for the constant speed method (Steen and Aarsnes, 2010, p.61)

When performing the self-propulsion test with this method, the model speed and desired propeller loading should be selected before each run, and the corresponding propeller thrust should be estimated (ITTC, 2008b, p.8). The towing carriage then accelerates the model to the desired speed, and simultaneously increases the propeller rate of revolutions so that desired thrust is reached closely after target speed is reached. The measurement of data starts when the running conditions have settled.

5.4.2.2 Constant loading method (the Continental method)

The constant loading method, also known as the continental method, requires a different set up than the one used for the resistance test and the constant speed test. This method introduces a tow force on the model, and the setup is shown in figure 5.6. The tow force should be computed before each run, and tuned such that it corresponds to the skin friction difference between model and full scale (ITTC, 2008b).

The test starts with the model being accelerated simultaneously as the propeller rate of revolutions are increased. When the model has reached the target speed, the propulsion system should (together with the applied tow force) propel the model freely at the same speed as the towing carriage. Measurements are started when a steady state is achieved.

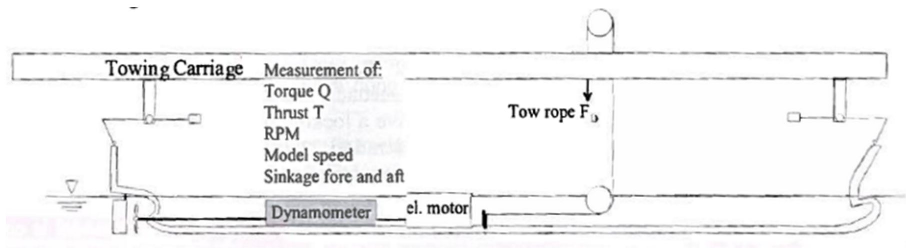


Figure 5.6: Set up for the constant loading method (Steen and Aarsnes, 2010, p.62)

This method requires several runs in a selected speed range. The lowest and highest test speeds should be selected such that it is at least 5 % below and above the scaled operational speeds for the full scale ship (ITTC, 2008b, p.9).

5.4.2.3 Comparison between the two methods

As mentioned earlier, both tests have their advantages and disadvantages. The load varying test requires several load varied tests for each speed, which obviously requires more time in the towing tank than the constant loading method which only needs one run for each speed. This makes the constant loading method cheaper to perform, and this is the reason why the constant loading method is the most used method today (Steen and Aarsnes, 2010, p.61). Thus it then seems little attractive to use more time and money on performing the load varying method, it also has an advantage over the constant loading method when it comes to using the results afterwards. As it has recordings for several loading conditions at each speed, it is easy to re-scale the results to different scaling ratios and different powering performance methods later(Steen and Aarsnes, 2010, p.62).

5.4.3 Analysis of the results

The analysis of the results presented in this chapter, follows the ITTC Recommended Practice ITTC (2008b). The aim with the analysis of the self-propulsion test data, are to determine the thrust deduction factor t, wake fraction w_T, relative rotative efficiency η_R and the hull efficiency η_H (ITTC, 2008b, p.11). To obtain data for the thrust deduction factor, results from a resistance test and an open water test (or similar information for the propeller used) is required, as seen in equation 5.10. Note that this formula, and the following formulas in this section, is only valid for single screw and symmetrical twin screw ships.

$$t = \frac{(\sum T + F_D - R_{TM})}{\sum T} \tag{5.10}$$

Here T is the propeller thrust, R_{TM} is the resistance measured during the resistance test and F_D the tow force necessary to counteract for the wrongly scaled skin friction resistance during the propulsion test.

Further the wake fraction w_T is calculated according to equation 5.11, by utilizing the open water diagram. This is done by using K_TM as input data in the diagram, and reading off J_{TM} and K_{QTM} from the graphs.

$$w_T = 1 - \frac{J_{TM} n_M D_M}{V_M} \tag{5.11}$$

The two quantities calculated in equation 5.10 and 5.11 is now combined together in the hull efficiency parameter η_H, as shown in equation 5.12. This parameter expresses the difference between delivered power and effective thrust.

$$\eta_H = \frac{(1-t)}{(1-w_T)} \tag{5.12}$$

The relative rotative efficiency describes the difference between the propeller data obtained from the open water test and the data from the self-propulsion test. See equation 5.13.

$$\eta_R = \frac{K_{QTM}}{K_{QM}} \tag{5.13}$$

Note that in a case of a twin screw ship, the mean rpm and the sum of thrust and torque are the values going into the procedure.

5.5 The extrapolation method

The extrapolation procedure aims to determine an estimate of the ships delivered propulsive power and the propeller rate of revolutions, through estimating full scale values for all relevant factors from the three model scale tests performed. To be able to extrapolate results using the method described in this section, results and analysis from a resistance test and from a self-propulsion test must be obtained. In addition, information from an open water test, or similar information about the propeller characteristics for the propeller used in the self-propulsion test, must be present (ITTC, 2008c, p.2).

The method is based on the thrust identity principle, a method that is recommended for use when predicting the performance of ships (ITTC, 2008c, p.2). This implies that the thrust deduction factor and the relative rotative efficiency are equal in model and full scale, hence the scaling effects will be taken into account on all the other coefficients.

5.5.1 Scaling the resistance test results

This subsection handles only the extrapolation of the results to full scale, analysis of the model scale values is found in subsection 5.2.2.

The equation that are going to be solved in this stage of the procedure is similar to equation 5.1, but includes some additional terms. This is shown in equation 5.14.

$$C_{Ts} = (1+k)C_{Fs} + \Delta C_F + C_R + C_{AAs} + C_A \tag{5.14}$$

Here k is assumed to be equal to the k-value from model scale, C_{Fs} are calculated according to the ITTC57 friction line as presented in equation 5.3 and C_{AAs} are calculated using equation 5.15.

$$C_{AAs} = \frac{1}{2} \cdot \rho_A \cdot V_S^2 \cdot C_{DA} \cdot \frac{A_{VS}}{S_S} \tag{5.15}$$

Here A_{VS} is projected area of the superstructure, S_S the wetted surface of the hull in full scale, ρ_A the air density and C_{DS} the air drag coefficient. This coefficient can either be determined by a wind tunnel test or be assumed to be 0.8, according to ITTC (2008c, p 5).

Further C_A is the correlation allowance and ΔC_F is the roughness allowance. In the original method these two coefficients were combined into the C_A, but the 19th ITTC Performance Prediction Committee proposed a separation of this value into two independent coefficients, to allow for the effects from newly developed hull coating systems (ITTC, 2008c, p 4). The formula for ΔC_F are given in equation 5.16.

The main purpose of the C_A value are to take into account effects that are not included in the prediction method, like systematic errors in the model test and powering performance prediction, including any facility bias (ITTC, 2008c, p 8). Due to different equipment in the different towing tanks, this value has to be tuned individually for each tank. Most tanks have built up their own correlation database, based on comparison between model tests and full scale trails. This database are usually used to determine the correlation factor C_A.

$$\Delta C_F = 0.044 \cdot \left[\left(\frac{k_S}{L_{WL}} \right)^{\frac{1}{3}} - 10 \cdot Re^{-\frac{1}{3}} \right] + 0.000125 \tag{5.16}$$

Here k_S is an indication on the hull surface roughness. If measured data of this parameter does not exist, $k_S = 150 * 10^{-3}$ is the recommended value to use according to ITTC (2008c, p 4).

At this point in the scaling procedure, all values except C_{Ts} are known. Equation 5.14 can therefore be solved to determine this value, and the procedure is closed when C_{Ts} is used to determine R_{Ts} through formula 5.17.

$$R_{TS} = \frac{C_{TS} \cdot \rho_S \cdot V_S \cdot S_S}{2} \tag{5.17}$$

5.5.2 Scaling the propeller characteristics

To be able to obtain a power estimate for the full scale ship, the entire propulsion system must be scaled. This implies scaling the propeller open water test to obtain information about the propellers performance in full scale. The full scale propeller characteristics are calculating according to equation 5.18:

$$K_{TS} = K_{TM} - \Delta K_T \tag{5.18}$$

Here K_{TM} is the model scale thrust and δK_T is a thrust factor difference between model and full scale, calculated according to equation 5.19.

$$\Delta K_T = -\Delta C_D \cdot 0.3 \cdot \frac{P}{D} \cdot \frac{cZ}{D} \tag{5.19}$$

Where P is propeller pitch, D is propeller diameter, Z is number of propeller blades, c is the chord length and ΔC_D is the difference in drag coefficient between the propellers, calculated according to equation 5.20. Equation 5.21 and 5.22 gives the expressions for the model and full scale drag coefficients, respectively.

$$\Delta C_D = C_{DM} - C_{DS} \tag{5.20}$$

$$C_{DM} = 2\left(1 + 2\frac{t}{c}\right)\left[\frac{0.044}{(Re_{c0})^{\frac{1}{6}}} - \frac{5}{(Re_{c0})^{\frac{2}{3}}}\right] \tag{5.21}$$

$$C_{DS} = 2\left(1 + 2\frac{t}{c}\right)\left(1.89 + 1.62 \cdot \log\frac{c}{k_p}\right)^{-2.5} \tag{5.22}$$

In the two equations above, Re_{c0} is the local Reynolds number using Kempf's definition (ITTC, 2008c, p.7), and cannot be less than $2 \cdot 10^5$. Further k_p indicates the roughness of the blade. This value can be set $k_p = 320 \cdot 16^{-6}$ if no other value is given, according to (ITTC, 2008c, p.7).

5.5.3 Scaling of propeller operating conditions

For twin screw ships the scale effect on wake are usually small, and there is common to assume that $w_{TS} = w_{TM}$. For other cases the full scale wake can be estimated using results from the self-propulsion test, see equation 5.23.

$$w_{TS} = (t + w_R) + (w_{TM} - t - w_R)\frac{(1+k)C_{FS} + \Delta C_F}{(1+k)C_{FM}} \tag{5.23}$$

5.6 Assumptions behind the resistance test

As mentioned in section 5.2, the resistance tests are performed on the basis of geometric-, kinematic- and dynamic-similarity criteria. In addition to these general criteria, there are several assumptions that are only valid for this explicit procedure. These assumptions often give limitations on how the results can be obtained and extrapolated for further use.

5.6.1 Friction line assumption

The main assumption, besides the criteria discussed in section 4, are the assumptions done to correct for the viscous (frictional) resistance problem. As mentioned, it is not possible to full-fill the dynamic criteria completely as it requires both Froude- and Reynold-scaled speeds at the same time. The way this procedure overcomes this problem, is by using the ITTC57 friction line to correct for the wrong Reynolds number, and hereby the wrongly scaled viscous effects during the test.

Thus the use of the ITTC57 friction line solves the problem in the first case, it also creates several other things that need to be discussed. The first and maybe most important thing to consider is the validity for the use of a friction line in the first place. The main argument against using the friction line is that it only gives friction values for a flat plate with the same wetted surface as the model. This approach has been assumed to be ok for slender body hulls, but reaches limitations for more full-body hulls. Therefore a form factor (k) has been added to correct for the fullness of the hull. This has for instance been discussed by Raven et al. (2008), and they concluded that the friction line approach corresponds well to the physics present. Further they bring up the question of the validity for the form factor approach.

5.6.2 Form factor assumption

The form factor is determined by Prohaska's method, which implies relatively low speed tows. This brings up the assumption about the properties of k. Since k is found by low speed model tests, and is widely used outside its measured field, it is necessary to assume that the value is independent of speed and scale effect. However according to Raven et al. (2008), who investigated viscous resistance using CFD-tools, the form factor actually changes with scale, making an underestimation of 7% compared to the CFD-calculated values if a fixed form factor is used to predict full scale resistance.

5.6.3 Turbulent flow assumption

The last assumption regarding the viscous resistance is the assumption about fully developed turbulent flow. However, most Reynolds numbers present in model scale suggests laminar flow around the hull. To correct for this, it is common to apply a turbulence stimulator on the model. This normally consists of a nylon thread, sand strip or similar devices to "disturb" the flow, and hereby forcing it from laminar to turbulent. Hence, when applying the turbulence stimulator it is also applied an added resistance. To take this into account, the turbulence stimulator are usually placed at a certain distance from the bow, and the lack of turbulence in front of this are assumed equal to the added resistance created by the presents of the turbulence stimulator.

Chapter 6

MARINTEK's powering performance procedure

6.1 Introduction

Marintek (Norwegian Marine Technology Research Institute) is a part of the Sintef group, Scandinavia's largest independent research institute (Sintef, 2011). In cooperation with Norwegian University of Science and Technology (NTNU) they operate several towing tanks and an ocean basin, located on Marine Technology Center in Trondheim. This chapter will give an overview on Marintek's powering performance prediction method, by discussing the differences compared to the standard ITTC78 method.

6.2 Differences from the standard procedure

Marintek's motivation behind choosing another scaling method than the standard ITTC method, was to come up with a method that was closer to the actual physics appearing in the model tests (Minsaas, 1982). Another motivation argued by Minsaas (1982) is the fact that the ITTC78 method only to a small extent is adaptable for implementation of further knowledge. Based on this Marintek decided to stick with the outline of the ITTC78 method, with the well-known three tests, but change several parameters appearing in the extrapolation method. These differences will be presented later in this section. Further Marintek also stated that, independent of the method chosen, they had to fine tune the correlation factors to their equipment and facilities. This meant that non method had an advantage in the decision process by already having an existing correlation database.

6.2.1 Frictional resistance

Marintek's approach to the frictional resistance appearing in model testing is based on an assumption that the frictional resistance can be expressed as a function of the well-known ITTC57 friction line. Further Marintek also recognizes the form factor approach from the ITTC78 method, but have a different view on how this should be incorporated in the scaling method. Where the ITTC78 method uses Prohaskas method to determine the form factor, Marintek relies on a purely empirical formula based on the ship main dimensions (Minsaas, 1982, p.2) . The formula is given in equation 6.1, where the input parameter ϕ is given in equation 6.2.

$$k = 0.6\phi + 145\phi^{3.5} \tag{6.1}$$

$$\phi = \frac{C_B}{L_{WL}}\sqrt{(T_{AP} + T_{FP})B} \tag{6.2}$$

Here C_B is the block-coefficient, L_{WL} is the length of the ship in the waterline, and T_{AP} and T_{FP} are the draft at fore and aft of the ship.

