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Propeller-Engine Interaction in a Dynamic Model Scale Environment

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ABSTRACT

Regulatory demands on ship designs, such as emission and manoeuvrability requirements, are becoming increasingly stringent, raising the need for advanced methods to predict and assess dynamic propulsion plant behaviour of a new design. At present, model scale experiments and numerical simulations are not able to predict this behaviour in full detail. To fill the resulting knowledge gap, this paper proposes to further develop existing scale model tests into so-called *dynamic model basin tests*. These tests aim to predict dynamic behaviour of the ship propulsion plant in complex, dynamic environments in more detail, leading to improved propulsion systems and controls and ultimately, lower emissions, lower fuel consumption and increased manoeuvrability.

KEY WORDS: ship model, ship propulsion, propeller-engine interaction, towing tank, model basin, Hardware In the Loop.

NOMENCLATURE

B	[m]	Beam
c_b	[-]	Block coefficient
D	[m]	Propeller diameter
F_n	[-]	Froude number
f	[Hz]	Wave encounter frequency
g	[m·s^2]	Gravitation constant
I_p	[kg·m^2]	Polar moment of inertia
L	[m]	Length
M	[N·m]	Torque
n	[s^{-1}]	Rotation speed
P	[W]	Power
R_n	[-]	Reynolds number
T	[m]	Draught
t	[s]	Time
v	[m·s^{-1}]	Speed
w	[-]	Wake fraction
λ	[-]	Geometric scale factor
ν	[m^2·s^{-1}]	Kinematic viscosity
ω	[rad·s^{-1}]	Angular speed

INTRODUCTION

As early as the 16th century, ship constructors conducted resistance tests by towing scale models through water, thus increasing insight into optimal hull shapes. Later, when combustion engines began superseding sail power, self-propulsion tests were conducted. These tests replace the external towing force by the thrust delivered by the model's own propeller, which usually rotates at a controlled, constant speed. As such, the ship's self-propulsion point can be identified. As a further development, self-propelled ship models were fitted with a rudder assembly, which allowed to conduct manoeuvrability tests; next to fuel consumption and attainable speed, a ship's manoeuvrability is another fundamental design quality. Finally, seakeeping performance of the ship can be predicted by introducing waves in the model basin.

However, the model is often a simplified version of full scale reality, meaning that the behaviour of the model does not completely correspond to that of the actual ship. For instance, one of the main limitations in manoeuvrability and seakeeping tests is the fact that the model propeller speed is (quasi) constant, irrespective of propeller load variations. This implies that dynamic interactions between environment, hull, propeller and machinery are not properly taken into account. At full scale, these interactions do have considerable influence: rough seas impose a fluctuating load on the propeller, possibly increasing the diesel engine's fuel consumption and wear. Moreover, so-called *hydrodynamic scale effects* occur: incorrectly scaled flow of water around hull and propeller result in incorrect scaling of forces. Despite such shortcomings, however, the aforementioned experiments have been relied upon for many decades to assess ship designs. Simplifications and scale effects can often be compensated for, or even neglected. Manoeuvrability tests, for instance, have shown to produce useful predictions (Hooft, 1994).

Nonetheless, the interest in more advanced predictions of ship

behaviour has been increasing in recent years. Traditionally, scale model tests have primarily been used to determine static operating points of the propulsion plant in order to meet contract and safety requirements, but the strive for ever lower operational costs (and hence, low energy consumption and high reliability) has generated interest in dynamic predictions of propulsion plant behaviour rather than only static predictions. Additionally, the increasingly stringent environmental regulations on ships such as the IMO's Energy Efficiency Design Index (EEDI), as well as increasingly stringent naval manoeuvrability standards such as the STANAG 4154 standard (Armaoglu et al., 2010), raise the need for more accurate manoeuvrability predictions, in which propulsion plant dynamics can play a crucial role. For example, the gradual decrease in installed propulsive power, as required by the EEDI regulation, has sparked doubts whether future ships will still have sufficient propulsive power and available propeller thrust to ensure safe navigation in adverse weather conditions (Papanikolaou et al., 2015).

