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Potential of Hardware-in-the-Loop Simulation in the Towing Tank

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***Abstract*—Traditionally, model scale tests of ships are carried out without taking into account the dynamics of the shipboard systems that are involved in the operation under consideration. An example of this is the way that model scale free sailing tests in waves are carried out. The ship model is mounted with an electric motor, shaft and propeller and subsequently tests are carried out with constant propeller speed. In some cases constant shaft torque or constant power are employed. However, neither of these options reflects realistic behaviour of the drive system, because in waves and during manoeuvres the propeller speed, torque and power are in fact variable and their dynamic behaviour is governed by the drive train characteristics. The question arises to what extent, and in which cases, the dynamics of the shipboard systems affect the overall system behaviour. In this paper the application of Hardware-in-the-Loop (HIL) simulation in a ship model basin or towing tank is explored as a means to answering that question.**

The ambition is to develop an instrumented model scale ship of which the components of the drive train and its control are included by means of a correctly scaled time domain computer simulation model of the propulsion system. This simulation model is to run on a real-time processor which, via IO cards, provides electric power to an electric motor on-board the instrumented model scale ship, which in turn drives one or multiple shafts and propulsors. In this paper the role of scale effects on the test set-up is discussed. It is also shown that, in order to simulate realistic drive train dynamics in waves and during manoeuvres, it must be ensured that the combination of partial simulated drive train on the one hand *and* electric motor dynamics plus shaft and propeller inertia on the other hand should, as a total, represent the real dynamics of the drive train system.

I. Introduction

In this paper the application of Hardware-in-the-Loop (HIL) simulation in a ship model basin or towing tank is explored. The foreseen experimental setup integrates model scale testing of ships with real-time dynamic simulation of the shipboard systems including their controls. In this way the traditionally separate disciplines of ship hydromechanics, marine engineering and controls can be considered simultaneously including interactions with each other and with the dynamic environment. This new integrated test setup offers the possibility to study integrated system behaviour including complex non-linear hydrodynamic behaviour of ships and platforms in demanding conditions such as waves. The setup can furthermore help to quantify expected full-scale system behaviour which can be verified during ship acceptance trials. This in turn will add to the safety, environmental friendliness, workability and performance of ships at sea. The setup provides opportunity to investigate and demonstrate the potential benefits of novel shipboard subsystems, system architectures and control strategies to ships and platforms, and thereby supports and stimulates further innovations.

To emphasise the importance of the foreseen setup, a real life example is given here: Low powered ships which, in bad weather conditions, can suffer a lack of course keeping capability. In 2013 the IMO introduced Energy-Efficiency Design Index (EEDI) regulations requiring a final 30% reduction of the EEDI number from the starting baseline, which comes down to a 30% reduction in installed power. Potentially, EEDI could then affect safety in adverse conditions as the lowered installed power diminishes course keeping capability and the ability to make sufficient headway [1]. This potential course keeping inability is worsened by propeller ventilation via the following mechanism: after a drop in required engine torque following a ventilation event, the fuel injection to the diesel engine is reduced by the engine governor. The reduction in fuel injection is followed by reduction of turbocharger speed which in turn is followed by a drop in charge air pressure. The moment the ventilation event is over, the required propeller torque is restored again, while the fuel injection is held back because of the lagging charge air pressure. As a result, the engine does not provide the required torque, and the requested propeller speed and related thrust cannot be delivered, which has serious safety consequences, especially in adverse weather where sufficient thrust and resulting rudder lift force is required to ensure course keeping capability. In case the course keeping (and recovering) capability is not sufficient the safety of ship and crew is at stake. IMO has recognised this potential shortfall of the EEDI regulations and defined Minimum Power Requirements (MPR) [2]. An evaluation of the MPR is ongoing at MARIN. Findings thus far suggest that the multidisciplinary nature of the problem requires that the disciplines hydrodynamics, marine engineering and controls should be combined in studying the described process and its potential effect on safety. This example demonstrates that, with increasingly complex interaction of the shipboard systems, in some cases the performance of a vessel or platform can hardly be verified, analysed or optimised by means of model scale tests if these do not include the interaction between the various subsystems.