The form factor formula is derived from experimental form factor values for a large amount of ships, and analytical expressions for rotational bodies and mirror models. These results where plotted in a graph (see figure 6.1), and the empirical form factor line were tuned to lie underneath the points.

The reason for tuning the line to the lower part of the dataset, is that high form factor values are assumed to contain a large portion of "false" viscous resistance components from induced and vortex resistance due to flow separation (Minsaas, 1982, p.8). These resistance components does not scale correctly if scaled with Reynolds number, and is therefore included as a part of the residuary resistance.

Note that (Minsaas, 1982, p.20-22) also discusses the form factor approach itself, and its speed dependency. The conclusion form the discussion is that it's not possible to say something precise about the validity. Thus it is stated that if the form factor is speed dependent, more of the residuary resistance C_R should have been scaled as viscous resistance. This speed dependency has later been proved to be present by Raven et al. (2008).

6.2.2 Roughness correction and correlation factor

In the original ITTC78 method, roughness correction was a part of the correlation factor C_A. As mentioned in section 5.5.1 these was later divided in two separate coefficients C_A and ΔC_F. Marintek's method has treated this factor individually since the method was implemented in the beginning of the 1980s. The main argument for this separation is that the roughness allowance should be allowed to influence the viscous resistance, as this is where this effect has its physical origin

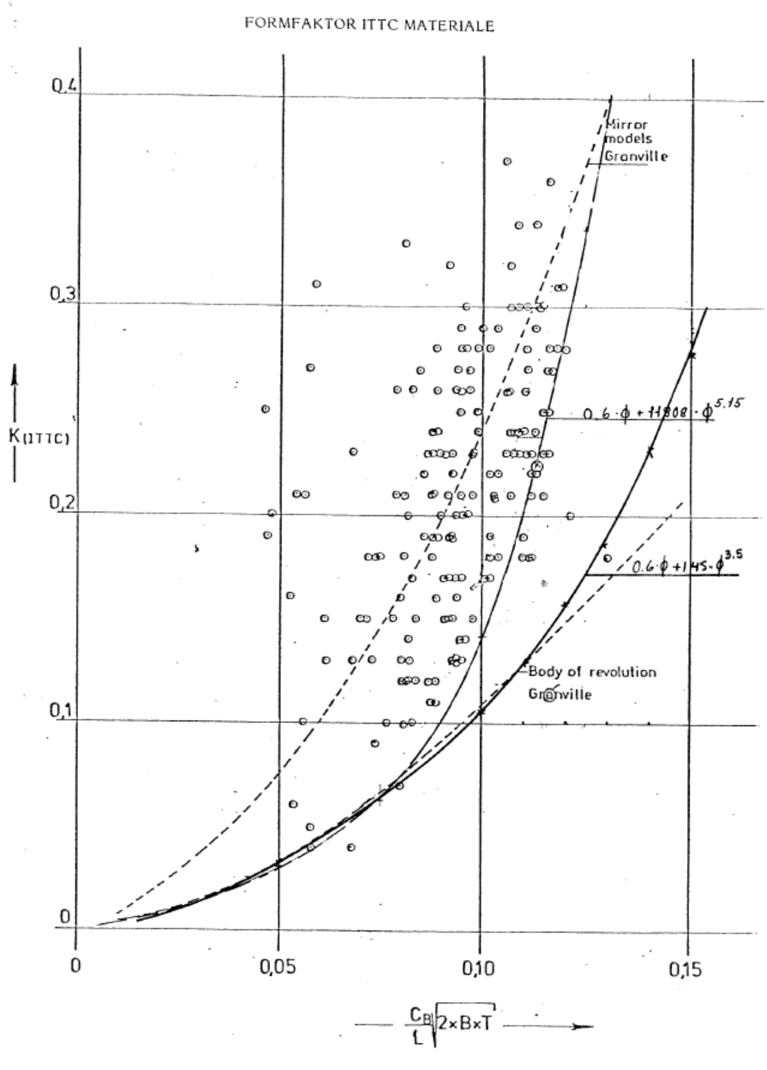


Figure 6.1: Marintek's form factor curve, plotted together with experimental and analytical values for ships and rotational bodies (Minsaas, 1982, p.7-8)

(Minsaas, 1982, p.9). This is only possible if these two coefficients are separated. However this separation brings up the question on how to estimate the roughness.

Roughness estimation is usually done by measuring the mean distance between the highest top and the deepest valley in a length of 50mm, on several different places on the hull (Steen, 2007, p.45). The average value of these measurement are the roughness parameter H. This parameter is the input to Marintek's roughness correction estimation, as shown in equation 6.3. Note that Marintek's roughness parameter H can be looked at in the same way as the ITTC78 roughness parameter k_S, but the unit (mm vs. m) and magnitude are different.

$$\Delta C_F = \left[110 \cdot (H \cdot V)^{0.21} - 403\right] \cdot C_F^2 \tag{6.3}$$

This is an empirical formula based on measurements done by Todd and Karlsson (Minsaas, 1982, p.10), that investigated the relationship between the roughness itself and the added resistance the roughness represents. The formula is based on an approach that suggests little vortex shedding in the boundary layer due to the roughness, which is present for most new builds the last decades (Minsaas, 1982, p.11). If no roughness parameter is known, $H = 150$ can be used as a standard value (Steen and Aarsnes, 2010, p.45).

6.2.3 Scaling of the propeller diagram

Marintek have decided to not scale the propeller diagram from model to full scale, hence they use the model scale value in the extrapolation method. Minsaas (1982) states that the correction of propeller diagrams should be one of the first things to look further into, especially since the approach chosen is missing an important link to the basic physics present and therefore in principal is incorrect.

The reason for this choice is that scale effects appear on both the moment and thrust curves in the propeller diagram, which means that a given K_Q value will have different K_T values in the model and full scale diagram(Minsaas, 1982, p.12). Further discussion by Minsaas (1982) shows that by scaling the propeller diagram, η_R will be scaled according to the change in C_D. In a case where other parameters than C_D also influences η_R, a scaling of the propeller diagram leads to a pessimistic speed prediction. Whereas in a case where the propeller diagram is not scaled, the corrections for full scale values are incorporated in the C_A coefficient. For further discussion and calculations proving this, see (Minsaas, 1982, p.11-15). As seen from this, Marintek is in principal not scaling their propeller diagrams, but in practice this have little to say as they correct for the wrong propeller characteristics when determining the C_A.

6.3 Discussion

Even though the procedure in use at Marintek follows the same outline as the ITTC78 method, there are several fundamental differences in the approaches founding the basis of the procedure as mentioned earlier in this chapter. As seen from the previous section, Marintek's approach has its basis in trying to separate and place all effects at its physical origin. This is seen by the form factor approach, where they try to separate "false" viscous resistance from the form factor expression to be able to scale the effects separately. It also follows with the roughness correction factor, which at the time Marintek's method was founded were not a separate factor in the standard ITTC78 method. Thus all this, they have chosen to not scale the propeller diagrams to full scale. This goes a bit against the physical approach which is the main basis of the method, but the results should not be affected to much by this as the scale effects are incorporated in the correction factor C_A.

Chapter 7

Load varying self propulsion method

7.1 Introduction

The load varying self-propulsion extrapolation procedure was originally proposed by Holtrop (2001), and was further investigated by Molloy (2001). The method seeks to predict full scale power based on the load varying self-propulsion test only, hence there are not necessary to conduct an open water test or a resistance test to obtain all information needed for completing the procedure. There are two approaches in use for conducting load varying tests:

- Constant revolutions.
- Constant increment of revolutions.

Section 7.3.1 and 7.3.2 will cover the two approaches, and discuss the different usage of them. The method covered by the rest of this chapter is valid for both approaches unless other is stated.

7.2 Set up and equipment

The method is founded on the recommendation to the ITTC from the 22^{nd} ITTC Specialist Committee on Unconventional Propulsors. The committee stated the following: «... a powering performance prediction for a ship equipped with unconventional propulsors should be tested as a unit, and not broken down into component tests of hull, propulsors etc.» (Bose et al., 1999, p.34). If this is fulfilled, the method will also be valid for ships with conventional propulsion systems.

The method has a setup similar to the British propulsion test setup (see section 5.4.2.1), hence a resistance dynamometer will transfer and measure the necessary force applied to the model during the tests. Further the thrust, torque and rpm will be measured separately for each propulsor in the vessel, to ensure proper input to the extrapolation procedure. The different parameters to be set and procedures to be completed before starting the tests will be discussed in the following sections.

7.3 Different approaches

To ensure selecting an appropriate increment value, and to decide which fixed rpm values to choose, a decision of which range of revolutions that is of interest must be made. The start and end values should be chosen for each speed in the selected range, as they are dependent on the model speed. The main focus should be to select these values such that they at least cover the area around the estimated self-propulsion point. Further there are recommended to also include thrust values as close to the zero thrust point and the zero tow force point as possible. If these points also are covered, a better estimation of respectively $F_{T=0}(\approx R_{TM})$ and $F_{T=100}$ can be made, which are two important parameters used in the extrapolation procedure.

7.3.1 The constant increment approach

The constant increment approach relies on a constant variation (increment) of the rate of revolutions throughout each run. The increment value has to be set prior to each run, and the ideal value will vary depending on several parameters such as tank length, model scale and number of propulsors installed.

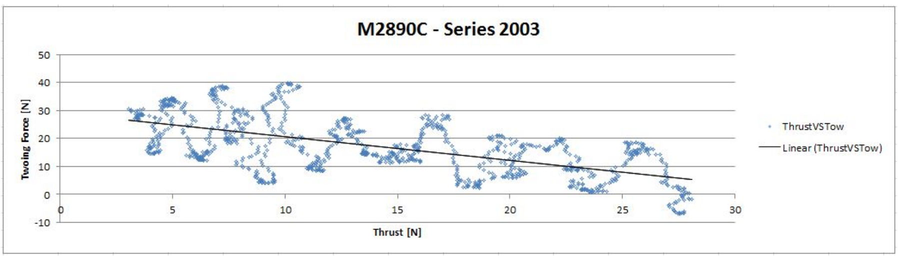


Figure 7.1: Thrust VS Towing Force - Visible oscillations

A large increment value will give a rapid change in revolutions during the run. This means that a large band of revolutions will be covered. A too large value can present visible oscillations in the recorded data, which can give some unexpected results. Figure 7.1 presents a thrust against towing force curve with oscillations visible. These oscillations will always be present in the data, hence it is not the increment value that introduces them. However the increment value makes them more pronounced as the high rate of change gives less data points around each revolution value. A smaller increment will give more data points around each revolution value, and therefore average out the oscillations and making them less visible in the data. Thus a very low increment value seems favorable to avoid such effects, the drawback by choosing a small increment value is that it may require several runs to cover the selected range of revolutions. This means more consumed time, without giving any better accuracy than a medium value. The most important part is therefore to select the appropriate increment value for the model. As guidance, based on the experience from the tests in this thesis, an estimation of the first increment value should be based on these criteria:

- Available tank length
- Model speed
- Estimated range of revolutions of interest

The available tank length is the tank length than can be used for recording data, after the first and last part is subtracted due to acceleration and deceleration of the model. This together with the model speed gives the available time for each run, which combined with the estimated range of revolutions gives an approximately increment value that could be the starting point for the analysis:

$$T_{available} = \frac{L_{available}}{V_M} \tag{7.1}$$

$$Increment = \frac{Range\ of\ rev.}{T_{available}} \tag{7.2}$$

Note that this should give an increment value somewhere around 0.05-0.3 rev/sec in a standard length tank.

To get a first impression of the quality of the data, a thrust against towing force curve can be plotted for each run (see figure 7.2). According to Holtrop (2001) the towing force against thrust curve should follow a linear trend, hence a linear trend-line should be a good representation of the mean value of the data. As seen from the figure, there are some small variations in the recorded data. This is however not a problem, as the procedure are using regression lines for all parameters. This means that it is the mean value throughout the recorded area that is used in the analysis.

The main argument for going with the increment approach is the time aspect, as long as the increment value is under control. This approach requires only one run for each speed, hence a significant amount of time can be saved compared to conducting the standard resistance, propulsion and open water tests. This is the approach preferred and used in this thesis, thus some issues with one of the models required additional testing with the constant revolution approach for verification.

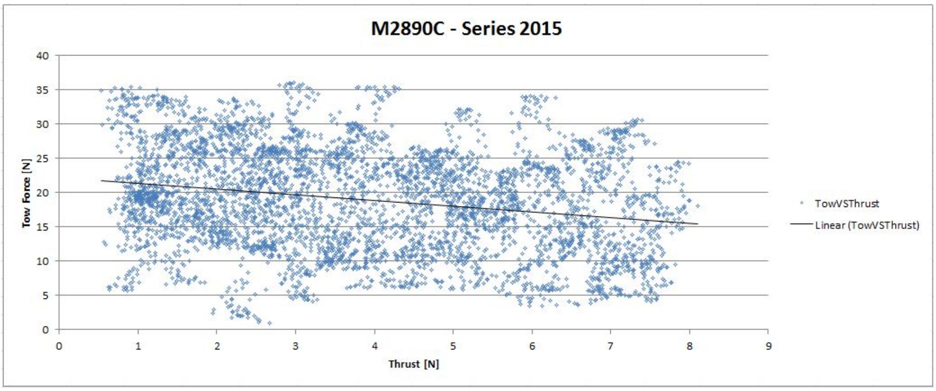


Figure 7.2: Thrust VS Towing Force - Constant increment

7.3.2 The constant revolution approach

The constant revolution approach is quite similar to the British propulsion tests, as it requires several runs at different propeller loadings, and interpolates between them afterwards. To be able to obtain reliable trend lines, the constant revolution approach requires at least three (preferably four) revolution points for all speeds in the selected speed range. Normally this would be done by recording one point during each run, but if the towing tank is of a certain length compared to the model speed (i.e. long enough to obtain two or more stable readings during one run) there are possible to record several points during each run. Note that a longer duration of the recording for each point, will give more certain results in the end.