Consequently, introducing propeller-engine interaction into existing prediction methods would provide a useful extension to these prediction methods. The question that naturally follows is how to achieve this in the best way: in the past decades, means to evaluate ship hydrodynamics other than model scale tests have seen considerable development. Computational Fluid Dynamics (CFD), a collective term for numerical approaches on fluid dynamics, plays an increasingly important role in ship resistance predictions (Hunt and Zondervan, 2007). However, computational intensity and varying accuracy still limit the applicability of CFD in assessments of manoeuvrability (Wang and Walters, 2012; Carrica et al., 2016). CFD-based determination of the self-propulsion point, too, incorporates simplifications of, for instance, free surface effects and propeller forces (Krasilnikov, 2013). Computational prediction of complex flows requires an analytical solution of the highly complex Navier-Stokes differential equations, which has not yet been found. Furthermore, CFD packages are, at the moment, primarily suited to simulate hydrodynamics over a time range in the order of (at most) seconds, owing to the considerable computational capacity that is required. This does not align with the desire to incorporate propulsion plant dynamics: reproduction of the interaction between prime mover, propeller and hull over periods of several minutes is required to investigate transient behaviour between equilibrium points and dynamic behaviour in waves.

As another option, one can obtain longer term, numerical predictions of propulsion plant behaviour using simulations based on first principles and regression. Regression-based estimation methods however generally depend on input from model scale and full scale steady-state measurements, the method proposed by Holtrop and Mennen (1982) being a notable example. As a result, these prediction methods give good approximations of reality while requiring limited computational power, but do not allow evaluating complex phenomena such as propeller ventilation events.

The aforementioned methods can be divided into strictly hardware or software oriented methods: model scale tests (using hardware) result in physical measurements, whereas CFD and other numerical methods rely on software to generate predictions. Both have their advantages, and combining physical models with numerical modules may, in some cases, offer the best of both worlds. In fact, such “hybrid" experiments have been used to predict operational behaviour for many years and are often referred to as *Hardware-In-the-Loop* (HIL). In HIL tests, some components are included as hardware, while other parts of the system are simulated by a software module, as is shown in Fig. 1. Reasons to include a hardware component may be the complexity of the

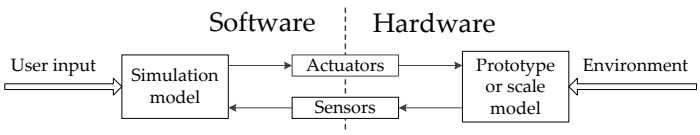

Fig. 1. Diagram of a generic Hardware-In-the-Loop setup.

component, which makes accurate numerical simulations of the component not feasible, or to test the proper functioning of the hardware component as a finished product - for instance, a controller - in a simulated, extreme environment. For instance, Schreiber et al. (2016) describe an HIL test assessing the dynamics of an automotive brake system design, while Li et al. (2006) use HIL to demonstrate a new control algorithm for wind turbines. Other applications of HIL comprise function checks of marine control systems for certification (Skjetne and Egeland, 2006) and factory acceptance tests (Johansen et al, 2005).

Applied in naval architecture, one could consider the flow regimes around hull and propeller as highly complex and hence, hard to model phenomena. Fully numerical approaches to ship propulsion dynamics, such as demonstrated by Schulten (2005), estimate wake fraction, oblique propeller inflow and other hydrodynamic effects using analytical models, regression-based variables or fixed values. Yet, analytical models of hydrodynamics are generally simplifications of complex phenomena, so physically including these complex phenomena in an experimental loop would add a level of detail to propulsion plant behaviour predictions. Propulsion system dynamics on the other hand can be adequately simulated by a numerical model, as was demonstrated by, among others, Campora and Figari (2003) and Geertsma et al. (2017). Vrijdag (2016) gives an overview of the possibilities of HIL in ship design, mentioning the possibility to control the electric propulsion motor of a model scale ship in such a way that its shaft torque behaves like that of a diesel engine. Tanizawa et al. (2013a,b) and Kitagawa et al. (2014a) describe results of such model basin HIL tests. In their experiments, some offsets and phase shifts become apparent - possibly owing to hydrodynamic scale effects and chosen hardware. In further work, they propose a method to correct in realtime for scale effects on measured propeller torque and speed, based on the ITTC-1978 performance prediction method. Thus, they obtain a static prediction of required propulsive power in waves which shows a good agreement with the actual, average required propulsive power at full scale (Kitagawa et al., 2014b). A later report demonstrated that such model basin HIL tests could be used to investigate effects of engine speed governor settings (Kitagawa et al., 2017).

