PERFORMANCE PREDICTION OF A 112M WAVE-PIERCING CATAMARAN

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E Kay, **J Lavroff** and **M R Davis**, University of Tasmania, Australia

SUMMARY

The prediction of power required to propel a high-speed catamaran involves the hydrodynamic interactions between the hull surface and the surrounding fluid that may be difficult to compute numerically. In this study model-scale experiments are used as a basis for comparison to full-scale sea trials data measured on a 112m Incat wave-piercing catamaran to predict the full-scale powering requirements from model-scale testing. By completing water jet shaft power measurements on an Incat vessel during sea trials, comparison of these results was made to model-scale test results to provide good correlation. The work demonstrates that the International Towing Tank Conference (ITTC) extrapolation techniques used provide a good basis for extrapolating the data from model-scale to full-scale to predict the power requirements for the full-scale catamaran vessel operating at high Froude Number with water jet propulsion. This provides a useful tool for future designers and researchers for determining the power requirements of a catamaran vessel through model tests.

NOMENCLATURE

A_V	Hull immersed cross sectional area, (m^2)
C_{AAM}	Wind resistance coefficient, model
C_{AAS}	Wind resistance coefficient, ship
C_A	Model to ship correlation allowance
C_{AppM}	Appendage resistance coefficient, model
C_{AppS}	Appendage resistance coefficient, ship
C_D	Drag coefficient, ship
C_{FM}	Frictional resistance coefficient, model
C_{FS}	Frictional resistance coefficient, ship
C_R	Residual resistance coefficient
C_{TM}	Coefficient of total resistance, model
C_{TS}	Coefficient of total resistance, ship
L	Length (m)
P_S	Ship power (resistance x speed) (W)
R_{TM}	Total model resistance (N)
R_{TS}	Total ship resistance (N)
S_M	Moving wetted surface area, model (m^2)
S_{0M}	Standing wetted surface area, model (m^2)
S_S	Moving wetted surface area, ship (m^2)
S_{0S}	Standing wetted surface area, ship (m^2)
T	Draught (m)
T_S	Ship thrust (N)
V_A	Wind speed (m/s)
V_j	Water jet velocity (m/s)
V_M	Model speed (m/s)
V_S	Ship speed (m/s)
Δ	Displacement mass (kg)
$1+k$	Form factor
η	Propulsion system efficiency (%)
λ	Model-scale ratio
ρ_A	Air density (kg/m^3)
ρ_M	Water density, model (kg/m^3)
ρ_S	Water density, ship (kg/m^3)

ABBREVIATIONS

AMC	Australian Maritime College, University of Tasmania
CFD	Computational Fluid Dynamics
HSMV	High Speed Marine Vessel
Incat	International Catamarans
ITTC	International Towing Tank Conference
JHSV	Joint High Speed Vessel
LCG	Longitudinal Centre of Gravity
MARIN	Marine Research Institute Netherlands
MCR	Maximum Continuous Rating
SKM	Sinclair Knight Merz

1. INTRODUCTION

The performance and powering prediction for vessels is of great importance for ship builders to determine the engine and propulsion requirements at the design stage to meet the agreed terms of the ship contract. If a ship does not meet the performance criteria of the contract between the ship customer and ship builder then the constructer often incurs a financial penalty (Watson, 2007). As such, much research has been conducted into the powering and propulsion of vessels in recent times.

A large amount of research has been conducted regarding the relationship between full-scale data and model-scale testing, including research by Doctors & Day (1997), Molloy (2001), Doctors (2003) and Bose & Molloy (2009), and for high-speed catamarans by Insel & Molland (1991), Molland (1994), Couser *et al*. (1997) and Rovere (1997), but significantly less has been done in relation to High Speed Marine Vessels (HSMV) and particularly high-speed wave-piercing catamarans.

Molloy (2001) developed an alternative to the traditional ITTC 1978 method, and recommended it as a possible solution to extrapolation of power for both standard and unconventional propulsion systems. Model test data was used to develop the new method, and compare its final results with the ITTC 1978 method. However, Bose & Molloy (2009) found that powering prediction of ships from the results of testing unconventional propulsion systems using the ITTC 1978 method or modified versions proved unsatisfactory. In order to perform a valid comparison it is crucial to have a set of full-scale trials for each data set with which the extrapolated data are compared.

As the size of water jet units has increased, more importance has been placed on the development of reliable powering prediction methods. From early work completed by Levy (1965), Walker (1971), Barham (1976), Etter (1976), and more recently by Allison (1993), Baba & Hoshino (1993), Bowen & Coop (1993) and van Terwisga (1996), the ITTC formed a waterjets group which reported to the 21st ITTC regarding a possible power prediction method for water jet systems (Kruppa 1996), an approach referred to as the momentum flux method.

Another aspect that needs to be considered when predicting the resistance and powering requirements for high-speed wave-piercing catamarans is the flow separation from the transom that occurs at a Froude number of around 0.23. If a portion of the transom is immersed, this leads to separation taking place as the flow from under the transom passes aft beyond the hull. This leads to flow vorticity in the separated region behind the transom and leads to a pressure loss behind the hull (Carlton, 2007). This loss is only taken into account in some analytical procedures.

In work by Doctors & Day (1997) it was assumed that the flow separated fully behind the transom, giving a fully un-wetted transom condition. As a consequence the matter of a partly un-wetted transom was not considered. It was stated by Doctors (2003) that the resulting predictions for the resistance in the low-speed regime would be too high.

Watson (2007) states that high-speed tests present more complexity relative to lower speed tests as the influence of Reynolds number and transom separation increase with speed. Multi-hull displacement hull forms also add complexity due to viscous and wave interference effects caused by the close proximity of multiple hulls (Haase, 2013). It was identified that the accurate determination of the form factor is difficult for high-speed craft as the value varies with speed and changes when transom separation occurs.

The current ITTC documents regarding testing and extrapolation methods for high-speed marine vehicles include ITTC (2008) for resistance testing and ITTC (2005) for waterjet propulsive performance prediction.