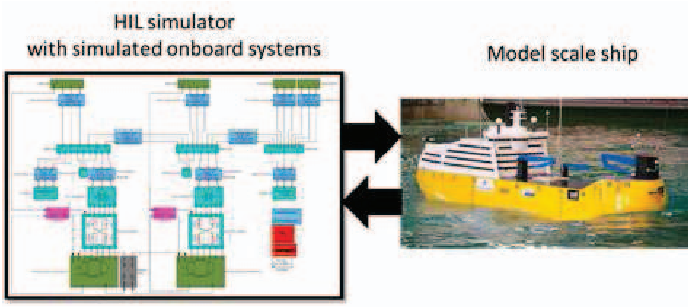

Fig. 1. HIL simulation in the model basin

A possible way to get in control of complex interaction between (maritime) systems is to consider Hardware-in-the-Loop (HIL) simulation, where parts of a system are replaced by simulation models, while other parts consist of actual hardware (in this case model scale hardware). The HIL approach is not new and has been applied for decades in other industries: it is for instance the industry standard approach in the automotive industry. In shipbuilding industry the HIL technique has mostly been used in naval projects of which the development and testing of the propulsion control system of the Dutch M-class frigates is an early example. Other examples include the development and testing of propulsion control systems for the Italian Navy [3], [4]. The offshore industry has embraced HIL techniques to verify and fine tune system control in order to reduce risk at late project stages [5]. The experimental setup as proposed in this paper can nevertheless be seen as a novel application of the HIL technique because it includes the model scale hull in order to incorporate real hydrodynamics into the system (see Figure 1). This inclusion of model scale hydrodynamics is important in those cases where at this point in time hydrodynamic computer models are not sufficiently accurate, not fast enough, or otherwise not suited to dynamically interact with simulation models from other disciplines.

Earlier attempts to accomplish such an integrated experimental setup exist, albeit limited. Noteworthy are an exploratory Delft University MSc-thesis project that applied HIL simulation in a model basin environment at MARIN [6]. However, shortcomings in the setup did result in unsatisfactory results, and the topic was not taken up further. Other publications on the continuous development of such a setup by the National Maritime Research Institute of Japan can also be found [7], [8], [9], [10].

Based on the foregoing, inclusion of a model scale hull in a HIL setup seems an attractive extension and important enrichment of the currently available toolset to simulate, analyse, and improve dynamic performance of integrated systems at sea. The unique aspect of the setup is that it truly includes the interaction between (complex) hydromechanics and the multidisciplinary onboard systems. There are however a number of scientific challenges related to the creation of the described experimental setup that need to be solved first. This paper aims to identify those challenges, and to explore options to deal with them.

II. ITTC RECOMMENDED PROCEDURES

As a starting point a review is made of the relevant guidance given by the International Towing Tank Conference (ITTC) on how to execute free sailing tests [11]. Note that these guidelines were never meant to set up experiments aiming to investigate dynamic behaviour of drive trains including prime movers.

With regards to the scale of the model ITTC recommends that *"the scale should be chosen as large as possible, meaning the model size should be as large as possible, keeping in mind that scale effects in manoeuvring are not yet fully understood, and the larger the model the smaller the scale effect."* With respect to the model scale propellers it is stated that *"generally stock propellers are used and the scale is chosen with respect to a suitable propeller design...."* While the point regarding the scale of the hull remains valid for the proposed HIL setup, the recommendation regarding the stock propeller is probably not strict enough. In order to have an equal as possible static operating point, the $J - K_T - K_Q$ relation in the operating point should be equal to the full scale, while in order to have equal dynamic behaviour the steepness of the open water diagram (or more general: the shape) should be equal to full scale. To what extent this is possible in the light of scale effects remains to be seen.

With respect to scale effects it is explained that *"In manoeuvring tests with free running models, the propeller(s) is used to give the model the desired speed, i.e. to produce the thrust to keep the desired speed, and also to produce a propeller induced flow over the rudder(s). Froude scaling of speed is generally applied and a tripping turbulence simulation device should be fitted, as it probably will give a more realistic boundary layer development and pressure distribution along the hull."* Furthermore two types of scale effects are mentioned: larger model wake fraction and the larger model resistance. *"Sometimes it might be necessary to compensate the larger frictional resistance of the model with an additional propulsion device, e.g. a wind fan or air jet device. Guidelines for this still need to be established and there is no worldwide consensus."* Further reference is made to scale effects on rudder performance and to unknown scale effects.