To get a first impression of the quality of the data, a thrust against towing force curve can be plotted for each speed. If the data follows the expected path presented by Holtrop (2001), a linear regression line should fit between the different points for each speed (see figure 7.3). If there are large scatter in the data points, this could be an indication of some methodical error in the measurement. If there are single points that do not follow the trend, this point should ideally be re-run for verification. However if this is not possible, a consideration whether or not to include these points in the following analysis must be made (Molloy, 2001).

The main argument for going with the constant revolution approach is that the recorded time for each point(rate of revolutions) is longer, hence it will therefore smooth out some of the natural scatter in the measurement. This method was for instance useful for validation during the tests conducted with model M2890C, as the large mass of the model was excited by small vibrations induced by the towing cart tracks (see section **??** for more information). The main drawback with this approach is the time consumed to perform the tests. When having to perform three or four tests for each speed, there will hardly be any saved time compared to performing the tests with the familiar ITTC78 method. The preferred approach is therefore the constant increment method, as long as scatter in the data are not making any trouble with the results.

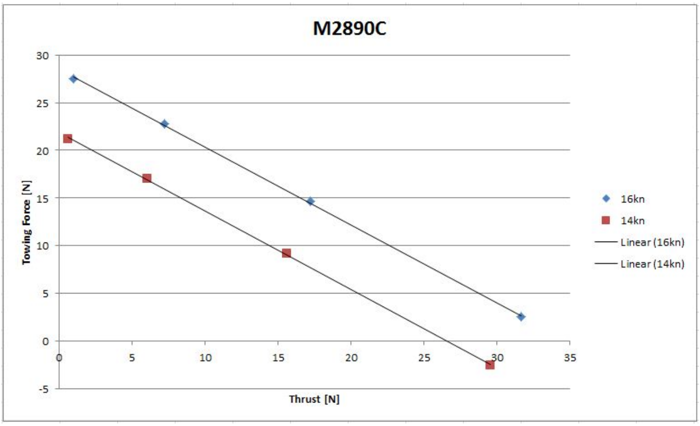


Figure 7.3: Thrust VS Towing Force - Constant revolution

7.4 Initial considerations and calculations

Before starting the test procedure, some considerations and calculations around the tests should be made. Most of these are also usually made during the tests with the ITTC78 method, like zero measurement settings and thrust and torque polynomials (see subsection 7.4.1. For the work with this thesis it was important to ensure repeatability, and a decision to have a fixed time between each run was made.

7.4.1 Thrust and torque polynomials

Thrust and torque polynomials can be considered in a similar way as the zero measurement points taken before initializing a test, and are included in both the ITTC78 and load varying procedure. The aim with the polynomials is to separate all effects related to the measurement system from the results. To ensure this, one torque and one thrust test is performed prior to the real measurements.

When taking the torque polynomial, several measurements are made to measure all losses in the measurement system due to effects from frictional effects in propeller bearings, the presents of the dynamometer and losses in torque due to the weight of the propeller. To be able to do this, a dummy boss with identical weight as the model propeller is placed on the propeller shaft. Then the shaft is accelerated to a value in the lower test range of revolutions, and the torque values are recorded for some time to ensure stable readings. This is repeated throughout the entire range of revolutions, and combined in a plot. From this plot a polynomial fit to the data is made, and the torque polynomial are extracted. See figure 7.4 for an example of a torque polynomial plot. The polynomial are applied to the raw data before the start of the analysis, this to ensure that only the effects related to the propulsion system itself are scaled in the following procedure. Note that a ship with multiple propulsors should record polynomials for each propulsor separately, as there can be individual effects appearing in the different systems.

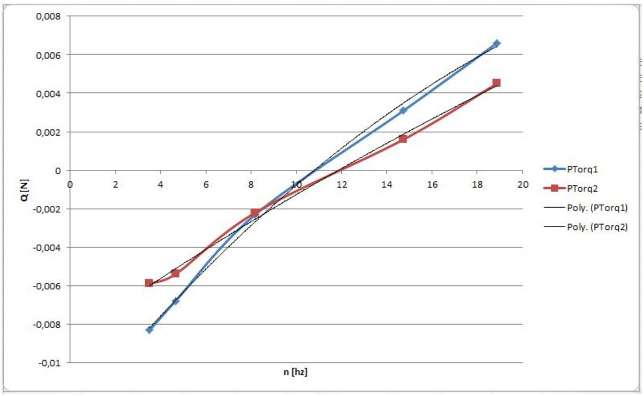


Figure 7.4: Torque polynomials for model M2375J (twin screw vessel)

The thrust polynomial is found in a similar way as the torque polynomial, but measures different parameters and effects. The thrust polynomial considers effects related to loss of thrust mainly due to trim of the vessel at different speeds. Therefore the thrust tests are performed at a fixed rate of revolutions, and the speed is varied. As in the torque test, there are no propeller attached to the propeller shaft in this test, but a dummy boss with identical weight is used as replacement. When the test is complete, the different thrust values recorded are plotted against the speed, and a regression line is fitted to the data. The thrust polynomial is applied to the raw data before the analysis is made, and ships with multiple propulsors should record values for each propulsor separately. A thrust polynomial plot is presented in figure 7.5.

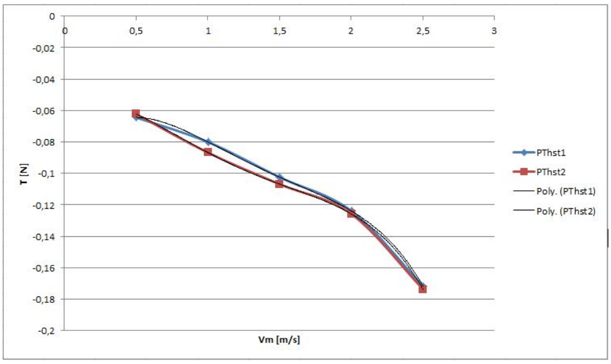


Figure 7.5: Thrust polynomials for model M2375J (twin screw vessel)

7.5 Frictional resistance parameters

The frictional resistance is an important parameter in the powering performance prediction method, and to obtain an accurate estimate is of great importance for the final results. As there are not conducted any separate resistance test, all necessary frictional parameters must be extracted from the load varying propulsion test.

7.5.1 Zero thrust point

The first frictional parameter to determine are the estimated zero thrust point $R_{TM_{T=0}}$. This point should, according to Holtrop (2001), be 1-4% larger than the measured calm water resistance from an ordinary resistance test. The difference between these parameters is assumed to be due to the propeller revolutions, and the effect it has on the water flow in the stern of the ship. The point can be determined by a linear regression line in the thrust against towing force diagram, see figure 7.2. The formula for the regression line should be at the form:

$$F = T_M \cdot (t - 1) + F_{T=0} \tag{7.3}$$

Here the last part of the equation, the interaction point with the y-axis, represents the necessary towing force at zero thrust $F_{T=0}$. This corresponds, as stated earlier, to the calm water resistance R_{TM} and are in the following procedure used as a substitute for this parameter. Note that the slope of the regression line follows (t-1), hence the thrust deduction factor t can be determined from this formula. This implies that the thrust deduction factor in this procedure is constant over the entire speed range. The validity of this assumption has been discussed by (Holtrop, 2001, p. 151), and there is at present time no evidence that this assumption is wrong.

7.5.2 Form factor

In the calculations done in this thesis the form factor is determined by MARINTEKs empirical form factor formula (see section **??** for further information). This is done to eliminate as many non-procedural errors as possible when comparing the two methods later in this thesis. Molloy (2001) chose Holtrops low-speed propulsion test approach when determining this parameter. This method has similarities with Prohaskas method, and is used at MARIN in the Netherlands. For more information about the alternative method for determining the form factor, see Molloy (2001) or Holtrop (2001).

7.5.3 Towing force

The towing force that normally would have been applied to the model during the test, can be calculated and incorporated in the results afterwards. This force corrects for the wrong Reynolds number in model scale, as it is impossible to scale correctly both with respect to the Froude and Reynolds number. The rope force is calculated according to the following formula:

$$F_D = C_S \cdot \frac{\rho_M}{2} \cdot V_M^2 \cdot S_M \tag{7.4}$$

where C_S are calculated according to formula 7.5. Further C_{FM} and C_{FS} are initially calculated according to the ITTC57 correlation line, and ΔC_F according to MARINTEKs standard procedure with H=150 (equation 7.6).

$$C_S = (C_{FM} - C_{FS} - \Delta C_F) \cdot (1 + k) - C_A \tag{7.5}$$

$$\Delta C_F = (110 \cdot (HV)^{0.21} - 404) \cdot C_{FS}^2 \tag{7.6}$$

The C_A coefficient is set equal to MARINTEKs standard value for the ship. This is determine by a correlation database, based on previous model and full scale tests performed in their facility.

$$C_{F_{ITTC57}} = \frac{0.075}{(log(R_N) - 2)^2} \tag{7.7}$$

7.6 Propulsion coefficients

7.6.1 Wake scaling

The wake fraction is often a parameter that introduces uncertainty in the analysis (Holtrop, 2001). The reason for this is the complexity of the flow in the aft of the ships, and the fact that the model scale wake differs significantly from the full scale wake. Finding an easy and strait forward way of calculating this parameter based on the load varying test only, is a challenge which still has to be solved. Several possibilities has been proposed, the most easy to use approaches is presented here.

Holtrop (2001) proposed the wake scaling formula presented in equation 7.8. This equation does however require data from a full scale trial to give the result, as it is dependent on V_A in full scale. This makes the formula inappropriate for most model scale tests as this in an unknown parameter.

$$\omega_{scale} = \frac{(\frac{V_A}{V})_s}{(\frac{V_A}{V})_m} \tag{7.8}$$

A solution to the problem can be to establish a wake fraction database based on previously tested ship designs, much alike the correlation database most towing tanks are using today. This is the recommended solution according to both Bose and Molloy (2001) and Holtrop (2001).

However there are several other possible solutions to the wake fraction problem. The use of data from an open water test will give good answers (Holtrop, 2001). This is the method currently in use when extrapolating with the ITTC78 method, and a lot of experience about this approach is found in towing tanks worldwide. The drawback by using this method is the need for an extra type of test, which is one of the things this method seeks to eliminate the use of.

Twin screw vessels usually have an open propeller arrangement with the propellers placed mostly outside the boundary layer, hence the wake scale effect is small (Holtrop, 2001). Due to this, the analyses of the twin screw vessel (M2375J) tested during the work with this thesis has been done without applying any wake scaling to the results.

The other two ships in this thesis has not wake scaling applied, as not enough information is obtained from the load varying tests. Different wake scaling methods has been tried in the analysis, but the results were not good enough to proceed. In the end a decision to not apply wake scaling was made, to stick to the idea that all information used in the extrapolation procedure should be accessible without additional tests. This decision introduces some uncertainties in the results, and the wake scaling problem is with no doubt the main argument against the load varying method. More investigations to try to obtain a more straight forward solution to this problem are required.

7.6.2 Estimation of the self-propulsion point

The self-propulsion point is defined as the point where the applied towing force is equal to the increased frictional resistance experienced by the model hull, due to the wrongly scaled viscous effects in model scale. It is important to get a good estimation of this point, as it is the main basis for the following extrapolation procedure. In this procedure the thrust identity approach is used to determine this point.

Unlike a standard ITTC78 test where the towing force is applied to the model during the tests, the towing force is calculated according to formula 7.4 (as discussed in section 7.5.3) and incorporated in the results before the estimation of the self-propulsion point is made. In practice this is done by determining the point where the measured towing force through the resistance dynamometer is zero. This implies that the all resistance forces acting on the model is counteracted by the thrust force alone. This point is in the following text denoted $F_{T=100}$. If the test has been performed in such way that this point is not a part of the result data, interpolation along the thrust vs. towing-force trend line established in section 7.5.1 can be used to determine this value. The equation used to determine the self-propulsion point is given in equation 7.9.

$$F_{SelfPropulsion} = F_{T=100} - F_D \tag{7.9}$$

This can also be seen in the thrust vs. towing force curve in figure 7.6, where point 1 is $F_{T=100}$. When adding the towing force, the self-propulsion point will change its position to point 2 in the figure. The thrust at this point is the true self propulsion thrust in model scale, and is used as the input thrust value in the following calculations.

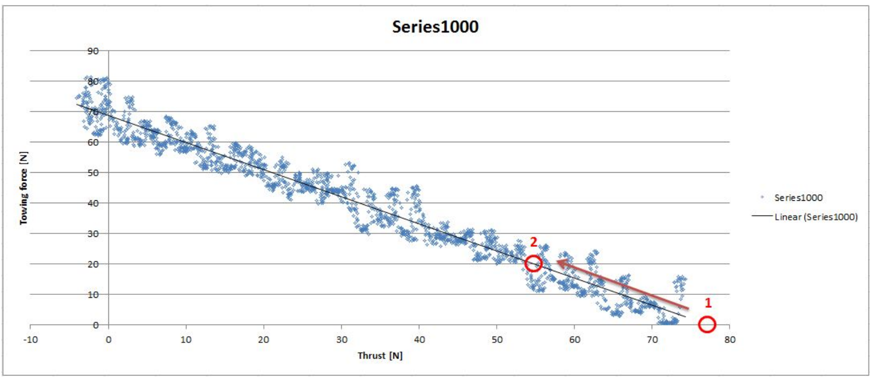


Figure 7.6: Self propulsion point changes position due to the calculated towing force

To determine the advance number for the self-propulsion point, the propeller rate of revolutions must be found. This is done by finding the corresponding rpm value for the self-propulsion thrust value determined by equation 7.9. This is presented in figure 7.7. Note that a twin screw vessel must divide the thrust on the two propellers according to their performance, and hence get one thrust and rpm value for each propulsor.