Encouraged by the results of the aforementioned work, which shows that HIL can be used to make static predictions of average power requirements in dynamic environments as well as to assess the influence of governor settings, the authors will investigate whether HIL can be used to assess in detail the oscillations of torque and speed (and possibly, other aspects of dynamic behaviour) of the propulsion engine in such dynamic environments. This paper gives an outline of the research project which aims to explore this subject.

RESEARCH GOAL AND SCOPE

The authors expect that application of HIL in the model basin can predict dynamic interactions between a ship's environment, hull, propulsor and machinery in a more detailed way than existing methods can. Eventually, the goal of this research project is to develop a new type of experiment that can be used (1) to evaluate the dynamic performance of the ship's propulsion system for research, providing

insight into the qualities of all-new, conceptual designs, (2) to confirm the capabilities of a ship in the design stage, and (3) to tune controllers of the ship before commissioning, thus saving precious time during sea trials.

The research goal can be condensed into a single, main research question:

To what extent can dynamic model basin tests add detail and reduce uncertainty regarding interactions between ship machinery, propulsor, hull and environment, compared to existing methods?
This question can be broken down into three objectives, each of which results into one or more sub questions:

1. Develop a setup for a *dynamic model test*, capable of recording interaction between simulated propulsion machinery, the propeller, the hull and the environment.
    a. Which requirements must a dynamic model basin setup meet?
2. Find a way to address mechanical and hydrodynamic scale effects.
    a. Which scale effects can be expected?
    b. What will be the magnitude of these scale effects?
    c. To what extent and how can scale effects be corrected for?
3. Assess how the results from a *dynamic model test* can be interpreted to assess dynamic propulsion plant performance.
    a. Which additional phenomena can be observed compared to existing methods?
    b. How does uncertainty in dynamic model tests compare to that in existing methods?

The term *dynamic model test* is elaborated in Section Proposed Method, while *mechanical* and *hydrodynamic scale effects* are described in Section Scale Effects.

The third objective mentions *dynamic propulsion plant performance* as a quality that will be evaluated. Predicting this dynamic performance however involves a broad range of subjects such as thermodynamics, chemistry and tribology. In this research project, the focus however lies on replicating full scale behaviour of dynamic drive torque rather than predicting all aspects of prime mover dynamics. Developing and adapting prime mover simulation models will be done only insofar as necessary to replicate dynamic prime mover torque.

Furthermore, the research scope is narrowed down to dynamic behaviour of the propulsion plant because of expected time and material constraints. In other words, manoeuvrability is kept out of the scope. This means that the number of degrees of freedom is reduced, simplifying the setup, while also eliminating the need for proper flow scaling around the rudder. The eventual goal of this research project is to develop an experiment which can reproduce full scale propeller-engine interaction in sufficient detail to predict the full scale diesel engine's dynamic load in waves and during accelerations and decelerations. In further research, this new type of experiment could be used (1) to improve numerical prediction models of ship propulsion plants by improving insight into propeller-engine interaction, (2) to produce detailed predictions of emissions, thermal load and wear of the

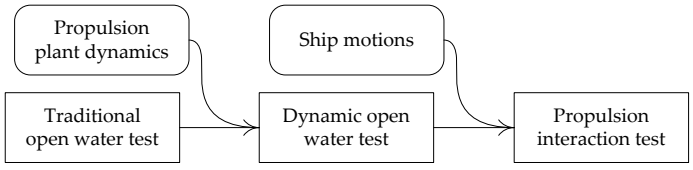

Fig. 2. Increasing complexity of proposed model basin experiments.