With respect to model setup it is stated that *"Generally the propeller is run at a constant rpm throughout the complete test, except for the stopping test. However, the real engine characteristics may be simulated by controlling rpm with computing dynamic response of the engine including torque limit. In order to model the engine, an instantaneous measurement of the propeller torque is necessary. For a sufficient accurate measurement of torque, the propeller diameter should be larger than 10 cm. The instantaneous measured torque should be fed into the model control system, which may reduce the RPM of the vessel to achieve a constant torque, power or otherwise."* The proposed minimum size of propeller diameter seems an appropriate starting point. The description of the ”response of the engine” is intended well, but as far as understood is a very limited representation of real drive train dynamics. For specific cases this might however be sufficient.

III. Similarity and Scaling Laws

Leaving the aforementioned scale effect aside for now, it is possible to derive similarity relations between model scale and full scale quantities. Geometric similarity is related to the similitude of length scales and resulting angles. Surface roughness also belongs to this type of similitude. Kinematic similitude requires geometric similarity and time similarity. The latter requires that at homologous time, a homologous particle lies at a homologous location. Dynamic similarity requires geometric and kinematic similarity plus mass (or force) similarity [12].

In ship model testing, ideally the experimenter seeks to achieve dynamic similarity between model scale and full scale experiments. This requires that all relevant non-dimensional parameter groups should have the same value for the model as in reality. This is however practically impossible for experiments involving model scale ships, which inevitably leads to scale effects.

The non-dimensional groups that govern the static and dynamic behaviour of a ship and its propulsion train can be determined by means of the Buckingham Pi Theorem. Without giving the full derivation, a list of 16 non-dimensional quantities (Π's) is given in the first column of Table I. Some of the non-dimensional quantities are well known and have a name, while others are less common.

The second column shows in what way the variable under consideration should be scaled in order to have equal non-dimensional quantity on model scale (subscript m) and full scale. The scaling of geometric characteristics is straightforward and only depends on the scaling ratio α, which is defined as $\alpha = \frac{L}{L_m}$, where L is the length of the ship and $\alpha > 1$.

Other required scaling factors include terms related to the difference in density and viscosity between fluid as used on model and on full scale. Normally the difference between the two due to for instance freshwater and seawater is limited, but for completeness the terms are kept here. Note that gravitational acceleration is assumed equal on model and real scale.

Other required scaling factors include a term involving the speed ratio $\frac{v_m}{v}$. This is where a choice has to be made between various possible meaningful velocity scaling methods. The most well known scaling methods are Reynolds (Re) and Froude (Fn) scaling. Froude similarity is normally adhered to at the cost of Reynolds similarity, as shown in the third column which reveals that the model scale Reynolds number is roughly $\alpha^{1.5}$ times too small.

Besides the consequential scale effects due to too low Reynolds number, there are a number of other consequences of Froude scaling for the onboard system simulation model. First of all such a simulation model has to run a factor $\sqrt{\alpha}$ faster than real-time, which increases the required computing power of the processor board. This also means that the model scale shaft speed should be $\sqrt{\alpha}$ times larger than full scale. Model scale torque should be multiplied by $\alpha^4 \cdot \frac{\rho}{\rho_m}$, and shaft inertia (as ship inertia) should be scaled by $\frac{1}{\alpha^5} \cdot \frac{\rho_m}{\rho}$ as shown in Table I.

Table I is not complete. For cases where cavitation or ventilation are present, additional variables play a role which result in additional non-dimensional quantities. Examples of such non-dimensional groups are for instance the cavitation number and the Weber number [13]. For now these are left out of scope.

Finally note that for completeness the Reynolds scaling is given in the fourth column, although such a scaling is not practical for model ship testing due to the high required actuating forces and high speeds. Reynolds scaling would in any case lead to erroneous wave resistance, although this might be acceptable in case of for instance HIL simulation of Dynamic Positioning operations.