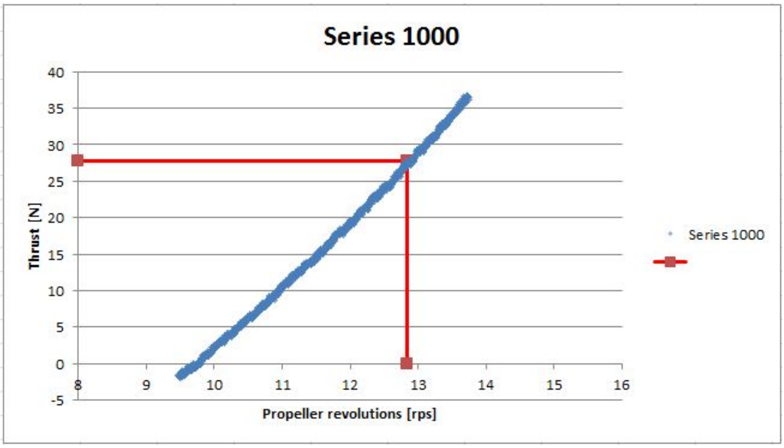


Figure 7.7: Thrust vs. rps curve. The corresponding rps value for the selected thrust value is indicated.

All parameters needed to determine the self-propulsion point is known, and equation 7.10 is used to calculate the value. This advance number is used as the primary input to the extrapolation to full scale.

$$J_0 = \frac{V_M}{n_M \cdot D_M} \tag{7.10}$$

7.6.3 Scaling of the propeller diagram

The ITTC78 method scales propeller characteristics to take the effect from the wrongly scaled Reynolds effects into account in the analysis. However, MARINTEK has chosen to not scale the propeller characteristics, as discussed by Minsaas (1982). Further discussion around this choice is covered in section **??**.

Holtrop (2001) also discusses the propeller diagram scaling, and states that following: «the performance of the real propeller, expressed in a no dimensional manner, deviates only to a minor degree from that of the model propeller» (Holtrop, 2001, p. 151). Further discussion leads to the following statement: «It is, however, not always the uncertainty in the propeller scale effect which is a factor of concern. In many cases, particularly in single-screw ships, it is often the uncertainty in the wake scale effects which is to be blamed for the poor correlation» (Holtrop, 2001, p. 152). Further discussion around the wake scaling effects is covered in section 7.6.1.

As MARINTEK does not scale the propeller diagram, a choice to not scale the propeller diagram in this method as well was made. This corresponds to what Holtrop (2001) discussed, in addition to the fact that the correlation coefficient

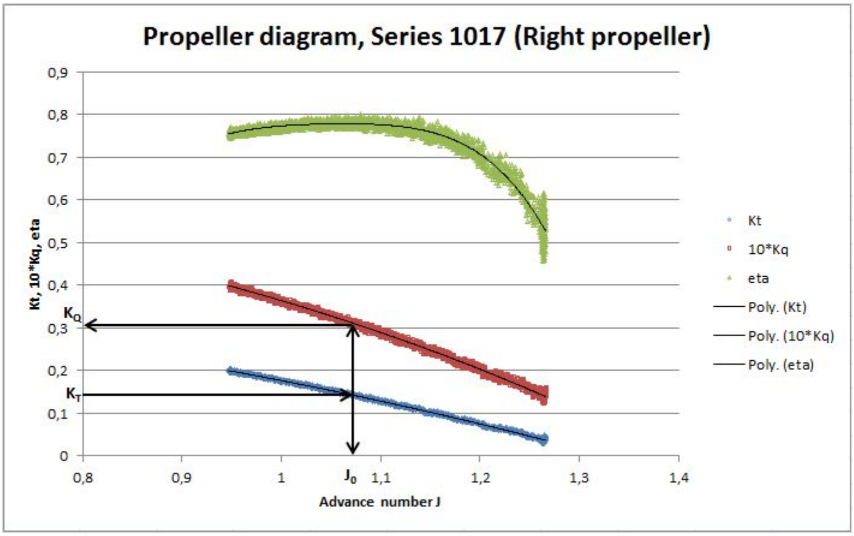


Figure 7.8: Propeller diagram in behind condition for model M2375J (right propeller). Thrust identity with K_T as input is used to obtain J_0. Further torque identity is applied to obtain K_Q and in the end Q_S

used in the analysis are based on a non-propeller scaled database.

7.6.4 Propeller diagram

The propeller diagram is at this stage not established, but this will be done here. Please note that the propeller diagram in this section is a propeller diagram for the propeller in the behind condition, hence coefficients like η_R cannot be calculated based on the diagram. Further the propeller diagram only covers the area covered by the load varying propulsion test, and not the entire range as a normal propeller diagram from an open water test usually do. In figure 7.8 a propeller diagram extracted from a load varying test is presented. The equations used to establish the thrust- and torque- coefficients are presented in equation 7.11 and 7.12. Further the advance coefficient is calculated according to equation 7.10 and η according to equation 7.13.

$$K_T = \frac{T}{\rho \cdot n^2 \cdot D^4} \tag{7.11}$$

$$K_Q = \frac{Q}{\rho \cdot n^2 \cdot D^5} \tag{7.12}$$

$$\eta = \frac{K_T \cdot J}{2 \cdot \pi \cdot K_Q} \tag{7.13}$$

The calculated J_0 value has a corresponding non-dimensional thrust- and torque-coefficient (see figure 7.8). This will be utilized when extrapolating the results to full scale in next section.

7.7 Extrapolation to full scale

The extrapolation to full scale values are based on the model self-propulsion point J_0. As discussed in section 7.6.3 the model propeller diagram is not scaled in this extrapolation procedure. However this does not necessarily mean that the self-propulsion point is equal in model and full scale. There are other parameters that can influence the self-propulsion point, for instance the wake. For most twin screw vessels and ships with ducted propulsors the wake fraction is small, which means that the wake influence on the results is small. However this is usually not the case for ordinary single screw vessels, where the wake fraction can be significant. In a case with a conventional single screw ship, the wake fraction should normally be scaled, as described in section 7.6.1. This scaling of the wake fraction will lead to a change in the full scale self-propulsion point, and the interpolation usually performed when scaling the propeller diagram must be made to ensure the correct location of the self-propulsion point. The interpolation formula is presented in equation 7.14 (Bose and Molloy, 2001).

$$K_{TS} = J_0^2 \cdot \frac{T_S}{2 \cdot \rho \cdot D_S^2 \cdot V_S^2} \tag{7.14}$$

Here T_S is calculated according to equation 7.15 (Bose and Molloy, 2001).

$$T_S = (\frac{F_{T=0} - F_D}{(1-t)}) \cdot \lambda^3 \cdot \frac{\rho_S}{\rho_M} \tag{7.15}$$

K_{TS} is plotted together K_T, and the intersection between the curves in the plot indicates the full scale self-propulsion point (see figure 7.9). This new full scale propulsion point is denoted J^*, and will be used as input in the following calculations. For twin screw ships, or other ships with so small wake fractions that scaling of the wake can be neglected, $J_0 = J^*$ in the following calculations.

In this extrapolation to full scale torque identity will be applied, hence the calculations are based on information from the torque coefficient. To obtain a powering prediction, equation 7.18 must be solved. This means that the full scale propeller revolutions and full scale torque must be calculated first.

Based on the full scale propulsion point J^* and the advance number formula (see equation 7.10 it is easy to obtain the full scale propeller revolutions. This is done according to formula 7.16.

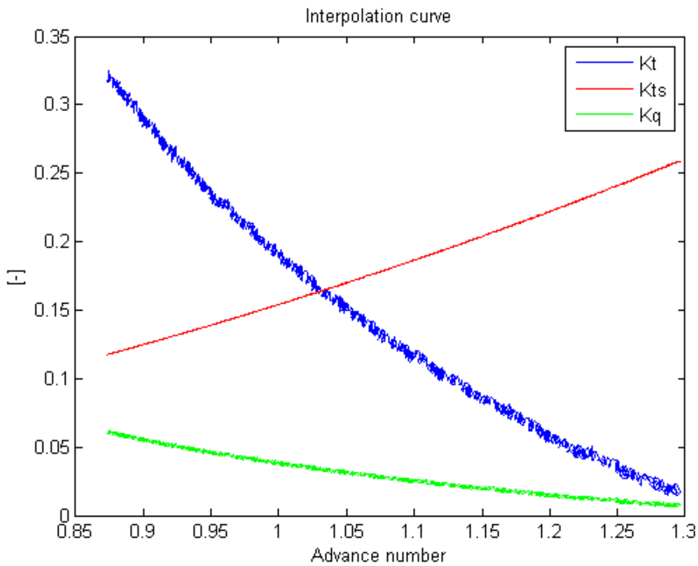


Figure 7.9: Interpolation curve for finding full scale propulsion point.

$$n_S[h_z] = \frac{V_S}{J^* \cdot D_S} \tag{7.16}$$

Further the torque coefficient for the self-propulsion point is obtained by finding the corresponding K_Q-value for J_* (see the green line in figure **??** for illustration). Based on the found K_Q value, the full scale torque can be calculated according to equation 7.17.

$$Q_S = K_Q \cdot \rho_S \cdot n_S^2 \cdot D_S^5 \tag{7.17}$$

At this point all components in the powering prediction formula is known, and the estimate of delivered power can be made according to formula 7.18.

$$P_D = 2 \cdot \pi \cdot n_S \cdot Q_S \tag{7.18}$$

Please note that delivered power is not the same as brake power, a common value appearing in many powering predictions and full scale trials. To obtain the relationship between delivered power and brake power, equation 7.19 can be used.

$$P_B = \frac{P_D}{\eta_M} \tag{7.19}$$

Here η_M is the mechanical efficiency. If this is not known, a standard value of 0.97 can be used.

Chapter 8

Results for M2375J

The twin screw car freighter was tested in MARINTEKs facility over two days in January 2012. The principal hull data for the ship is given in table 3.1 and the towing schedule can be found in appendix A.1. The ship was tested with MARINTEKs standard procedure, in addition to the testes with the constant increment approach. One of the main targets with these tests was to see whether or not the increment value has an effect on the final results, in addition to look into the repeatability of the tests.

8.1 Full scale trials

The model tested in this part lead to the building of three sister ships, and all of them went through full scale trial upon delivery. Table 8.1 presents the main data of interest from the full scale trials. Based on the speed trail data, the speed range for the model tests were chosen to be between 18-23kn.

Table 8.1: Corrected speed trial data - M2375J

S1184			
Speed [kn]	**BHP** [kW]	**RPM**	**% Power**
17.92	7799	118.4	50 %
19.94	11370	134.4	65 %
21.2	13349	141.6	85 %
22.14	15005	147.1	100 %
S1185			
Speed [kn]	**BHP** [kW]	**RPM**	**% Power**
17.75	7742	117.8	50 %
20.32	11285	134.1	65 %
21.28	12696	139.7	85 %
22.41	14897	147.6	100 %
S1186			
Speed [kn]	**BHP** [kW]	**RPM**	**% Power**
20.08	11074	133.5	75 %
21.29	13194	141.7	85 %
22.32	15085	148	100 %

8.2 Extrapolation using the standard procedure

Before conducting the load varying tests, a full set of ordinary resistance, propulsion and open water tests were conducted according to MARINTEKs standard procedure (see chapter 6.1). This was done to ensure as equal conditions between the standard tests and the load varying tests as possible, so that errors connected to model hull condition, model loading and set up could be minimized. The main results from these tests are presented in table 8.2 and the power and rpm curve from ShipX in figure 8.1.

Table 8.2: Main results from ordinary tests - M2375J

V_S[kn]	F_N[-]	R_{TM}[N]	J_0[-]	t[-]	N[Rpm]	P_B[kW]
18.07	0.251	44.724	0.937	0.136	123.7	7783.2
19.08	0.265	49.938	0.935	0.145	131.4	9376.8
20.08	0.279	55.028	0.933	0.152	138.7	11029.1
21.10	0.293	60.690	0.932	0.155	146.1	12942.0
22.10	0.307	66.885	0.929	0.153	153.3	15015.8
23.12	0.321	74.507	0.921	0.162	161.7	17929.9

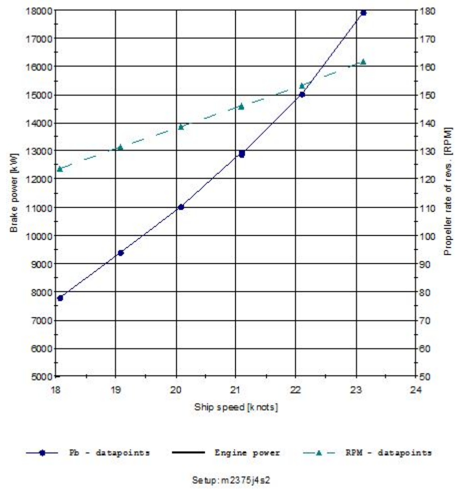


Figure 8.1: Predicted power and RPM from ShipX - M2375J

8.3 Extrapolation from load varying tests

The results from the load varying tests cover several different areas of interest, and will be presented according to these. First the increment value influence will be presented. Then the total load varying series will be compared to the speed trials and MARINTEKs method before the reliability of the method will be discussed by looking at the repeatability of the tests.

8.3.1 Increment influence

The increment value is further discussed in section 7.3.1. The increment value are an important parameter in the load varying tests, and there is therefore of great importance to find out how this value influences the results. For the ship in this part the increment tests were performed at a full scale speed of 22kn ($F_N = 0.306$), with increment values in the range 0.5-2.5 [$\frac{rev}{s}$]. The tests used in the analysis are presented in table 8.3. Note that all series are cut so that they cover exactly the same range of revolutions, starting at 10 [rps] and ending at 13.2 [rps].

The results from the analysis are presented in figures 8.2, 8.3 and 8.4, and in table 8.4.

From the results, no conclusion on the increment influence can be made. If we compare the results with Marintek's procedure it may look like an increasing increment value gives more uncertainty. However, the opposite is the case when comparing with the full scale trials.