Sea trials of the constructed ship are an important prerequisite for the acceptance of the ship by the shipowner and are always specified in the contract between shipowner and shipyard (Bertram, 2000). This is of particular importance in the design of high-speed wave-piercing catamarans. A major selling point is their ability to transport a large amount of deadweight relative to displacement at speeds often above 40 knots. If the contract speed is not met during commissioning sea trials and the vessel is subsequently delivered without having met this speed requirement, then the ship builder may incur a financial cost. This can have an effect on both the reputation and financial standing of the ship builder and ongoing development of the industry. To this end the present research is of importance to ship designers as it provides a useful tool for predicting the power requirements of high-speed catamaran vessels. This was undertaken through the use of model-scale experiments where the results from the model test work were extrapolated for comparison to the full-scale sea trials data. This has shown to provide good correlation confirming the methods developed by ITTC for predicting the resistance and power of the full-scale catamaran vessel, as set out by ITTC (2008) when applied to water jet propelled vessels operating at high Froude number.

Data from two sets of model testing was used, one set of data from the Australian Maritime College (AMC) by Watson (2007) and the other from the Marine Research Institute Netherlands (MARIN) by Lafeber & Hulshop, (2008). This provided a broader cross-section of model-scale data to compare to full-scale data. The work completed is unique in that model-scale data of a HSMV wave-piercing catamaran is directly compared to the full-scale sea trial power measurements.

The present research also aimed to build upon the findings of DePaoli (2011) and further develop the ability to determine the most suitable engine for a vessel operating at medium speeds. This is particularly crucial with increasing emission regulations, market competitiveness, and economic challenges to be faced in the future. These experimental results can then be used to validate Computational Fluid Dynamic (CFD) models (Iliopulos *et al*., 2013) which will then be used to design and optimise hull forms for future medium speed catamarans with a speed of approximately 25 knots (Haase, 2013).

2. SEA TRIAL MEASUREMENTS

With reference to the work to be described in the present paper, Sinclair Knight Merz (SKM) was commissioned by Wärtsilä to conduct a test program on the propulsion system of the 112m Incat vessel (Hull 064) during her commissioning sea trials in 2007 (Gillespie, 2007).

A similar 112m vessel (Incat Hull 067) is shown in Figure 1. In the present paper full-scale data was sourced from SKM following the sea trials undertaken on Hull 064. The power data collected by SKM was then used for consolidation with the MARIN model-scale test data (to be discussed in the sections to follow). Table 1 lists the vessel particulars for Hull 064.

Figure 1: Incat 112 m vessel Hull 067 during sea trials (Incat, 2013).

Table 1: Vessel Particulars, Hull 064.

Length Overall, *L* (m)	112.6
Length Waterline (m)	105.6
Overall Beam (m)	30.5
Beam of Hulls (m)	5.8
Draft, *T* (m)	3.93
Speed (Knots)	40
Main Engines	4x MAN 28/33D, 9000kW
Water Jets	4x Wärtsilä LJX 1500

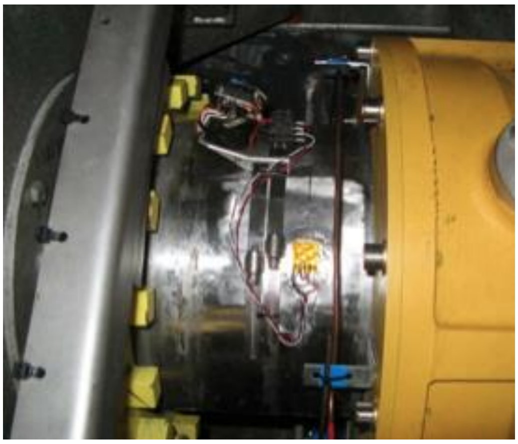

Figure 2: SKM shaft power instrumentation on Hull 064 112m wave-piercing catamaran (Gillespie, 2007).

Shaft speed was measured on all four jet shafts using reflective tape applied to the jet shaft speed collar and optical sensors (Keyence PZ-41 and PZ2-41) to generate one digital pulse train for each revolution (Figure 2). A Somat E-Daq recording instrument was used to calculate speed from the digital pulse train using pulse frequency data acquisition modes. A Somat Edaq Plus field computer equipped with 24 low level analogue input channels, 18 digital input / pulse counter channels and a GPS serial interface was used to record test signals from the measurements.

Shaft torque measurements were taken on all four jet shafts aft of the gearboxes using a full bridge strain gauge installation set-up directly on the water jet shaft. The torsional strain signal was obtained using four telemetry transmitter-receiver systems (SRI- PMD SS-580 digital telemetry system on starboard side and PMD T-20/R-102 FM telemetry system on port side). Strain gauges were bonded to the 150 mm long section of jet shaft between the jet seal and the thrust bearing. The strain gauges were configured in a full bridge to measure shaft torsional strain and the signal was obtained from the rotating shaft using four telemetry transmitter-receiver systems (Figure 2).

Ship speed was measured using a GPS receiver unit, which output and transmitted position data to the Somat Edaq Plus field computer. Latitude, longitude and heading were also recorded from the ship systems. All test signals (except GPS) were sampled at a frequency of 500 Hz with a low pass filter set at 167 Hz. GPS data was acquired at a rate of 1 scan per second.

During the sea trials, performance tests were carried out at two different displacements, 1,823 tonne total displacement on the first day and 2,023 tonne total displacement on the second day. Testing on the first day included two subtests, where the diesel engines were systematically operated over a range of engine speeds. Testing during the second day included the two same subtests but at the increased displacement of 2,023 tonne displacement overall. In each test the engine speed was held constant for one to two minutes while maintaining a predominantly straight heading. Water jet performance measurements (ship speed, shaft speed) were recorded during each of the four engine tests on both days and was processed to produce a set of measurement results for each test speed increment.

Some significant variations in power levels were evident during certain test runs at each displacement, which were likely related to changes in ship heading since sea conditions were calm. Higher power levels were developed during starboard direction turns and lower power levels with direction turns to port. This is likely due to the rotational direction of the water jet impellers. If water enters the jet pre-swirling then less power is produced, as there is less momentum flux across the water jet. In this case the rotational direction of the impellers meant that the port side jets generated less power.

Based on these observations, it is recommended that future speed trials are at constant heading and trim to give more accurate shaft power measurements.

To allow comparison to bare hull resistance predictions obtained from model-scale testing, thrust curves are used to determine the thrust provided by the waterjets (propulsive force) from the known speed of the vessel and the power output of the engines (as measured from the jet shafts). By using thrust curves for Wärtsilä LJX 1500SR water jets, and knowing the fraction of maximum continuous rating (MCR) and ship speed at any point in time, the corresponding net thrust (kN) can be determined from the thrust curve.