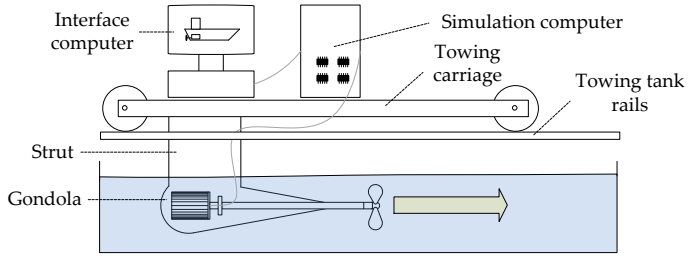

Fig. 3. General layout of a dynamic open water test. The computer on the left provides user interface and passes on set points to the simulation computer (also referred to as simulator). The simulation computer in turn simulates prime mover behaviour and controls the electric motor.

prime mover by introducing advanced simulation models, and (3) as an intermediate step towards scale model manoeuvrability tests incorporating realistic dynamic propulsion plant behaviour. As such, the knowledge gap on complex, dynamic behaviour at full scale can be bridged.

In the following sections, the research will be introduced, referring to the research questions when appropriate. This presents the reader with motives for the research questions as well as a method to answer these questions.

PROPOSED METHOD

To limit the number of problems that require attention, the direction of research must be made concrete. For instance, one can only determine the functional requirements to the experimental setup - the main goal of research question 1 - if it is clear what one wishes to measure with that setup. Hence, an outline of the proposed method and experiments is given first, taking into account the research scope. To this end, the authors choose to divide the subject of dynamic model basin tests into two principal stages, as shown in Fig. 2.

First, the possibilities to introduce dynamic behaviour of the prime mover and governor to a “traditional", static open water test, are explored. A static open water test involves a propeller rotating at (quasi) constant speed, moving forward at (quasi) constant speed in undisturbed water, so torque and thrust coefficients in function of advance coefficient can be determined. This test will be extended so propeller torque or speed is controlled by a simulation model of a prime mover including a governor, see Fig. 3. Furthermore, motions of the propeller in multiple directions will be made possible, using a hexapod suspension. For this extended open water test, the term *dynamic open water test* is proposed. This first step results in experience with dimensioning and controlling a setup which combines a propulsion

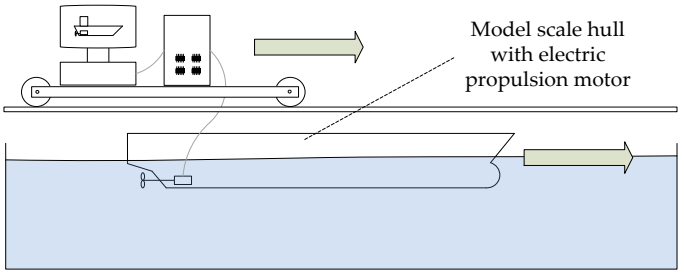

Fig. 4. General layout of a propulsion interaction test. The hardware on the towing carriage is the same as in the dynamic open water test.

plant simulator with a model propeller, at the same time enhancing insight into scale effects on the flow around the propeller.

When the dynamic open water test is sufficiently developed, the propeller and simulator will be used to propel a model ship hull, thus increasing the complexity of the test environment. This again is a departure from traditional, free sailing model tests, where thrust is generated by a propeller running at constant speed. The term *propulsion interaction test* is proposed to refer to tests with self-propelled ship models using a dynamic, simulation-controlled propulsion plant; see Fig. 4 for a simplified representation. *Interaction* in this nomenclature indicates that interaction between hull, propulsor and machinery in a dynamic environment is the main subject of interest during such experiments. *Propulsion interaction tests* conducted in the context of this research will focus on dynamic behaviour of the propulsion plant, rather than on replication of full scale manoeuvrability.

The *dynamic open water tests* and *propulsion interaction tests* differ from each other considerably in terms of hardware, meaning that the preparations are different, too. To make optimal use of available time, the *dynamic open water test* will be set up and explored first: the absence of a hull as well as a smaller number of degrees of freedom will likely lead to fewer practical and theoretical problems. If these problems are solved, a setup for *propulsion interaction tests* can be developed. For brevity, the term *dynamic model test* will refer to any test with simulated dynamic propulsion system behaviour.