Table 8.3: Series used in the increment analysis - M2375J

Series #	Speed	Increment	Start rev.	Stop rev.
1001	22	0.05	9.5	11.5
1002	22	0.05	11	13.2
1021	22	0.10	9.6	13.6
1018	22	0.15	10	15
1019	22	0.20	10	15
1020	22	0.25	10	14.6

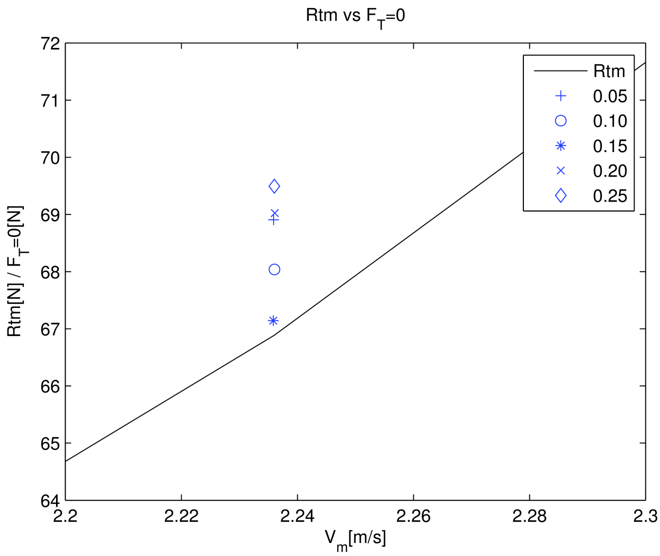


Figure 8.2: R_{TM} and $F_{T=0}$ for varying increment values - M2375J

Table 8.4: Results from increment analysis, compared with sea trials and standard tests - M2375J

Sea trials		N	145.70	P_B	14422.67
	0.05	0.10	0.15	0.20	0.25
N	152.84	152.13	151.36	151.09	151.90
Diff	4.90 %	4.41 %	3.88 %	3.70 %	4.25 %
P_B	14881.16	14630.07	14403.46	14193.54	14534.15
Diff	3.18 %	1.44 %	-0.13 %	-1.59 %	0.77 %
Marintek		N	152.96	P_B	15113.80
	0.05	0.10	0.15	0.20	0.25
N	152.84	152.13	151.36	151.09	151.90
Diff	-0.08 %	-0.55 %	-1.05 %	-1.22 %	-0.70 %
P_B	14881.16	14630.07	14403.46	14193.54	14534.15
Diff	-1.54 %	-3.20 %	-4.70 %	-6.09 %	-3.84 %

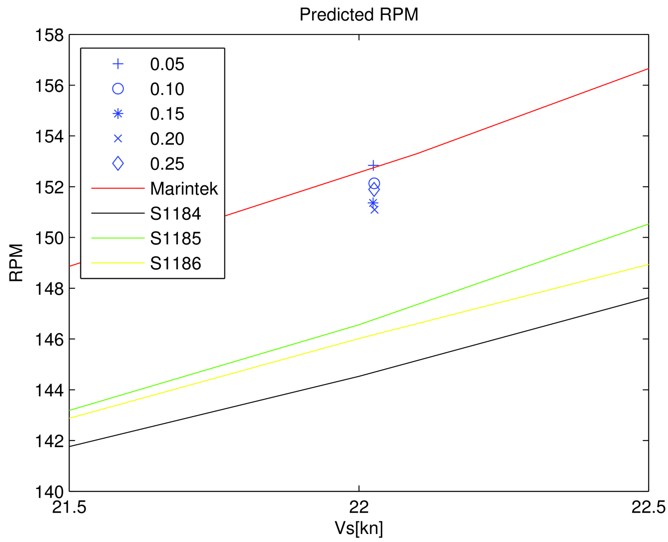


Figure 8.3: Propeller revolutions for varying increment values - M2375J

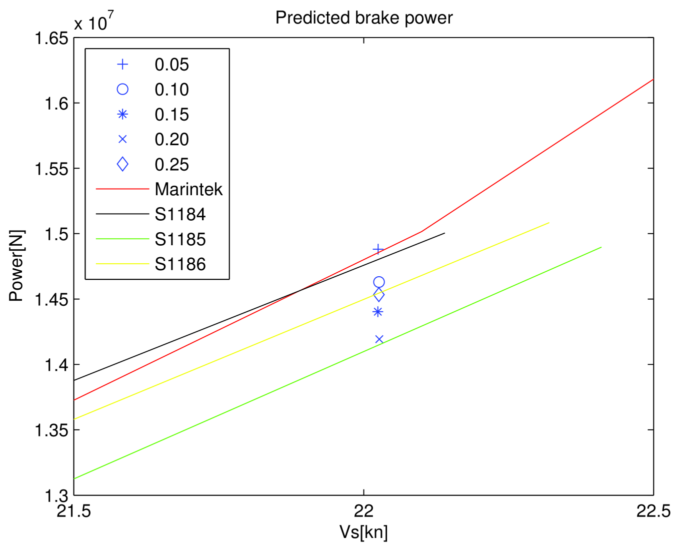


Figure 8.4: Brake power for the varying increment values - M2375J

Table 8.5: Results from load varying tests with increment=0.1 compared against full scale trial values and Marintek's results. Note that the gray cells values are partly based on extrapolated results, and should be used with care.

Load Varying						
Speed	18	19	20	21	22	23
$F_{T=0}$	45,50	50,57	56,17	61,60	67,89	75,75
N	121,39	129,32	137,00	143,98	152,08	159,72
P_B	7252,42	8836,97	10524,04	12277,42	14657,31	17162,17
Marintek						
Speed	18	19	20	21	22	23
R_{TM}	44,72	49,94	55,03	60,69	66,89	74,51
Diff	1,74 %	1,27 %	2,08 %	1,50 %	1,50 %	1,67 %
N	122,92	130,43	137,94	145,45	152,97	160,48
Diff	-1,24 %	-0,85 %	-0,68 %	-1,01 %	-0,58 %	-0,47 %
P_B	7172,20	9157,60	11143,00	13128,40	15113,80	17099,20
Diff	1,12 %	-3,50 %	-5,55 %	-6,48 %	-3,02 %	0,37 %
Average full scale						
Speed	18	19	20	21	22	23
N	119,46	126,08	132,69	139,31	145,93	152,54
Diff	1,62 %	2,57 %	3,25 %	3,35 %	4,22 %	4,71 %
P_B	7956,40	9564,20	11172,00	12779,80	14387,60	15995,40
Diff	-8,85 %	-7,60 %	-5,80 %	-3,93 %	1,87 %	7,29 %

8.3.2 Fixed increment analysis

The results indicate that the load varying procedure gives a good, but somewhat conservative, estimation of required power for this ship. This means that the method under-predicts the power slightly, but actually is quite close to the values from one of the full scale trials. The predicted rpm is close to MARINTEKs predicted rpm, and follows this closely. The results therefore look reasonable.

8.3.3 Repeatability of the tests

The repeatability, hence the accuracy, of the tests are of great importance when comparing results from the tests. By determining the accuracy in the repeated dataset it is possible to say something about whether the natural scatter in the data is under control, or if it is skewing the final results severely. The following section will look into the repeated test for model M2375J.

Model M2890C had three repeated tests of series 1000, making it a total of four identical tests (series 1000,1021,1023 and 1025) as input to these calculations. The tests were conducted at 22kn with an increment value of $0.1\frac{rev}{sec}$. The range of

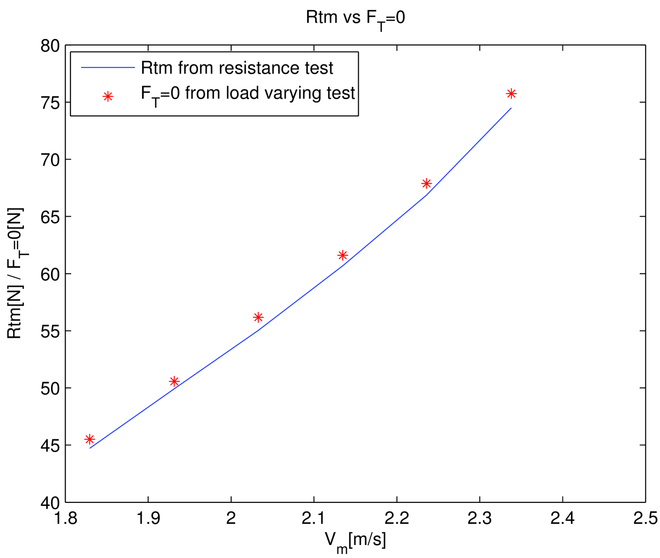


Figure 8.5: Results for calculated $F_{T=0}$ from load varying tests, compared with R_{TM} from ordinary tests

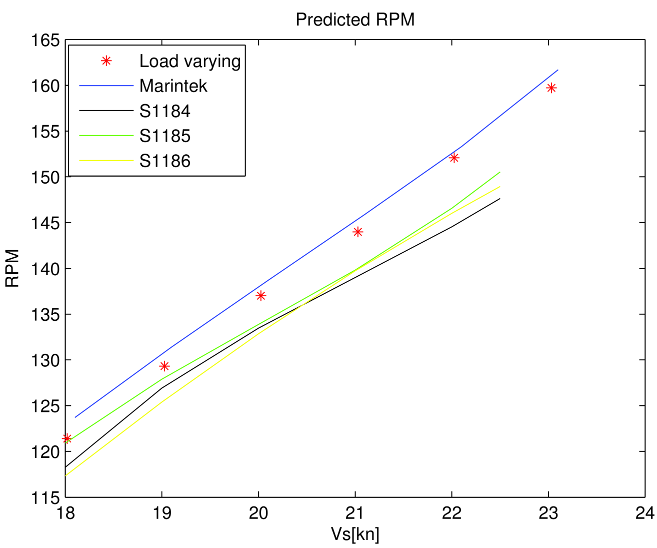


Figure 8.6: Results from calculated N_S from load varying tests, compared to Marintek's results and full scale trials.

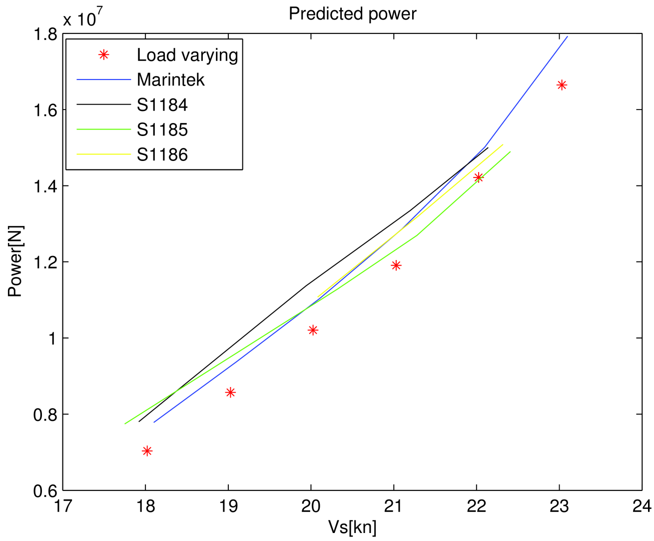


Figure 8.7: Results from calculated P_B from load varying tests, compared to Marintek's results and full scale trials.

revolutions covered were from 9.6rps to 13.5rps. Some important results from the analysis of these runs are presented in table 8.6 and in figures 8.8 and 8.9.

Table 8.6: Results from repeated tests with M2375J

Run	1000	1021	1023	1025
R_{TM}	68.69	67.87	67.50	67.47
n	152.94	152.09	152.26	151.95
P_B	14927.82	14651.90	14713.13	14614.04

To obtain useful information about the accuracy of the tests, the mean (equation 8.1) and standard deviation (equations 8.2 and 8.3) was calculated according to the formulas given by (Walpole et al., 2007).

$$\overline{x} = \sum \frac{x_i}{n} = \frac{x_1 + x_2 + \ldots + x_n}{n} \tag{8.1}$$

$$s^2 = \sum \frac{(x_i - \overline{x})^2}{(n-1)} \tag{8.2}$$

$$s = \sqrt{s^2} \tag{8.3}$$

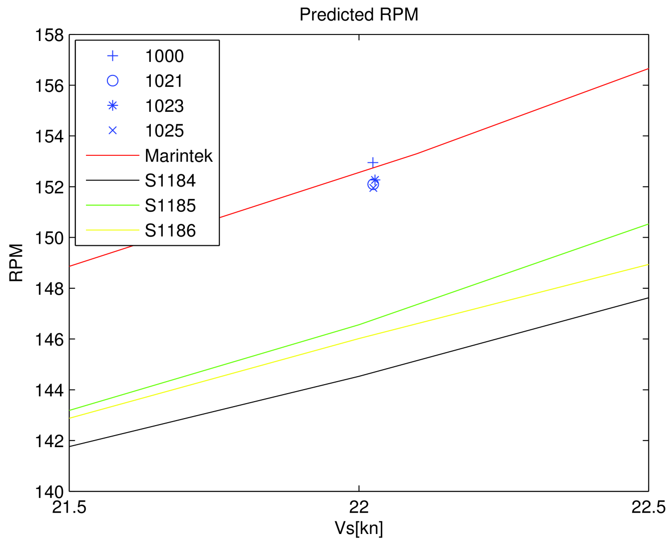


Figure 8.8: Propeller revolutions for the repeated tests - M2375J

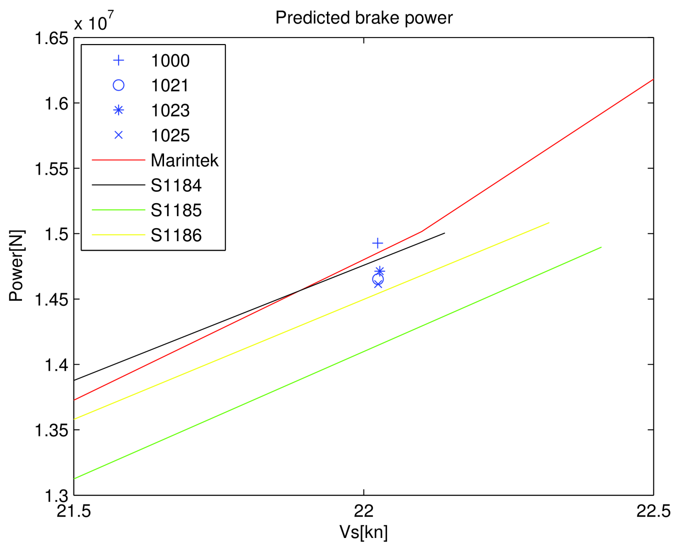


Figure 8.9: Brake power for the repeated tests - M2375J

Table 8.7: Standard deviation from repeated tests with M2375J

	Mean	Std	Std %
R_{TM}	67.88	0.57	0.84 %
n	152.31	0.44	0.29 %
P_B	14726.72	140.14	0.95 %

The results from these calculations are presented in table 8.7. As seen from the results, the standard deviation is small. This indicates good repeatability of the tests.