The MAN 28/33D engines are each rated 9000 kW and by measuring the power on each jet shaft the jet input power can be determined since transmission inefficiencies (such as gearbox losses and other mechanical losses) are accounted for when taking measurements directly on the water jet shaft. This can then be calculated as a percentage of the maximum engine output of 9000 kW, i.e. as a percentage of the MCR based on the shaft power measured directly on the water jet shaft.

Knowing the ship speed and percentage of MCR, thrust curves (which are supplied by the water jet manufacturer, Wärtsilä (2013)) are then used to determine the net thrust provided by the water jets at the specific condition. An example chart provided by Wärtsilä is shown in Figure 3 with reference to one particular operating condition shown with ship speed (dashed line). From such a specific shaft power measurements undertaken at full-scale it was determined that the engines were running at a given percentage of MCR, and by interpolating the net thrust between the MCR lines the net thrust was subsequently identified.

The results from the thrust curves were thus used to calculate the net thrust that was used as a basis for calculating the ship power (resistance x speed) P_s, and non-dimensional thrust.

Non-dimensional Thrust is here defined as:

$$\text{Non-dimensional Thrust} = \frac{T_S}{\rho_S {V_S}^2 L^2} \qquad (1)$$

where:

- T_S is ship thrust, (N).
- ρ_S is water density (kg/m^3).
- V_S is ship speed (m/s).
- L is ship length (m).

Figure 4 shows the non-dimensional thrust vs. ship speed for the two loaded conditions: 1,823 tonne total displacement and 2,023 tonne total displacement.

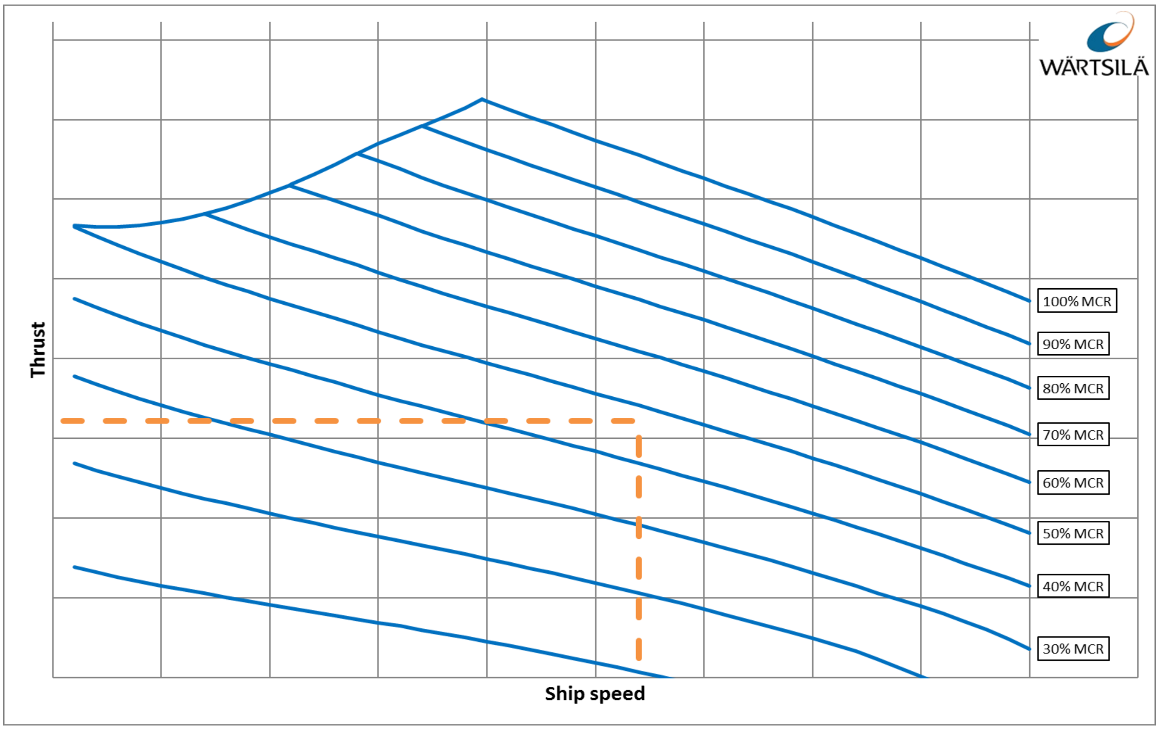

Figure 3: Example thrust curve provided by Wärtsilä, typical of a thrust curve for a Wärtsilä LJX 1500SR water jet.

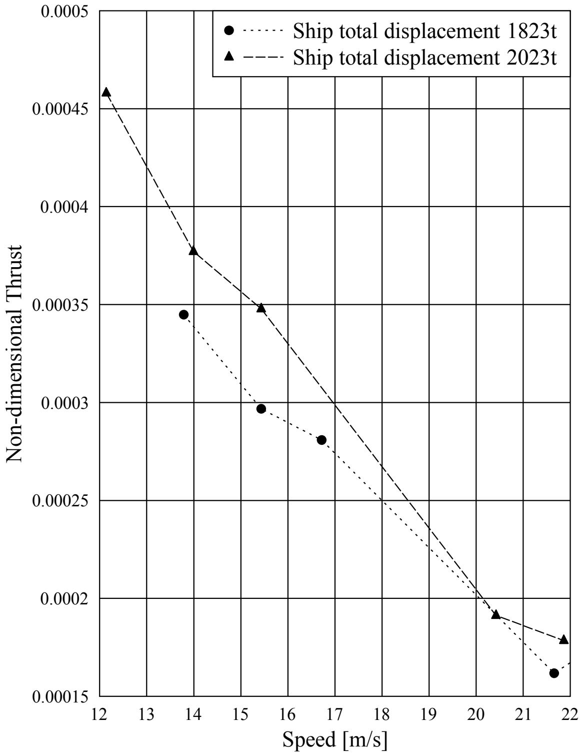

Figure 4: Non-dimensional thrust as a function of ship speed for Hull 064, 112m Incat catamaran.

From these results it can be seen that the net thrust required at any given speed is higher at the heavier displacement tested. This is primarily due to the increase in hull wetted surface area that increases the total resistance at the higher displacements as would be expected. As shown, the ship produced higher thrust at the 2,023 tonne displacement over all ranges of ship speeds tested.