SCALE EFFECTS

Scale model experiments in the towing tank are distorted by scale effects. Scale model and real ship are never dynamically similar, which means that forces and velocities in the flow around the scale model do not relate to full scale by a constant, predictable factor. This becomes evident in, for example, hull resistance, which can be divided into wave making and breaking resistance, and viscous resistance (Larsson and Baba, 1996). Force ratios related to wave making and breaking resistance of the scale model can be kept similar to full scale by maintaining Froude identity. As can be concluded from Eq. 1a, this implies that the speed of the scale model must be reduced. However, doing so changes the Reynolds number R_n, which is a measure for similarity of viscous forces. As a consequence, the ratio of viscous resistance to wave making and breaking resistance of the scale model differs from full scale. From Eq. 1b, one could conclude that R_n and hence, viscous resistance can be corrected by decreasing the kinematic viscosity of the fluid ν. In practice, however, there is no way to achieve this, resulting in a scale effect in the form of distorted model hull resistance. For scale effects related to fluid mechanics such as described before, the term *hydrodynamic scale effects* will be used from now on.

$$F_n = \frac{v}{\sqrt{g \cdot L}} \tag{1a}$$

$$R_n = \frac{v \cdot L}{\nu} \tag{1b}$$

Incorrect scaling of hydrodynamic forces, however, is not the only source of distortions. The geometry of the scale model's propulsion system has an influence on its behaviour, too. Consider, for example, a ship with a geared, diesel-mechanical propulsion system. The corresponding scale model could be powered by an electric motor - often a brushed DC motor - delivering a correctly scaled drive torque directly to the propeller. Ignoring hydrodynamic scale effects for the time being, one can expect that this setup would result in correct steady state torque. Yet, the model's behaviour will still not correspond to that of the actual ship, as accelerations of the shaft would likely be incorrect: correctly scaled torque is not the only requirement for correctly scaled shaft acceleration. As Eq. 2 shows, the angular acceleration of a body depends on the sum of torques acting on the body and the polar moment of inertia of the body. The direct drive DC motor likely has a different polar moment of inertia than a geared, downscaled diesel engine (after gearbox multiplication). As a result, the angular acceleration and hence, the dynamic behaviour of the shaft will not scale correctly, even if $\sum M(t)$ does correspond to the full scale situation. For such scale effects, related to the mechanics of the scale model, the term *mechanical scale effects* is proposed.

$$\frac{\mathrm{d}\omega(t)}{\mathrm{d}t} = \frac{\sum M(t)}{I_p} \tag{2}$$

Besides their classification as either hydrodynamic or mechanical scale effects, there is another important difference between these two scale effects. The effect of different viscous hull resistance influences the static equilibrium point, whereas the effect of different inertias influences the dynamic behaviour. The former is hence classified as a *static scale effect*, while the latter is a *dynamic scale effect*.
Traditional model basin tests generally aim to find static equilibrium points of the ship and its propulsion plant. As a result, only static scale effects pose an issue during such tests: dynamic distortions are simply not relevant. The dynamic model basin tests proposed in this research project, however, aim to also predict the dynamic behaviour of the full scale propulsion plant. This means that dynamic scale effects do become an issue. Since dynamic scale effects have never posed a problem for traditional model basin tests, little is known about how they exactly influence the dynamic behaviour of scale model propulsion systems. Previously reported dynamic model basin tests only spend little attention to the subject. Tanizawa et al. (2013a), for instance, corrected for improper I_p scaling by simulating and controlling shaft speed rather than drive torque, without further elaboration on possible disadvantages and alternative correction methods. However, the choice between torque and speed simulation is important from a control theory perspective: simulation of shaft speed, for instance, requires an additional integrator in the simulation loop, adding a phase delay at higher - but possibly relevant - frequencies.