Chapter 9

Results for M2890C

The single screw full body bulk carrier was tested in February 2012 at MARINTEKs facility in Trondheim, Norway. The tank schedule for the load varying tests are given in appendix A.2, and principal hull data for the ship can be found in table 3.2. In addition to the load varying tests described in the towing schedule, a set of standard resistance, propulsion and open water tests were conducted. The main target with these tests were to look at the increment effect on the powering prediction, but during the tests some issues appeared that made this a secondary target during the testing.

9.1 Full scale trials

The model tests lead to the building of one ship, which went through full scale speed trials in November 2011. The speed trial data are given in table 9.1. These data are used as a reference point for the load varying tests.

Table 9.1: Corrected results from speed trials - M2890C

Speed[kn]	BHP[kW]	RPM	% Power
13.92	12309.424	63.00	% 25
15.43	15429.455	67.99	% 50
16.27	19068.498	72.47	% 75
16.65	22430.986	75.99	NOR
16.87	24777.164	78.39	MCR

9.2 Extrapolation using the standard procedure

Before conducting the load varying tests, a full set of ordinary resistance, propulsion and open water tests were conducted according to MARINTEKs standard procedure (see chapter 6.1). This was done to ensure as equal conditions between the standard tests and the load varying tests as possible, so that errors connected to model hull condition, model loading and set up could be minimized. The main results from these tests are presented in table 9.2 and the power and rpm curve from ShipX in figure 9.1. Note that only a minor difference between the extrapolation using the low rate of revolution open water test and the new open water test were found. Hence this was not the reason for the differences between the full scale trials and MARINTEKs prediction. There is more likely that this is related to the wake scaling.

Table 9.2: Main results from ordinary tests - M2890C

V_S[kn]	F_N[-]	R_{TM}[N]	J_0[-]	t[-]	N[Rpm]	P_B[kW]
13	0.116	18.662	0.368	0.210	56.8	10186.2
14	0.125	21.431	0.366	0.214	61.1	12679.8
15	0.133	24.483	0.364	0.219	65.8	15842.2
16	0.142	27.930	0.363	0.222	70.8	19923.9
17	0.151	31.914	0.361	0.219	76.2	25196.9

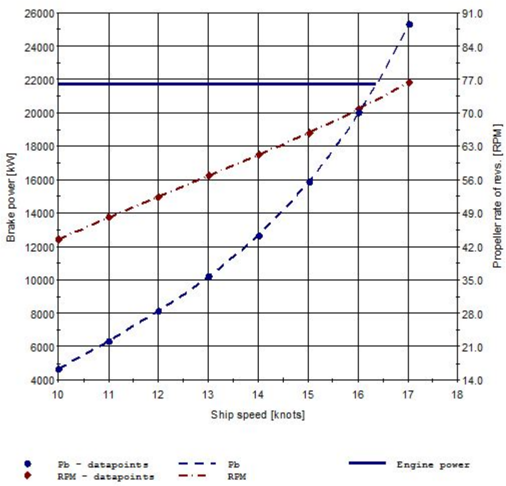


Figure 9.1: Predicted power and RPM from ShipX - M2890C

9.3 Extrapolation from load varying tests

The load varying tests conducted with M2890C have some differences from the other two set of tests, as some issues regarding oscillations in the data appeared during testing. In practice these issues limited the increment value that could be used during the tests, making it necessary to divide the thrust range into several separate runs to cover the entire range. When the limit on the increment value were set so low, there was no point in making an increment analysis from the data, as there were difficult enough to obtain useful data for one of the increment values. Instead a decision to look at how the different thrust regions (low, medium and high) influences the load varying results were made. In addition some series with the constant revolution approach were conducted, to see whether or not this method gives better results when these effects appear. The different series used to obtain the results are presented in table 9.3, and all series are cut to ensure proper results when merging them to obtain combination series.

Table 9.3: Different series used to obtain results for M2890C. The indicated revolution range is for the cutted series.

Low thrust range				
Series	Speed	Increment	Start rev.	Stop rev.
2021	13	0.025	2,6	4,7
2002	14	0.025	3,2	5,2
2016	15	0.025	3,2	5,4
2004	16	0.025	4	6
2018	17	0.025	3,8	5,7
Medium thrust range				
Series	Speed	Increment	Start rev.	Stop rev.
	13	None		
2024	14	0.025	5,2	7,3
2029	15	0.025	5,6	7,5
2023	16	0.025	6,1	8
2031	17	0.025	6,2	7,9
High thrust range				
Series	Speed	Increment	Start rev.	Stop rev.
2027	13	0.025	6,1	8,5
2010	14	0.025	7,3	9,5
2017	15	0.025	7,5	9,1
2009	16	0.025	8	9,4
2019	17	0.025	8	9,8

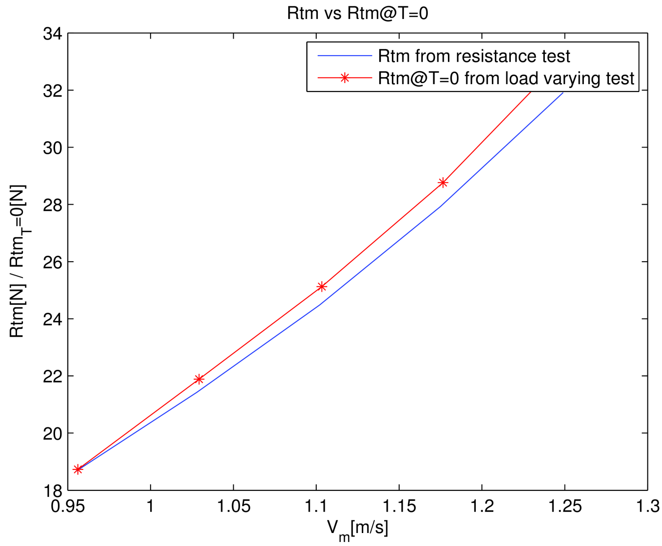
9.3.1 Low thrust range

The range of the low thrust analysis is presented in table 9.3. The results are presented in table 9.4 and in figures 9.2, 9.3 and 9.4.

As seen from the figures, and the table, the estimation of power and propeller revolutions are not good when only using the low thrust range for extrapolation. However, the estimation of $F_{T=0}$ is acceptable with a difference around 2 %, the same as Holtrop (2001) indicated. The reason for this is that the low thrust range is close to the zero thrust point, hence a good estimation of this point can be made. The same is not the case for the power and revolution estimation, as information from two of the most critical parameters (full thrust point $F_{T=100}$ and propeller characteristics around the propulsion point) are heavily extrapolated and therefore not a good representation of the reality. Therefore a low thrust run should not be used for extrapolation alone.

Table 9.4: Results from extrapolation the low thrust range - M2890C

Load varying				
Speed	14	15	16	17
$F_{T=0}$	21,88345108	25,12396344	28,76502249	33,20467013
N	64,7229768	67,60540307	68,28701424	71,12516697
P_B	13529,21467	14479,87271	16989,9031	17013,12058
Full scale				
Speed	14	15	16	17
N	62,3844	67,4079	72,4314	77,4549
Diff	3,75 %	0,29 %	-5,72 %	-8,17 %
P_B	12347,6	13517,5	17937,6	25607,9
Diff	9,57 %	7,12 %	-5,28 %	-33,56 %
Marintek				
Speed	14	15	16	17
R_{TM}	21,43	24,48	27,93	31,914
Diff	2,12 %	2,63 %	2,99 %	4,04 %
N	61,1	65,8	70,8	76,2
Diff	5,93 %	2,74 %	-3,55 %	-6,66 %
P_B	12679,8	15842,2	19926,9	25196,9
Diff	6,70 %	-8,60 %	-14,74 %	-32,48 %



![h]

Figure 9.2: R_{TM} against $F_{T=0}$ for low thrust range - M2890

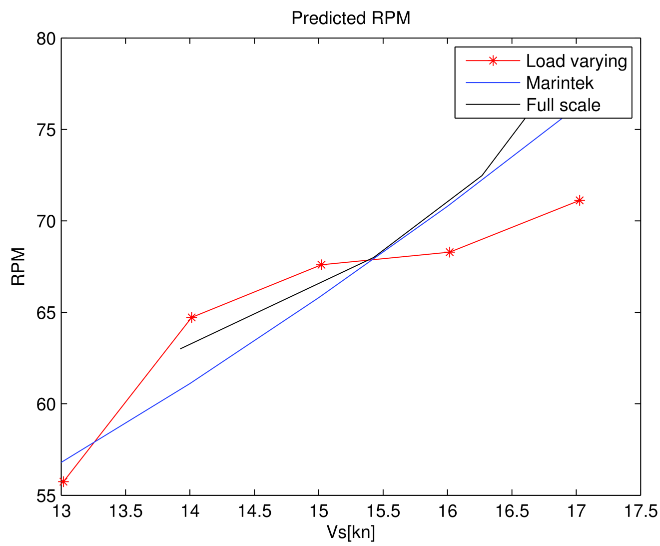


Figure 9.3: Full scale propeller revolutions for low thrust range - M2890

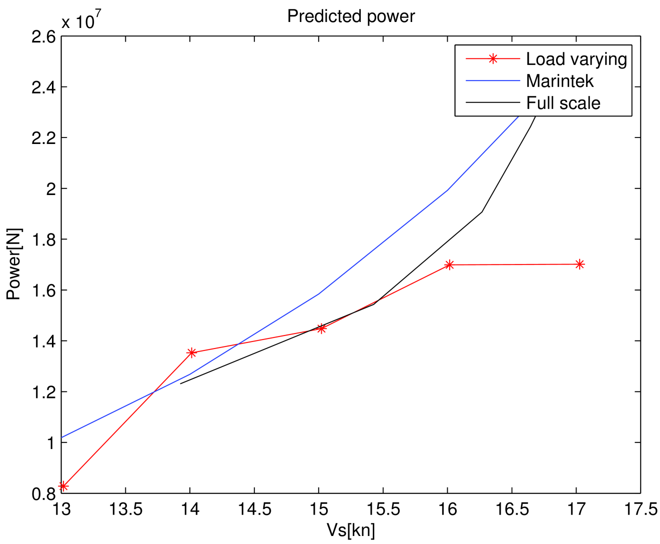


Figure 9.4: Brake power for low thrust range - M2890

9.3.2 Medium thrust range

The results from extrapolation using the medium thrust range gives reasonable estimations, as seen in figure 9.5 and table 9.5. This is most likely due to the fact that the self propulsion point can be found in this area. In addition it should give usable estimations of the zero and full thrust points, as the extrapolation to reach these points are within a normal validity range.

Table 9.5: Results from extrapolation using medium thrust range - M2890

Load varying				
Speed	14	15	16	17
$F_{T=0}$	22,44	26,29	29,69	34,05
N	58,82	62,85	68,59	73,31
P_B	13433,47837	16236,05875	21148,11581	25450,85276
Full scale				
Speed	14	15	16	17
N	62,3844	67,4079	72,4314	77,4549
Diff	-5,71 %	-6,76 %	-5,30 %	-5,35 %
P_B	12347,6	13517,5	17937,6	25607,9
Diff	8,79 %	20,11 %	17,90 %	-0,61 %
Marintek				
Speed	14	15	16	17
R_{TM}	21,43	24,48	27,93	31,914
Diff	4,71 %	7,39 %	6,30 %	6,69 %
N	61,1	65,8	70,8	76,2
Diff	-3,73 %	-4,48 %	-3,12 %	-3,79 %
P_B	12679,8	15842,2	19926,9	25196,9
Diff	5,94 %	2,49 %	6,13 %	1,01 %

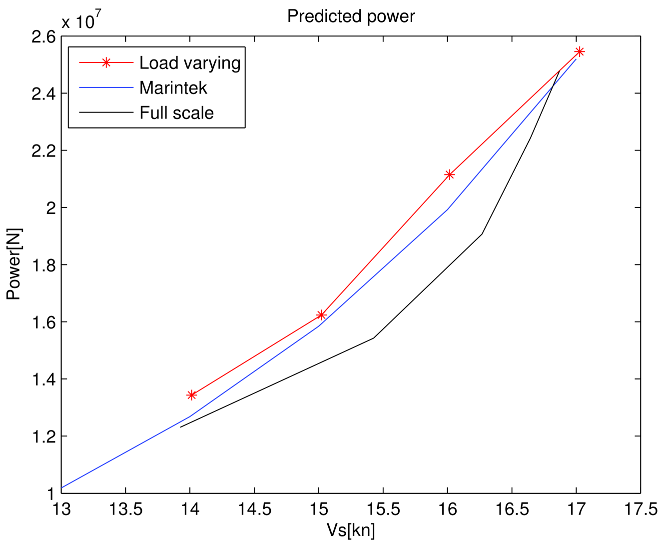


Figure 9.5: Brake power for medium thrust range - M2890

9.3.3 High thrust range

The results from the high thrust range extrapolation is presented in table 9.6 and in figures 9.6 and 9.7. As seen from the $F_{T=0}$ values, the high thrust range struggles to give a good estimation of this point. This is again due to the extrapolation far out from the data in the series. This can especially be seen in figure 9.6 at 16kn, where the zero thrust point suddenly is about 15 % larger than the corresponding R_{TM}-value, where all other values are close. It is therefore not recommended to conduct a powering estimation based on high thrust values only.