3. MARIN MODEL SELF PROPULSION TESTS

In 2008 INCAT Bollinger requested MARIN to perform calm water, maneuvering and sea-keeping model test programs for the Joint High-Speed Vehicle (JHSV) wave piercing catamaran design, now known as the Spearhead class. Bollinger shipyards in USA and Incat jointly developed this ship for the US Navy. This vessel is of the similar hull design to Hull 064, the only difference being that instead of being powered by four diesel engines, the JHSV is powered by two MAN 28/33D diesel engines and two GE LM2500+ gas turbines. It also has a somewhat shorter length of 103 m, beam of 28.5 m and draft of 3.83 m.

For the purpose of this paper, the model-scale power data produced by MARIN was obtained with permission from Incat (Lafeber & Hulshop, 2008). The wooden model tested by MARIN was geometrically similar to the 112 m Incat catamaran with a total model length of 5 m and displacement of 219.48 kg.

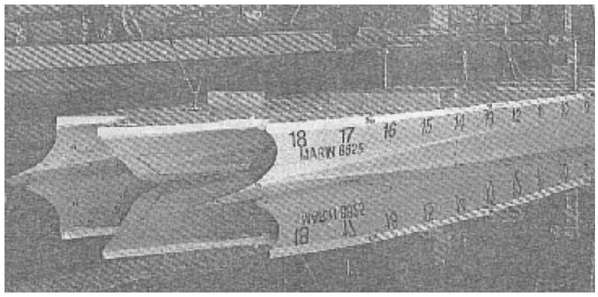

Figure 5: The 5 m MARIN model used for resistance testing at the MARIN towing tank (Lafeber & Hulshop, 2008).

The MARIN testing program comprised self-propulsion tests at five different loading conditions as listed (refer to Table 2):

- Transport Departure Condition (T1).
- Transport Arrival Condition (T2).
- Self-Deploying Departure Condition (T3).
- Self-Deploying Arrival Condition (T4).
- Transport Departure Condition, Additional Trim (T5).

The MARIN Deep-water Towing Tank used for testing the model has dimensions of 250 m by 10.5 m, and is 5.5 m deep. The maximum carriage speed is 9 m/s (Lafeber & Hulshop, 2008). The testing also included resistance trials, but the self-propulsion tests are of most interest here. Four water jet systems were installed, with the inlet geometry and nozzle diameter geometrically scaled (Lafeber & Hulshop, 2008).

Table 2: Relevant ship conditions for the MARIN catamaran test model (positive trim is bow up).

Condition	**Full-scale Displacement (Tonnes)**	**Static Trim (°)**
T1	2808	0.146
T2	2566	0.3
T3	2664	-0.142
T4	1975	0.4
T5	2808	-0.0833

Conditions T1, T2, T3 and T5 are referred to collectively as loaded ship, whilst condition T4 is referred to as light ship condition. Trim is either bow up or bow down, and is referred to as trimmed by the stern and trimmed by the bow respectively.

To extrapolate the power data from model-scale to full-scale, MARIN applied the ITTC 1957 formula incorporating a form factor ($1+k$) and model-ship correlation allowance (C_A) of 1.05 and 3.5×10^{-4} respectively (Lafeber & Hulshop, 2008). The power and speed data collected during the self-propulsion testing allowed the total resistance at model-scale to be determined.

MARIN also included in the correlation allowance the effects of the still-air drag of wind on the exposed ship superstructure. The value for C_A was determined from previous data collected by MARIN and from available full-scale trials data. The form factor was determined based on statistics and data from comparable vessels previously tested by MARIN. Revolution Design suggest a more appropriate correlation allowance of around $2.0x10^{-4}$, and suggests that the MARIN value is conservative in order to account for non-ideal sea conditions or a dirty, non-freshly painted hull.

In extrapolating the test results Froude similarity laws are followed. According to Froude's hypothesis, the resistance of a model can be spilt into two independent components, one being proportional to the resistance of a flat plate of the same length and wetted surface area when towed at the same speed (viscous component), and another component which follows the Froude similarity law. The proportionality factor between the viscous component and the flat plate resistance is called the form factor, (*1+k*), because it corrects for the effect of the three-dimensional hull form (Lafeber & Hulshop, 2008).

Figure 6 shows the total resistance coefficient, C_{TM}, for each MARIN model test condition. The general trends indicate that condition T1 recorded the largest coefficients, then T2, T5, T3 and T4 in that order. This can be broadly explained because the heavier the vessel, the greater the wetted surface area and hence the greater resistance. Dynamic trim also has a part to play on the total resistance, and the distribution of weight will play a part in influencing the dynamic trim. These effects have led to variations in total resistance between each model test condition as will be discussed in following sections of this paper.