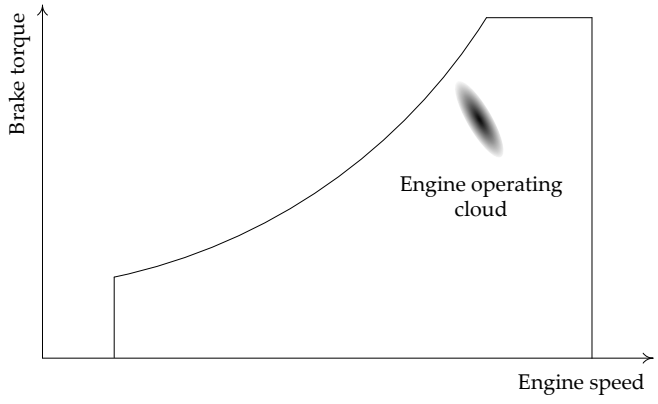

Fig. 5: Generic example of an engine operating cloud in the motor diagram. *x* and *y* axes can be modified to show different variables, such as fuel rack position instead of brake torque.

Considering this knowledge gap, a thorough analysis of dynamic scale effects will be the subject of future work. As an introduction to the topic of scale effects, this paper illustrates the effect of increased viscous hull resistance and polar moment of inertia of the power train by demonstrating their influence on the engine's operating cloud. The engine operating cloud is a recording of engine speed and engine brake torque over a prolonged period of time, and generally appears as an ellipsoid (or, in irregular waves, a superposition of ellipsoids) surrounding a steady-state equilibrium point. It is a good criterion of the ability of engine and governor to respond to load fluctuations; Van Spronsen and Toussain (2001), for instance, illustrate dynamic overloading of a propulsion engine of a RNLN M-class frigate by presenting the engine's measured operating cloud. Fig. 5 shows a generic example of an engine operating cloud.

Non-linear simulations of a full scale ship and the corresponding scale model were run to obtain an estimation of the two mentioned scale effects on the simulated engine operating cloud.
In the non-linear simulation model, the diesel engine is represented by a fuel injection map, which delivers brake torque in function of engine speed and fuel rack setting. Dynamic behaviour of the turbocharger is neglected, which means that delays in available air for combustion are not taken into account. In reality, a considerable, stepwise increase of injected fuel may cause the air-to-fuel ratio to drop to a level where not all fuel is burnt. The limiting effect on developed engine torque however diminishes within two revolutions for a four-stroke engine (Vrijdag and Stapersma, 2017), rendering a model based on a fuel injection map valid if only the engine torque is of interest. The benefits of this approach are a simpler model, requiring less numerical power, and a better correspondence between the non-linear model and the linear model proposed by Vrijdag and Stapersma (2017), allowing a better comparison of predictions. In cases where quantities such as temperatures in the combustion chamber are of interest, the engine model can be expanded with a turbocharger model; this is however outside the scope of the simulations described here.
The simulation is run at Froude similarity. From Eq. 1a, it follows that speed scales with $\lambda^{-0.5}$ (assuming constant g). This implies that time is also scaled, and that all other simulation parameters related to time must be scaled as well. The settings of the PID controller, used to govern engine speed, is a prime example of this requirement. The theory behind PID control is considered to be common knowledge and will not be covered here. If one considers a PID controller with dimensionless input and output values, dimensional analysis shows that at Froude similarity, the integrator gain K_i (unit: [s]) and differentiator gain K_d (unit: [s^{-1}]) scale from full scale to model scale with factors $\lambda^{0.5}$ and $\lambda^{-0.5}$ respectively. Static gain K_p, which is dimensionless, does not change.

Table 1. Main properties of a generic bulk carrier and corresponding scale model. The ship's propulsion diesel engine runs at constant speed and drives a Wageningen C4-40 propeller through a gearbox.