Table 9.6: Results from extrapolation using high thrust range - M2890

Load varying				
Speed	14	15	16	17
$F_{T=0}$	22,71	25,36	32,04	32,63
N	58,75	63,44	67,27	73,98
P_B	13791,38317	16791,57645	19962,83771	26649,27068
Full scale				
Speed	14	15	16	17
N	62,3844	67,4079	72,4314	77,4549
Diff	-5,83 %	-5,89 %	-7,13 %	-4,49 %
P_B	12347,6	13517,5	17937,6	25607,9
Diff	11,69 %	24,22 %	11,29 %	4,07 %
Marintek				
Speed	14	15	16	17
R_{TM}	21,43	24,48	27,93	31,914
Diff	5,97 %	3,59 %	14,72 %	2,24 %
N	61,1	65,8	70,8	76,2
Diff	-3,85 %	-3,59 %	-4,99 %	-2,91 %
P_B	12679,8	15842,2	19926,9	25196,9
Diff	8,77 %	5,99 %	0,18 %	5,76 %

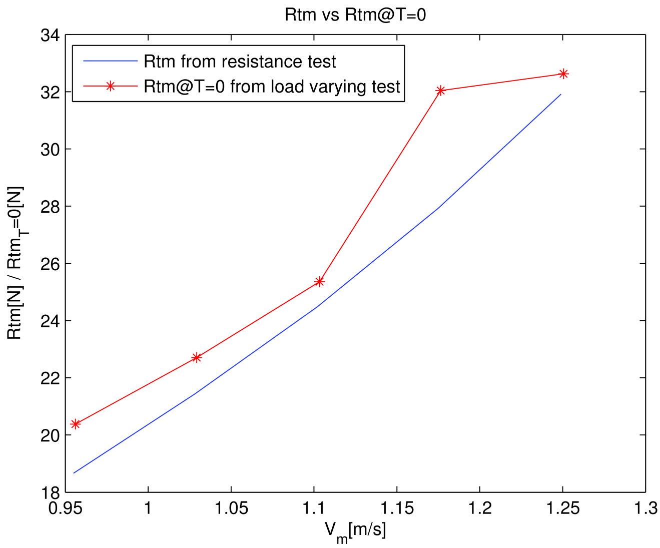


Figure 9.6: R_{TM} against $F_{T=0}$ for high thrust range - M2890

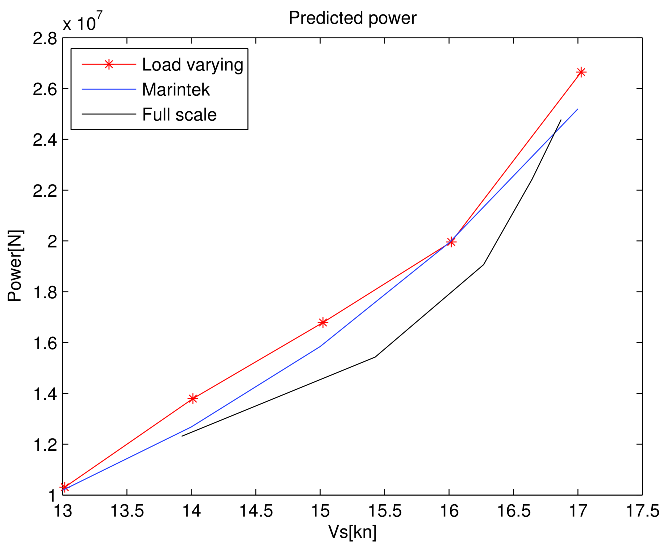


Figure 9.7: Brake power for high thrust range - M2890

9.3.4 Combined thrust range

As only the medium thrust range extrapolation gives reasonable answers, it would be interesting to see if a combination of these thrust ranges will give a better estimation. Combinations of low/medium, medium/high and low/high can also be calculated in the program, but the results from these combinations does not give any significant changes from what that is already presented. This is most likely due to where the "weigh" of the thrust are placed in the thrust range covered by the tests. For instance a low/medium combination will give good approximation of the zero thrust point and the self propulsion point, but less good approximation of $F_{T=100}$. The trend lines will also be effected by this, as they will have their "weight" in the lower thrust range and therefore estimate this part better.

The results presented in this section are combinations of low, medium and high thrust ranges, with no overlap between them. This should therefore, in an ideal world, be similar to conducting one long run from low to high thrust. This is similar to what was done with the other ships tested. The results are presented in table 9.7 and figure 9.8. As seen from the figure, the results from the combination series are following the same trend as the results from Marintek. However the results are further away that for instance the medium thrust range results. This indicates that the combination of series should be done with care, and avoided if possible. The next section will look into the results from the fixed revolution approach.

Table 9.7: Results from extrapolation combining the entire thrust range - M2890

Load varying				
Speed	14	15	16	17
$F_{T=0}$	22,34	25,5	28,89	33,19
N	58,82	63,38	69,23	73,85
P_B	14771,75019	17694,34656	22472,75491	26946,49145
Full scale				
Speed	14	15	16	17
N	62,3844	67,4079	72,4314	77,4549
Diff	-5,71 %	-5,98 %	-4,42 %	-4,65 %
P_B	12347,6	13517,5	17937,6	25607,9
Diff	19,63 %	30,90 %	25,28 %	5,23 %
Marintek				
Speed	14	15	16	17
R_{TM}	21,43	24,48	27,93	31,914
Diff	4,25 %	4,17 %	3,44 %	4,00 %
N	61,1	65,8	70,8	76,2
Diff	-3,73 %	-3,68 %	-2,22 %	-3,08 %
P_B	12679,8	15842,2	19926,9	25196,9
Diff	16,50 %	11,69 %	12,78 %	6,94 %

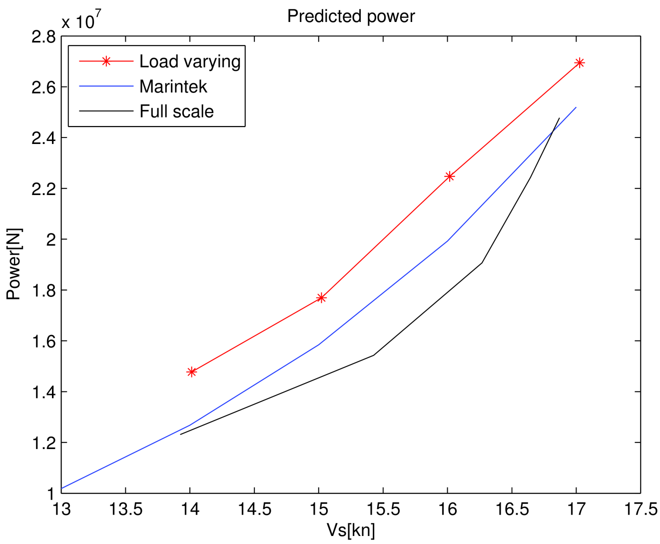


Figure 9.8: Brake power for combined thrust range - M2890

9.3.5 Fixed revolution tests

Mainly for control purposes, two sets of fixed revolution tests were also conducted at 14 and 16kn. The initial control during the tests indicated that the fixed revolution estimation and the fixed increment estimation should give approximately the same results. Figure 9.9 and table 9.8 presents the results.

The results from the fixed revolution approach corresponds well to the results from the combined thrust range results, but does not correspond well to MARINTEKs method or full scale trials. Based on this it seems that the medium thrust range gives the best results. This is not surprising as it covers the self propulsion point, in addition to weight the trendlines good.

Table 9.8: cap

Load varying		
Speed	14	16
$F_{T=0}$	21,93	28,56
N	58,84	68,39
P_B	15267,81912	22968,11785
Full scale		
Speed	14	16
N	62,3844	72,4314
Diff	-5,68 %	-5,58 %
P_B	12347,6	17937,6
Diff	23,65 %	28,04 %
Marintek		
Speed	14	16
R_{TM}	21,43	27,93
Diff	2,33 %	2,26 %
N	61,1	70,8
Diff	-3,70 %	-3,40 %
P_B	12679,8	19926,9
Diff	20,41 %	15,26 %

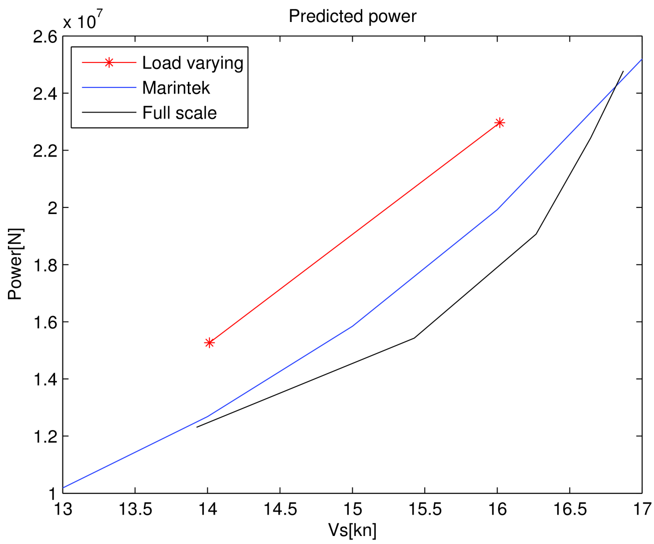


Figure 9.9: Brake power for fixed revolution runs - M2890

Chapter 10

Results for M3025A

M3025A is an offshore vessel with a ducted propeller. The ship has not been constructed yet, and there is therefore restrictions on the information about it.

10.1 Full scale trials

The vessel has not yet been constructed, hence no full scale trials are available at this point. This means that no iteration to find the true C_A can be done. It also introduces an uncertainty in the results, as the only comparable data are the ordinary set of resistance, propulsion and open water tests. If there are large scatter between the load varying tests and the ordinary tests, there will be difficult to find out which of the datasets that are closest to representing the true values. Therefore the results in this section should be verified against speed trials when/if the ship is built.

10.2 Extrapolation using the standard procedure

Before conduction the load varying tests, a control run to verify the previously recorded series of conventional tests were run at 13kn. This was less than 0.5 % deviation, which is well inside the expected value. The previously recorded series is therefore assumed representative for the model in its present state and will be used for verification purposes. The main results are presented in table 10.1 and the power and rpm curve from ShipX in figure 10.1

Table 10.1: Main results from ordinary tests - M3025A

V_S[kn]	F_N[-]	R_{TM}[N]	J_0[-]	t[-]	N[Rpm]	P_B[kW]
12.1	0.228	45.19	0.554	0.232	133.3	1625.4
13.1	0.247	53.89	0.548	0.232	146.3	2160.0
14.1	0.266	72.63	0.490	0.250	170.1	3533.4

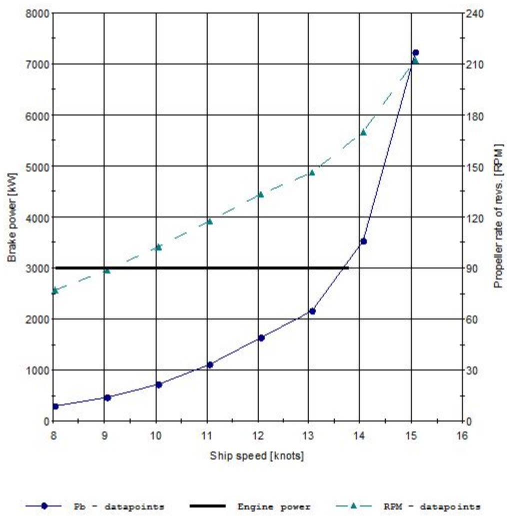


Figure 10.1: Predicted power and RPM from ShipX - M3025A

10.3 Extrapolation from load varying tests

10.3.1 Increment influence

The increment influence has been tested at 13kn with increment values between 0.1 and 0.25 rev/s. The results are presented in table 10.2 and figure **??**. From the results, no indication of a trend regarding the increment value is visible.

Table 10.2: Results from varying increment test - M3025A

Increment	0.1	0.15	0.2	0.25
$F_{T=0}$	55,68	55,65	55,78	55,33
Diff	3,34 %	3,28 %	3,51 %	2,68 %
N	147,30	148,81	148,01	148,12
Diff	0,69 %	1,72 %	1,17 %	1,25 %
P_B	2242,03	2325,61	2302,83	2255,47
Diff	3,80 %	7,67 %	6,61 %	4,42 %

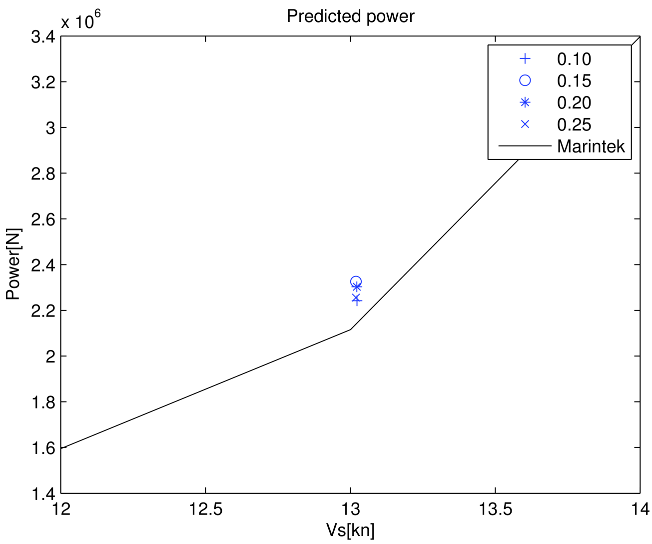


Figure 10.2: Brake power from varying increment analysis - M3025A

10.3.2 Fixed increment analysis

The fixed increment analysis were conducted with an increment of 0.15 rev/s at 12-14kn. The results are presented in table **??** and figure 10.3. As seen from the table the powering prediction has some deviation for the low speed values. At the highest speed tested the deviation is only 0.82 %, which is very good.