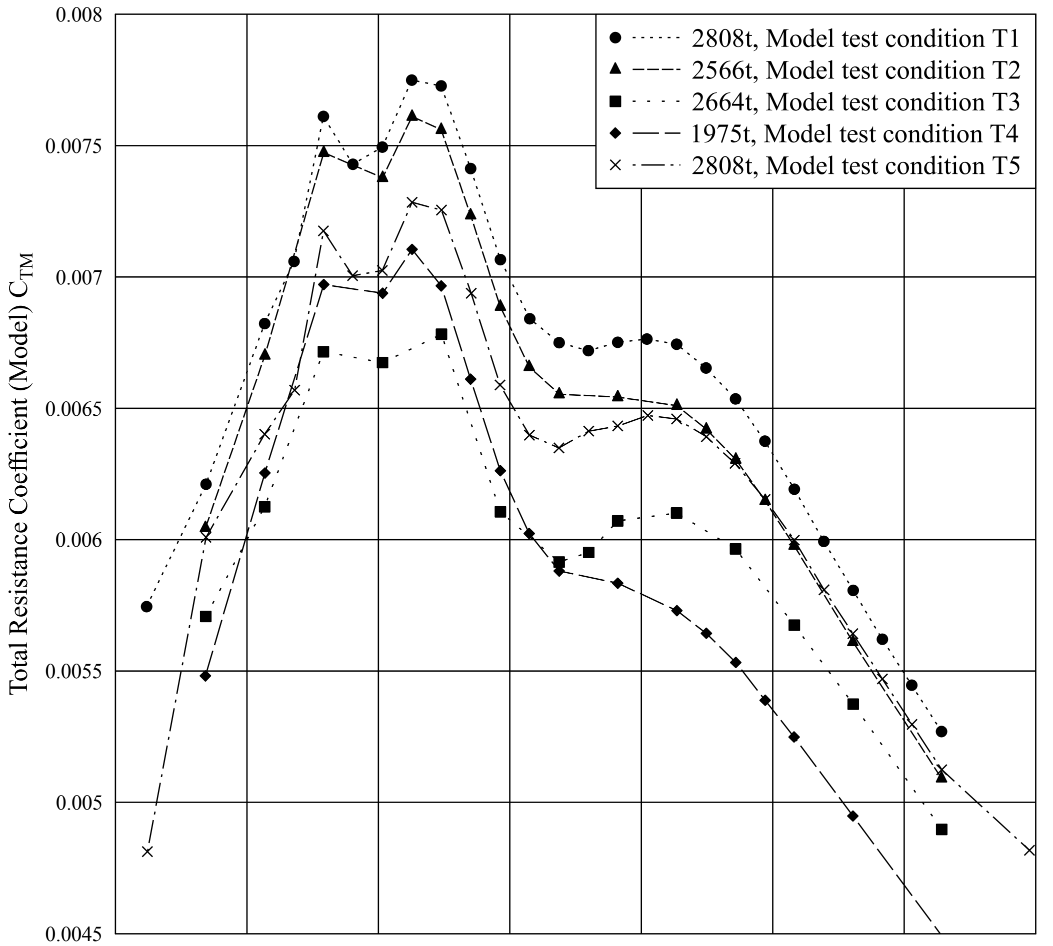

Figure 6: Total Resistance Coefficient (MARIN Model) vs. Speed.

The weight distribution for each model test condition not only determines the static trim but also has an effect on the dynamic trim. The stern tabs on the 5 m MARIN model were fixed (Lafeber & Hulshop, 2008), and as such the weight distribution that influences the static trim has an effect on the dynamic trim, which in turn influences the overall resistance measured. Table 2 shows significant variation of static trim and it is noted that whilst condition T2 has slightly smaller displacement than condition T3 it has much greater bow up trim. As seen subsequently it seems likely that these variations of trim between conditions in the MARIN tests have significantly affected the observed hull resistance.

Of note is the hump speed that occurs at a speed of around 3.0 m/s (a Froude number of around 0.44). This is a condition typically observed during sea trials and occurs at a speed of around 28 knots for a 112 m vessel. At hump speed the ship captain generally needs to increase engine power to overcome this added resistance due to the hydrodynamic interaction between the ship hull and the surrounding water. The hump speed presented here is unique to the Incat wave-piercer catamaran design. By identifying the hump speed and when this occurs further insight is gained into the development of medium speed hull forms with maximum ship operational efficiency since medium speed designs will operate close to the hump speed.

The following scaling equations as used here are sourced from ITTC (2008). They allow the resistance values measured during model-scale towing tank testing to be extrapolated to full-scale equivalent values. The total model resistance, R_{TM}, measured during the MARIN tests can be expressed in non-dimensional form as the coefficient of total resistance, C_{TM}:

$$C_{\mathrm{TM}} = \frac{R_{\mathrm{TM}}}{\frac{1}{2}\rho_{\mathrm{M}} S_{0\mathrm{M}} V_{\mathrm{M}}^2} \tag{2}$$

where:

- ρ_M is the water density at model scale (kg/m^3).
- S_{0M} is the wetted surface area for zero speed at model-scale (m^2).
- V_M is model speed (m/s).

The coefficient of residual resistance, C_R, is then calculated by:

$$C_{\mathrm{R}} = C_{\mathrm{TM}} - C_{\mathrm{FM}} \cdot \frac{S_{\mathrm{M}}}{S_{0\mathrm{M}}} - C_{\mathrm{AAM}} - C_{\mathrm{AppM}} \tag{3}$$

The model friction coefficient, C_{FM}, is derived from the ITTC 1957 correlation line for the model. C_{AAM} is the model wind resistance coefficient and C_{AppM} is the model appendage resistance coefficient, found from the difference in resistance by testing with and without appendages. S_M represents the running/dynamic wetted surface area for model-scale (m^2). S_{OM} is the standing wetted surface area at zero speed for the model.

The total resistance coefficient of the ship, C_{TS}, is then calculated by:

$$C_{\mathrm{TS}} = C_{\mathrm{R}} + C_{\mathrm{FS}} \cdot \frac{S_{\mathrm{S}}}{S_{0\mathrm{S}}} + C_{\mathrm{AAS}} + C_{\mathrm{Apps}} + C_{\mathrm{A}} \tag{4}$$

where:

- C_R is the residual resistance coefficient.
- C_{FS} is the frictional resistance coefficient, ship.
- S_S represents the running/dynamic wetted surface area for full-scale (m^2).
- S_{0S} is the wetted surface area for zero speed at full-scale (m^2).
- C_{AAS} is the wind resistance coefficient.

$$C_{\mathrm{AAS}} = \frac{\rho_{\mathrm{A}} V_{\mathrm{A}}^2 A_{\mathrm{V}} C_{\mathrm{D}}}{\rho_{\mathrm{S}} V_{\mathrm{S}}^2 S_{0\mathrm{S}}} \tag{5}$$

- ρ_A is the air density (kg/m^3).
- V_A is the wind speed (m/s).
- A_V is the ship frontal surface area (m^2).
- C_D is the ship drag coefficient.
- ρ_s is the water density at full-scale (kg/m^3).
- V_S is the ship speed (m/s).
- S_{0S} is the ship standing wetted surface area (m^2).
- C_{AppS} is the appendage resistance coefficient, obtained from extrapolation of C_{AppM} using the friction line.
- C_A is the model to ship correlation allowance, which allows for changes in hull roughness from model to full-scale.