	Parameter	Symbol	Unit	Value
Ship	waterline length	L	m	150.0
	beam	B	m	22.5
	draught	T	m	10.5
	block coefficient	c_b	-	0.80
	nominal engine power	$P_{b,nom}$	W	6800E3
	nominal engine torque	$M_{b,nom}$	N·m	130E3
	propeller diameter	D	m	4.5
	propulsion inertia	$I_{p,FS}$	kg·m^2	70628
	service speed	v	m·s^{-1}	6.8
Model	geometric scale	λ	-	20
	propulsion inertia	$I_{p,MS}$	kg·m^2	0.1003
Waves	axial orbital velocity ampl.	v_w	m·s^{-1}	1.25

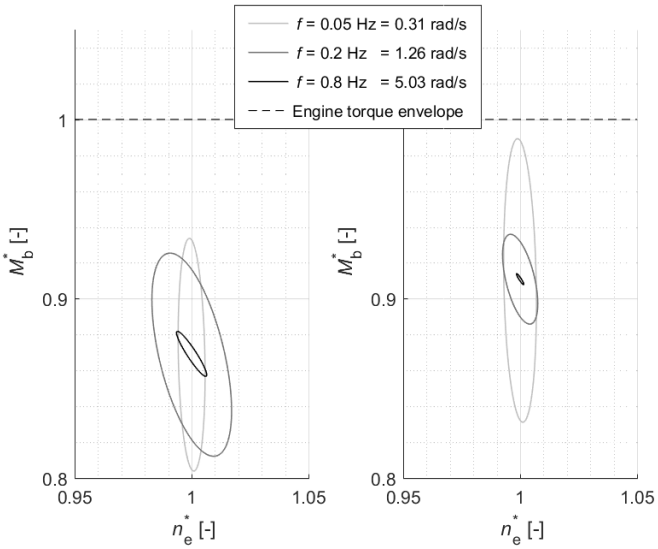

Fig. 6. Simulated engine operating clouds for ship (left) and model (right) described in Table 1, sailing in regular waves with different wave encounter frequencies. In this simulation, scale effects occur only on hull friction resistance and polar moment of inertia of the propulsion train. Brake torque, engine speed and wave frequencies are converted to full scale equivalents for ease of comparison.

These models of governor and prime mover are connected to a non-linear simulation of a Wageningen C propeller, as well as a hull resistance and efficiency function based on the prediction method by Holtrop and Mennen (1982). As an example case for this paper, the non-linear simulation model and the linear model mentioned before are used to evaluate a combination of two scale effects: incorrectly scaled viscous hull resistance, and incorrectly scaled polar moment of inertia of the propulsion train.
The viscous friction resistance component of the model hull is 202.6% that of the downscaled ship (in accordance with the ITTC-1957 viscous resistance line), while the model's propulsion train I_p is 4.54 times larger than it should be. The main properties of the simulated ship and scale model are given in Table 1, while Fig. 6 shows the engine operating clouds resulting from the simulation. One could expect scale effects to distort four properties of the ellipses:

1. Equilibrium value;
2. Ellipse size (area);
3. Ellipse eccentricity;
4. Ellipse orientation.

Recalling the categorisation of scale effects made earlier, item (1) refers to static scale effects while items (2) through (4) refer to dynamic scale effects. A closer look at the operating clouds in Fig. 6 suggests that items (1) and (2) are indeed affected by scale, while items (3) and (4) apparently remain the same for a given (full scale equivalent) frequency, independent of scale.

A particularly interesting observation is that the size of the scale model's operating ellipse evolves differently in function of wave encounter frequency than does the full scale operating ellipse. At a wave encounter frequency of 0.05 Hz (0.31 rad/s), for instance, the operating clouds have nearly the same dimensions, whereas at a