Table 10.3: Results from fixed increment analysis - M3025A

Load varying			
Speed	12	13	14
$F_{T=0}$	46,94	55,65	74,02
N	134,66	148,81	173,59
P_B	1785,41	2325,61	3562,39
Marintek			
Speed	12,00	13	14
R_{TM}	45,19	53,89	72,63
Diff	3,87 %	3,28 %	1,92 %
N	133,3	146,3	170,1
Diff	1,03 %	1,72 %	2,05 %
P_B	1625,4	2160	3533,4
Diff	9,84 %	7,67 %	0,82 %

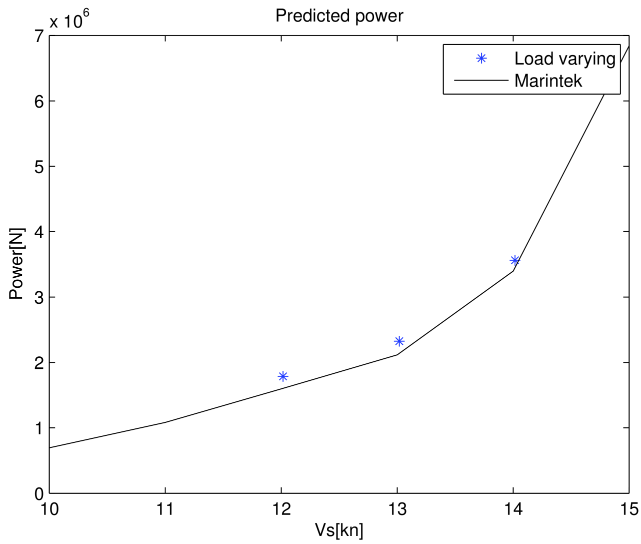


Figure 10.3: Brake power from fixed increment analysis - M3025A

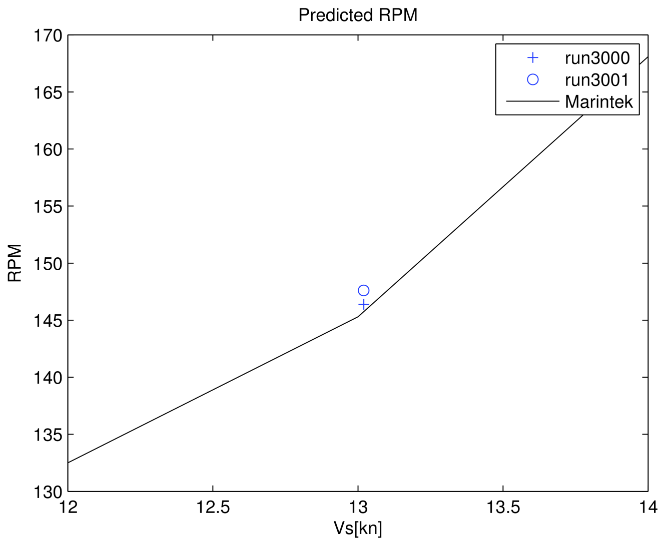


Figure 10.4: Propeller revolutions for the repeated tests - M3025A

10.3.3 Repeatability of the tests

The repeatability, hence the accuracy, of the tests are of great importance when comparing results from the tests. By determining the accuracy in the repeated dataset it is possible to say something about whether the natural scatter in the data are under control, or if it is skewing the final results severely. The following section will look into the repeated test for model M3025A.

Model M3025A had one repeated test of series 3000, making it a total of two tests for this analysis. This is to few to obtain a reliable repeatability calculation, but the two tests will be compared to at least give an indication on how well the tests with this model repeats. Another weakness with the repeatability in this test series, is that the two tests were conducted staight after each other. The reason for the few tests is that it only was conducted a total of 7 load varying runs with this model. The repeatability tests were conducted at 13kn with an increment value of $0.1\frac{rev}{sec}$. The range of revolutions covered were from $7rps$ to $10rps$. The results are presented in table 10.4 and in figures 10.4 and 10.5.

Table 10.4: Results from repeated tests with M3025A

Run	3000	3001
R_{TM}	55.78	55.59
n	146.39	147.60
P_B	2218.5	2262.3

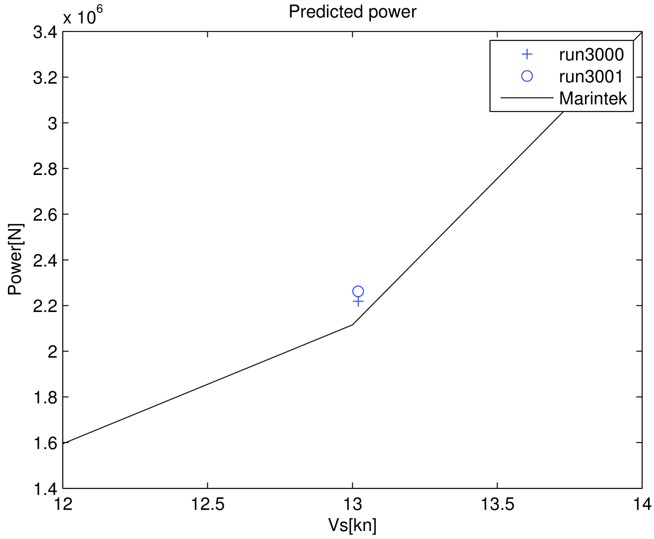


Figure 10.5: Brake power for the repeated tests - M3025A

Table 10.5: Standard deviation from repeated tests with M2375J

	Mean	Std	Std %
R_{TM}	55.685	0.134	0.24 %
n	146.995	0.855	0.58%
P_B	2240.4	30.971	1.38 %

To obtain useful information about the accuracy of the tests, the mean (equation 8.1) and standard deviation (equations 8.2 and 8.3) was calculated according to the formulas given by (Walpole et al., 2007). The results from these calculations are presented in table 10.5. As seen from the results, the standard deviation is small. This indicates good repeatability of the tests.

Chapter 11

Conclusion

This thesis has looked at several different aspects of the load varying self-propulsion tests. In the introduction some hypothesis was presented, and based on the results from the analysis these will be commented. The first hypothesis were:

The results from the load varying extrapolation procedure inside the expected range, when comparing against the results from MARINTEKs standard procedure and full scale speed trials.

Based on the results from M2375J and M3025A this hypothesis is confirmed, but results for M2890C is outside the expected range making it questionable. This can be due to the oscillation issue that appeared during the tests, but most likely it it related to wake scaling problem. M2890C is the only conventional single screw vessel, and is therefore the vessel that is most likely to have significant wake contributions. As discussed in the section about wake scaling, no wake scaling has been applied during the analysis as there are not enough information from a load varying test to calculate this value. The wake scaling issue needs to be looked further into, and until a good solution is found it is difficult to conduct extrapolation based on load varying tests only.

Repeated tests will give the same results. From the repeated tests with M2375J and M3025A this seems true. The standard deviation between the equal runs are about 1 %, which is good repeatability. This indicates that if similar tests are conducted, a result within +/- 1 % is expected. This means that the method is stable.

he increment value used in the tests effects the results from the extrapolation procedure. If true, there is a certain limit where the results are starting to get severely skewed.

This has not been proven the case with neither of the tests. There has not been found any link between the increment value and the results, if we look away from the oscillation problem with M2890C. It seems that in a standard length tank, the increment value will limit itself before it starts to create scatter in the results.

The chosen thrust range influences the results, when the thrust range of interest cannot be covered by one test. This is true, the chosen thrust range does influence the results. It seems favourable to choose a thrust range around the self propulsion point, with no pronounced weight to either side. This should however be investigated more.

Based on this the load varying method seems a promising and reliable powering performance method if a satisfying solution to the wake scaling problem is found. It should ideally be possible to determine this value based on information from correlation databases or the self propulsion test itself. I therefore recommend to do further work regarding the wake scaling issue.

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Appendices

Appendix A

Towing schedule

A.1 M2375J

LOGG FOR FORSØKSKJØRING MT-K4-S110 Rev.00			**Prosjektnr.: 846002.30** **Tittel: Slep, prop, ,2375j**
Dato	**Kl.**	**Test nr.**	**Kommentarer**
30.01.12			M2375j2_slep_1 tradisjonell slep
			M2375j2_prop_1 tradisjonell prop
		1000	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.1 rps/v Startturtall 9.5rps
		1001	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.05 rps/v Startturtall 9.5rps, slutt ca 11,5rps
		1002	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.05 rps/v Startturtall 11.5rps, slutt ca 13rps
31.01.12	08.55	1010	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0,15rev/v Startturtall 7rps. Testtur, resultater skal ikke brukes (babor trim)
	09.10	1011	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0,15rev/v Startturtall 7rps, slutt ca 13,5rps (babor trim)
	9.35	1012	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 7rps, slutt ca 10rps. Konst på 10rps siste del.
	9.55	1013	Kjørt 18kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 8rps, slutt ca 12rps. Konst på 12rps siste del.
	10.20	1014	Kjørt 19kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 8,5rps, slutt ca 13rps. Konst på 13rps siste del.
	10.35	1015	Kjørt 20kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 8,5rps, slutt ca 12rps. Konst på 8,5rps i starten del.
	10.55	1016	Kjørt 21kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 9,5rps, slutt ca 12,5rps. Konst på 12,5rps siste del.
	11.15	1017	Kjørt 23kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 10rps, slutt ca 14,5rps. Konst på 14,5rps i siste del.
	12.20	1018	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0,15rev/v Startturtall 10rps, slutt ca 15rps.
	12.40	1019	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0,2rev/v Startturtall 10rps, slutt ca 15rps.
	13.00	1020	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0,25rev/v Startturtall 10rps, slutt ca 14,5rps.
	13.20	1021	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.1 rps/v Startturtall 9.5rps (Repetisjon av test nr 1000)
	13.40	1022	Kjørt 21kn fastholdt, uten towrope, med stigningstall 0,1rev/v Startturtall 11rps, slutt ca 14rps.
	14.00	1023	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.1 rps/v Startturtall 9.5rps (Repetisjon av test nr 1000)
	14.20	1024	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.1 rps/v

LOGG FOR FORSØKSKJØRING MT-K4-S110 Rev.00			**Prosjektnr.: 846002.30**
			Tittel: Slep, prop, ,2375j
Dato	**Kl.**	**Test nr.**	**Kommentarer**
			Startturtall 12rps, slutt 13,5. Konst på 12 og 13,5.
31.01.12	14.35	1025	Kjørt 22kn fastholdt, uten towrope, med stigningstall 0.1 rps/v Startturtall 9.5rps (Repetisjon av test nr 1000)
	14.50	1026	Kjørt 18kn fastholdt, uten towrope, med stigningstall 0,05rev/v Startturtall 10rps, slutt ca 12rps.
	15.05	1027	Kjørt 19kn fastholdt, uten towrope, med stigningstall 0,05rev/v Startturtall 10rps, slutt ca 12rps.
	15.20	1028	Kjørt 20kn fastholdt, uten towrope, med stigningstall 0,05rev/v Startturtall 11rps, slutt ca 13rps.

A.2 M2890C

LOGG FOR FORSØKSKJØRING MT-K4-S110 Rev.00			**Prosjektnr.:** **846002.40** **Tittel:** **Slep, prop, ,2890c**
Dato	**Kl.**	**Test nr.**	**Kommentarer**
16.02.12			slep
			prop
		2000	Kjørt 14kn fastholdt, uten towrope, med stigningstall 0.1 rps/v Startturtall 4rps (konst),rampe til 6rps, konst 6rps, rampe til 8 rps, konst 8rps
		2001	Kjørt 14kn fastholdt, uten towrope, med stigningstall 0.05 rps/v Startturtall 3rps, slutt ca 7,5rps
		2002	Kjørt 14kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 3rps, slutt ca 5,3rps
17.02.12	8.30	2003	Kjørt 16kn fastholdt, uten towrope, med stigningstall 0.2 rps/v Startturtall 4,5rps, slutt ca 9rps, ned igjen til 4,5rps (Dagens første)
	8.45	2004	Kjørt 16kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 4rps, slutt ca 6,5rps
	9.00	2005	Kjørt 16kn fastholdt, uten towrope. Fixed rpm 3,5
	9.15	2006	Kjørt 16kn fastholdt, uten towrope. Fixed rpm 5,5
	9.30	2007	Kjørt 16kn fastholdt, uten towrope. Fixed rpm 7,5
	9.45	2008	Kjørt 16kn fastholdt, uten towrope. Fixed rpm 9,5
	10.00	2009	Kjørt 16kn fastholdt, uten towrope, med stigningstall 0,025rps/v Startturtall 7,5rps, slutt ca 9,5rps
	10.15	2010	Kjørt 14kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 7rps, slutt ca 9rps
	10.30	2011	Kjørt 14kn fastholdt, uten towrope. Fixed rpm 3
	10.45	2012	Kjørt 14kn fastholdt, uten towrope. Fixed rpm 5
	11.00	2013	Kjørt 14kn fastholdt, uten towrope. Fixed rpm 7
	11.10	2014	Kjørt 14kn fastholdt, uten towrope. Fixed rpm 9
	11.25	2015	Retest av 2002
	12.20	2016	Kjørt 15kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 3,2rps, slutt ca 5,7rps
	12.35	2017	Kjørt 15kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 6,8rps, slutt ca 9,2rps
	12.50	2018	Kjørt 17kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 3,7rps, slutt ca 6rps
		2019	Kjørt 17kn fastholdt, uten towrope, med stigningstall 0.025 rps/v

LOGG FOR FORSØKSKJØRING MT-K4-S110 Rev.00			**Prosjektnr.: 846002.40** **Tittel: Slep, prop, ,2890c**
Dato	**Kl.**	**Test nr.**	**Kommentarer**
			Startturtall 7,8rps, slutt ca 10rps
		2020	Kjørt 17kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 2,5rps, slutt ca 6rps (Feil hastighet, skulle vært 13kn)
		2021	Kjørt 13kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 2,5rps, slutt ca 5rps
		2022	Kjørt 13kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 5,8rps, slutt ca 8rps
	13.55	2023	Kjørt 16kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 6rps, slutt ca 8,5rps
	14.10	2024	Kjørt 14kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 5,5rps, slutt ca 8rps
	1430	2025	Retest 2023 (usikkerhetsanalyse)
	14.40	2026	Kjørt 13kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 5rps, slutt ca 7,5rps
	14.50	2027	Kjørt 13kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 6rps, slutt ca 8,5rps (feilkjøring)
	1500	2028	Retest 2023 (usikkerhetsanalyse)
	1515	2029	Kjørt 15kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 5,5rps, slutt ca 8rps
	1525	2030	Retest 2023 (usikkerhetsanalyse)
	1540	2031	Kjørt 17kn fastholdt, uten towrope, med stigningstall 0.025 rps/v Startturtall 6rps, slutt ca 8,5rps

A.3 M3025A

LOGG FOR FORSØKSKJØRING MT-K4-S110 Rev.00			**Prosjektnr.: Intern** **Tittel: M3025A**
Dato	**Kl.**	**Test nr.**	**Kommentarer**
29.03.12	9.30	3000	13kn, 0.1 incr, 7-10rev
	9.55	3001	13kn, 0.1 incr, 6-10,5rev
	11.15	3002	13kn, 0.15 incr, 5-11rev
	1245	3003	13kn, 0.2 incr, 5-11
	1330	3004	13kn, 0.25 incr, 4-12
	14.05	3005	14kn, 0.15 incr, 6-12
	14.50	3006	12km, 0.15 incr, 5.5-13