Thus, the full-scale ship resistance, R_{TS}, is:

$$R_{\mathrm{TS}} = \frac{1}{2}\rho_{\mathrm{s}} V_{\mathrm{S}}^2 S_{0\mathrm{M}} \lambda^2 C_{\mathrm{TS}} \tag{6}$$

where:

- ρ_s is water density, ship (kg/m^3).
- V_S is ship speed (m/s).
- S_{0M} is standing wetted surface area, model (m^2).
- λ is the model-scale ratio.

giving a full-scale ship power, P_S, of:

$$P_{\mathrm{S}} = V_{\mathrm{S}} R_{\mathrm{TS}} \tag{7}$$

4. AMC MODEL TOWING TANK TESTS

A 2.5 m catamaran model was tested by Watson (2007) to investigate methods for correcting the dynamic trim of the model to replicate the full-scale Incat catamaran and investigate determination of the form factor. The catamaran model as shown in Figure 7 was based on the 112 m Incat wave-piercer catamaran (Hull 064), using a 1/44.8 scale model of the design with particulars listed in Table 3.

Figure 7: AMC 2.5 m model (Watson, 2007).

Table 3: AMC 2.5m catamaran model particulars.

Model	HSM01
Scale	1:44.8
***Length, L* (m)**	2.353
***Draught, T* (m)**	0.062 m

Model experiments were conducted in the AMC towing tank. The tank is rectangular in cross section with overall dimensions 100 m in length and 3.55 m in width. The carriage is capable of speeds up to 4.6 m/s and tests were conducted at 1.5 m water depth. Drag measurements were recorded using a Marin strain gauge force transducer, output to a dedicated filter (1 Hz) and then to a combined amplifier and filter. Trim and sinkage were measured using linear variable differential transducers (LVDTs). The LVDTs converted the linear displacement of the model to an electrical signal, which was subsequently amplified and filtered. The voltages were recorded by a LabVIEW data acquisition program, and converted to displacement in millimeters using calibration factors reported by Watson, (2007). All sensors were calibrated regularly as part of the standard tank testing procedure.

Watson (2007) undertook work into investigating trim tab lift and drag characteristics, and advantages and disadvantages of achieving a specific trim using only tab deflection or deflection combined with modifying the longitudinal centre of gravity (LCG). Following on from this a series of resistance tests were completed. Four different displacements were tested with a range of static trims (Table 4). The first displacement was 2,025 tonnes and was tested at static trims of 0º (level), -0.397º (bow down), -0.668º (bow down), 0.316º (bow up) and 0.586º (bow up). The remaining displacements were 2,500 tonnes, 2,800 tonnes and 3,000 tonnes, all tested at level static trim.

Table 4: AMC resistance testing conditions for the 2.5 m catamaran model.

Model Displacement (kg)	**Equivalent Full-scale Displacement (Tonnes)**	**Static trim (º, bow up positive)**
21.972	2025	-0.668, -0.397, 0, 0.316, 0.586
27.126	2500	0
30.381	2800	0
32.551	3000	0

Figure 8 shows the total resistance coefficients for each displacement at level static trim (0º), extrapolated to full-scale. The 2,025 tonne displacement had the lowest resistance coefficients, the 2,500 tonne displacement second lowest, 2,800 tonne second highest and the 3,000 tonne displacement had the highest resistance coefficients. This is as expected, as the lightest displacement should produce the least resistance, and the heaviest the most resistance. Of note is the resistance hump that occurs at a model speed of 2.1 m/s (Froude number of 0.44, as also observed in the MARIN tests with a much larger model).

It is to be noted that the dynamic trim of the model is arguably the most significant issue facing model-scale resistance testing and might be significantly different to the trim of the full-scale vessel. Changes in trim can have a large effect on wetted surface area and the hydrostatic force distribution between the transom and bow as well as to transom separation. ITTC (2008) guidelines recommend that the dynamic lift and trim, air resistance and scale effects of appendages during testing be given special attention.

For comparison with full-scale testing the 2,025 tonne displacement at level static trim is most appropriate, as this most closely matched the SKM sea trial conditions. Watson (2007) concluded that the main problems with resistance comparisons were the difference between full-scale and model trim, the difficulty in determining an accurate form factor and scaling of the frictional resistance. The transom drag term also appears to be a significant factor in the HSMV extrapolation process and the form factor determined by testing at the same trim is the best way to estimate the actual form factor of a full-scale vessel. It was also suggested that the use of a correlation allowance can produce a blanket correction for better model resistance extrapolations.

5. COMPARISON OF MARIN AND AMC MODEL TESTS WITH SKM FULL-SCALE TRIALS RESULTS

The aim of the study was to predict required power at full-scale using model-scale test data. ITTC methods were used to scale up model test data to full-scale. As described in the previous sections model test data was sourced from MARIN tests of a 5 m model, and AMC tests of a 2.5 m model based on the 112 m full-scale Incat catamaran. From this, non-dimensional power was calculated for all test results as follows:

$$\text{Non-dimensional Power} = \frac{P_S}{\rho_S {V_S}^3 L^2} \tag{8}$$

where:

- P_S is ship power (resistance x speed), (W).
- ρ_S is water density (kg/m^3).
- V_S is ship speed (m/s).
- L is ship length (m).