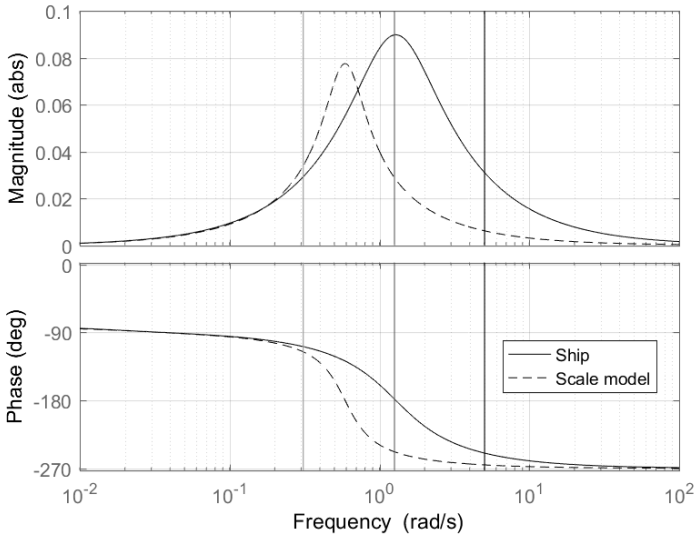

Fig. 7. Bode diagram of non-dimensional shaft speed response to non-dimensional wake fraction variations, $\delta w^*/\delta n^*$. Frequencies for the scale model have been corrected to full scale equivalents for ease of comparison. Verticals are drawn at frequencies of 0.31, 1.26 and 5.03 radians per second to facilitate comparison with Fig 6.

frequency of 0.2 Hz (1.26 rad/s), the scale model's operating ellipse is considerably smaller. This means that the simulated setup does not reproduce the dynamic behaviour of the real ship's propulsion plant: quite unfortunate, as this is exactly the purpose of the experiment. The same conclusion is drawn from predictions by linear models. Fig. 7 shows the bode diagram of the ship and model's shaft speed response to fluctuations of wake fraction (or advance speed), using the linear model for speed-controlled ship propulsion systems proposed by Vrijdag and Stapersma (2017). Simply put, this diagram indicates how far the ellipse stretches in horizontal direction (along the shaft speed axis). It appears that both the maximum gain and the (full scale equivalent) frequency at which this maximum occurs are not the same for ship and scale model. Linear and non-linear models predict a similar evolution of speed response with increasing disturbance frequency. Both models can hence be used to analyse scale effects on the properties of the operating ellipse; such an analysis will be performed in later work.

Apart from the two described scale effects, a range of other scale effects are expected. Before attempts are made to mitigate these effects, understanding of their underlying mechanisms is essential. To this end, differences in dynamic behaviour between model and full scale will be examined using linear and non-linear simulations. As the simulation models can be easily tuned, this approach allows to relate differences in dynamic behaviour to individual scale effects, and to identify the physical mechanisms behind each scale effect. This, in turn, allows to answer research sub questions 2(a), (b). Then, the subject of correcting (or circumventing) these scale effects remains, as formulated in sub question 2(c). This subject, too, will be covered extensively in future work.

EXPECTED RESULTS

The eventual goal of the research project is to predict full scale dynamic behaviour of the propulsion plant by means of model basin tests. Logically, validation would occur by comparing the operating cloud of a propulsion engine measured at full scale with the exact same situation at model scale. However, there are three reasons why this is not as easy as it might sound:

1. Full scale measurements are expensive, and hence, scarce;
2. Not all variables describing the environment are known (measured) during full scale measurements;
3. Full scale situations cannot always be entirely replicated in the model basin.

Consequently, there is no guarantee that a validation of dynamic scale model measurements against dynamic full scale data can take place. An alternative, however, is to compare the results from a dynamic model basin test with another, presently available prediction method for dynamic propulsion plant behaviour, as is suggested by research sub questions 3(a) and (b).

For example, one could use a HIL setup and a fully numerical model, using the same machinery simulation, to predict dynamic propulsion behaviour in the same environment. Subsequently, a qualitative comparison can be made to determine which phenomena are visible in the measurements from the HIL experiment, and on the other hand, in the results from the fully numerical prediction. Additionally, uncertainties in the results from HIL and fully numerical predictions can be compared, quantifying possible improvements of dynamic model basin experiments over available prediction methods.

CONCLUSION

As of yet, the difficulties encountered during dynamic model basin tests are largely unknown: HIL in the towing tank is relatively new. At the same time, these dynamic tests combine the advantages of simulation and physical experiments into a single setup. Using such a setup, one could potentially predict full scale interaction between machinery and flow around the hull with a high level of detail, and use it as a tool to solve a range of complex problems related to ship manoeuvrability and machinery performance. Considering these promising features, the research project introduced in this paper aims to solve both theoretical and practical problems, and as such, lay the foundations of this new prediction method.

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