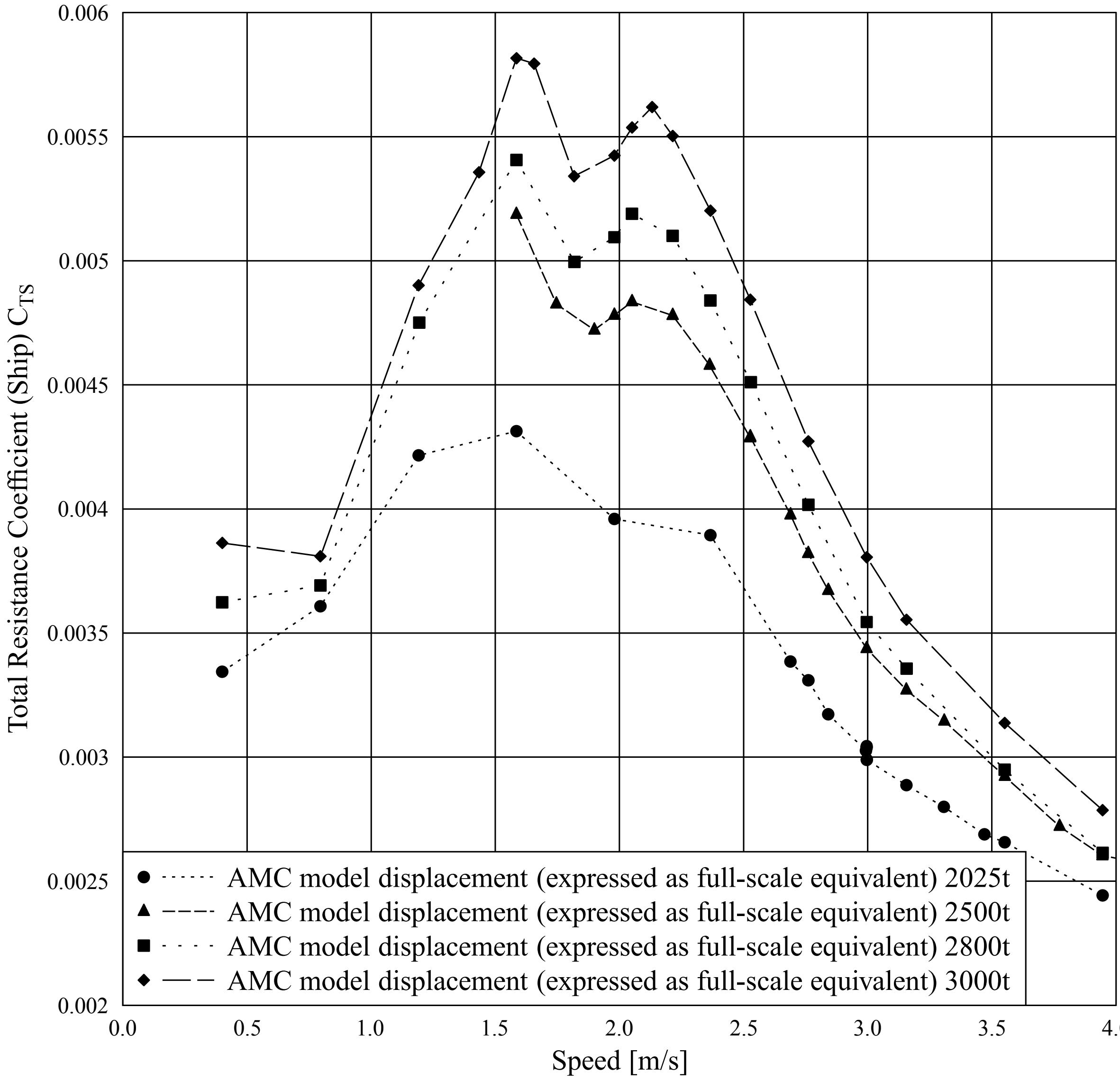

Figure 8: AMC 2.5 m catamaran model resistance tests at level static trim (0°) showing the total resistance coefficient measured with forward speed (at model scale) as a function of equivalent full-scale ship displacement.

Figure 9 shows non-dimensional power as a function of ship speed for model tests by MARIN, model tests by the AMC and the 112 m Incat catamaran sea trial results reported by SKM for both displacements (1,823 and 2,023 tonnes). These results now provide a direct comparison of the ship power measured at model-scale for comparison to the power measured at full-scale. Whilst the displacements for the various tests shown in Figure 9 are variable, these fall into two broad load conditions—light ship condition (AMC, SKM and MARIN T4) and heavy ship condition (MARIN T1, T2, T3 and T5). It is noted that there is significant variation of non-dimensional power due to displacement and trim and that there are significant differences between the two model tests and the full scale trials results, the lowest (T4) and highest (SKM) results having similar displacements.

Outlier points in the full-scale SKM data have been excluded in Figure 9 as these points exceeded the rest of the data by around 25%. Trim tab settings are usually selected to give maximum speed, but a sub-optimal setting of the trim tab or variation of sea conditions, which were not reported, could have caused this variation. The full-scale trials may of course have been affected by wind, sea conditions or water depth causing a variation in resistance leading to the difference between model scale results and full-scale results as reported here, in particular near the hump speed at 10-15 knots. However, the powering sea trials are usually conducted in relatively deep, calm water without significant wind and these effects would be expected to be small.

For the AMC model tests trim was level at zero speed. For the MARIN tests, trim tab position was optimised for each test condition, with either zero, five or ten degrees of downwards trim tab used. For the full-scale ship, trim was controlled dynamically using the ride control system. At full-scale the ship is trimmed using the ride control system to give best speed, and generally the vessel is controlled to go bow down at speed. This is achieved by applying a down deflection of the stern tabs by around 5 degrees.

From the MARIN tests we see that increasing displacement increases the non-dimensional power at about 1.5 x 10^{-7} per tonne for speeds above 16 m/s full-scale and at about 1.75 x 10^{-7} per tonne at about 10 m/s. This is broadly consistent when positive trim cases are compared (T1, T2 and T4) although the trim does vary somewhat between these cases. From the MARIN tests we also see that an increase of bow up trim leads to an increase of non-dimensional power at about 1.0 x 10^{-4} per degree when cases of similar displacement are compared (T1/T5 and T2/T3). Only the MARIN tests give direct evidence of the effect of trim.

The AMC tests have non-dimensional power about 1.0 x 10^{-4} above the MARIN tests at high speed (>16 m/s) and about 0.5 x 10^{-4} at low speed (8 to 10 m/s) when cases of similar light ship displacement are compared (AMC and MARIN T4).

The full-scale SKM tests appear much more variable than the model tests with three data points which appear to be outliers and may have been affected by some other external effect such as wind, wave or manoeuvres. The full-scale tests are about 1.0 x 10^{-4} above the AMC tests at low speed and about 0.5 x 10^{-4} above at high speed, these comparisons being for similar light ship displacements. The full-scale SKM tests show an increase of non-dimensional power at a rate of about 1.5 x 10^{-7} per tonne, a sensitivity to displacement that is broadly consistent with that observed in the MARIN model tests. At similar displacement it thus appears that the full-scale tests were closer to the AMC tests than the MARIN tests but significantly above both model tests data sets.

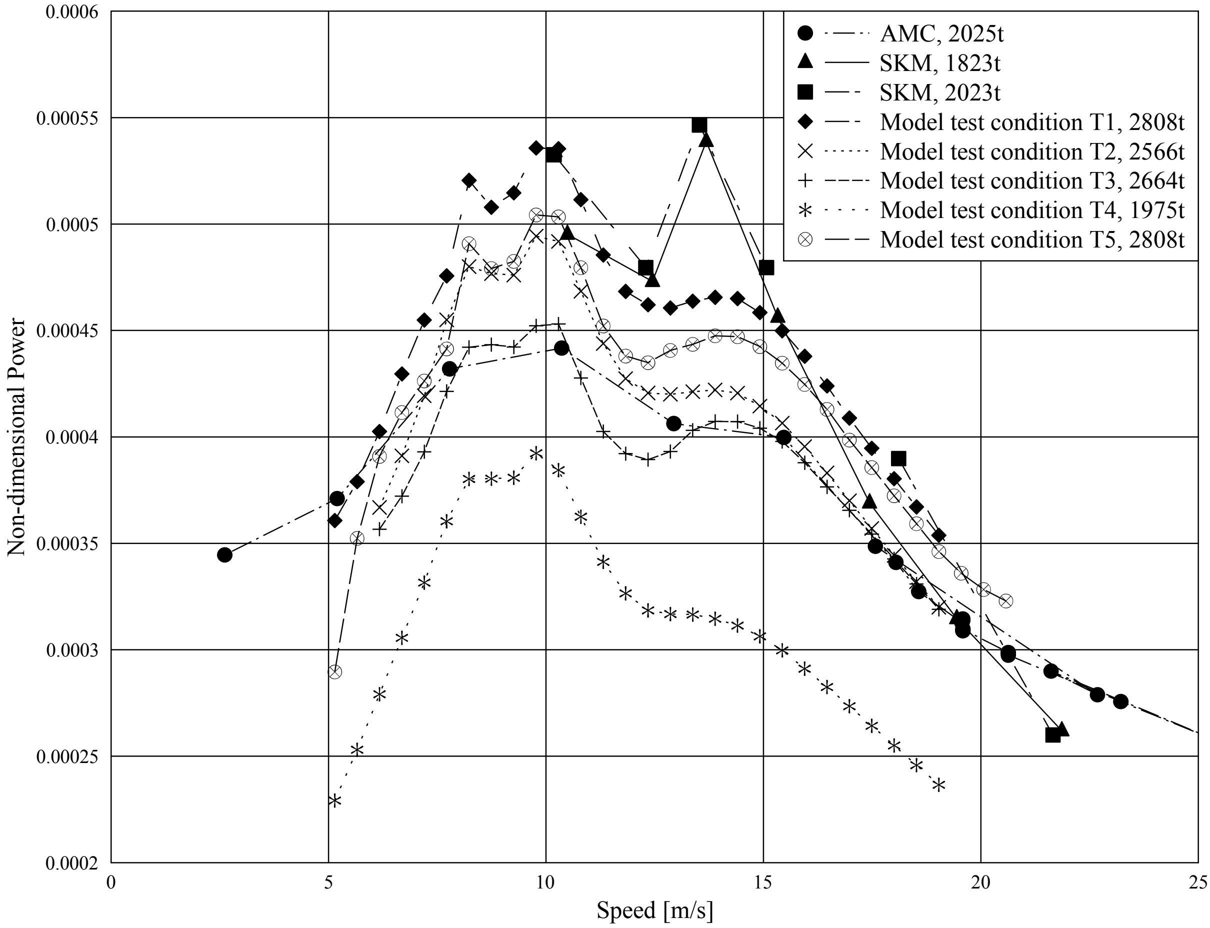

Figure 9: Non-dimensional Power vs. Ship Speed.

6. CONCLUSIONS

In this investigation model-scale experiments were used as a basis for comparison with full-scale sea trials data measured on a 112m Incat wave piercing catamaran to determine the capacity to predict the full-scale powering requirements from model-scale testing. Water jet shaft power measurements on the full scale Incat vessel formed the basis of comparison with model-scale tests.

From the model test results presented it was shown that these methods can be used to not only identify the hump speed, but together with the ITTC procedures can be used as a basis to predict the power requirements at full-scale. This is a useful tool for future designers and researchers where they can effectively apply these techniques for appropriately determining the power required at full-scale.

During sea trials of Incat Hull 064, performance tests were carried out at displacements of 1,823 and 2,023 tonnes with different engine speeds. Model-scale data from MARIN tests using a 5 m model and from AMC tests using a 2.5m model was scaled to full scale following ITTC 1978 procedures.

The total resistance coefficients showed an increase with increase in displacement and hence increased power required to achieve a given speed in model tests and at full scale due to increase of hull wetted surface area. Propulsive power increased at a rate of approximately 3.3% for each additional 100 tonnes of displacement in both the full scale tests and scaled 5 m model tests. The 5 m model tests also showed an increase of required power with bow up trim at a rate of approximately 22% per degree of bow up trim as is evident in Figure 9. These new outcomes showing sensitivity to trim and displacement are of appreciable value for ship design and operations.

The full scale test non-dimensional power data was somewhat irregular but in general exceeded the 2.5 m model test data by approximately 11% at high speed and by 22% at low speed. Further, the 5 m test data showed non dimensional power that was approximately 22% less than the 2.5 m model data for the light ship condition.

It is apparent from the foregoing discussion that whilst the trends in both model and full scale non-dimensional power data with displacement are broadly similar, the absolute values of the non-dimensional power coefficients at full scale significantly exceed the values scaled from the model test data. However, it should be borne in mind that the non-dimensional power coefficient is quite sensitive to vessel trim and that variation of trim using the stern tabs may not always have been optimal as is usual operating practice. Therefore it seems most likely that these differences are due to differences in vessel trim which was not reported specifically for the full scale trials.

From the results presented there is clearly a hump in resistance coefficient at a similar scaled speed (10 m/s full scale) at both model-scale and full-scale. All model testing also demonstrated a second hump in the resistance coefficients occurring at a Froude number around 0.44 (around 14 m/s as shown in Figure 9 when extrapolated to full-scale). Below this speed, at Froude numbers around 0.35, there is opportunity to develop large displacement catamarans operating at medium speed to give significant reduction in fuel consumption and so to increase the transport efficiency.

Following this investigation it is clear that future work should investigate more closely the effect of vessel trim on powering requirements. Other aspects deserving further work include the effect of shallow water on the resistance of high-speed wave-piercing catamaran hull forms as it is known that there have been significant effects of water depth on the performance in shallow water. Finally powering test data can be used as a basis for validating numerical CFD software and developing the confidence of designers in applying numerical computations for powering predictions.

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