Based on the given Froude scaling, and for now assuming no scale effects, one would expect that, as long as the shaft inertia is scaled correctly and the torque delivered to the shaft is equal to (scaled down) full scale delivered torque, in principle it should be possible to achieve similar dynamic behaviour of the model scale drive train compared to the full scale. This leads to the next paragraph in which the challenges with respect to the HIL setup are outlined.

IV. Challenges

A clear challenge, which is addressed in [9] and [10] is how to deal with the detrimental scale effects that are present due to the reduced scale of the model hull and propeller. Due to these scale effects it is impossible to fully replicate the full scale operating point of hull and propeller on reduced model scale. This issue is well known, and in free sailing tests it is common practise to (partly) mitigate scale effects of the flow around the hull by means of turbulence tripping. Addition of a small additional propulsive force by means of a ducted airfan to compensate for too high ship resistance on model scale is suggested and reported [9], [11].

The scale effects on the propeller operating point in normal free sailing tests are normally of less interest because such tests are mostly carried out to investigate manoeuvring behaviour and not to investigate propeller thrust and torque in detail. For the foreseen test setup the behaviour of propeller thrust and most importantly torque are however very important because the propeller torque is the variable that links the hydromechanical performance of the propeller to the mechanical performance of the shaft and drive system. To highlight the importance an example is given here: without any corrective measures the engine operating point at model scale would lie at a different location in the engine diagram, as shown in Figure 2. The consequence of this is that the protective features of the propulsion control system aiming to prevent operation outside the diesel engine envelope will become active too early or too late, compared to full scale, which would give an erroneous impression of system behaviour in waves or during manoeuvres. It is not unthinkable that, in order to simulate drive train behaviour realistically, it might be necessary to,

TABLE I

OVERVIEW OF NON-DIMENSIONAL QUANTITIES AND SCALING RULES RELATED TO SHIP RESISTANCE.

Non-dimensional quantity	Required scaling	Froude Scaling	Reynolds Scaling
$\Pi_1 = x^+ = \frac{x}{L}$	$\frac{x_m}{x} = \frac{L_m}{L} = \frac{1}{\alpha}$		
$\Pi_2 = t^+ = \frac{v \cdot t}{L}$	$\frac{t_m}{t} = \frac{L_m}{L} \cdot \frac{v}{v_m} = \frac{1}{\alpha} \cdot \frac{v}{v_m}$	$\frac{t_m}{t} = \frac{1}{\sqrt{\alpha}}$	$\frac{t_m}{t} = \frac{\rho_m}{\rho} \cdot \frac{\mu}{\mu_m} \cdot \frac{1}{\alpha^2}$
$\Pi_3 = a^+ = \frac{a \cdot t}{v}$	$\frac{a_m}{a} = \frac{t}{t_m} \cdot \frac{v_m}{v}$	$\frac{a_m}{a} = 1$	$\frac{a_m}{a} = \frac{\rho}{\rho_m} \cdot \frac{\mu_m}{\mu} \cdot \alpha^3$
$\Pi_4 = \lambda_B = \frac{B}{L}$	$\frac{B_m}{B} = \frac{L_m}{L} = \frac{1}{\alpha}$		
$\Pi_5 = \lambda_D = \frac{D}{B}$	$\frac{D_m}{D} = \frac{B_m}{B} = \frac{1}{\alpha}$		
$\Pi_6 = C_B = \frac{\nabla}{L \cdot B \cdot D}$	$\frac{\nabla_m}{\nabla} = \frac{L_m \cdot B_m \cdot D_m}{L \cdot B \cdot D} = \frac{1}{\alpha^3}$		
$\Pi_7 = Ar = \frac{m}{\rho \cdot \nabla} = 1$	$\frac{m_m}{m} = \frac{\rho_m \cdot \nabla_m}{\rho \cdot \nabla} = \frac{\rho_m}{\rho} \cdot \frac{1}{\alpha^3}$		
$\Pi_8 = \frac{m \cdot v^2}{T \cdot L}$	$\frac{T_m}{T} = \frac{L}{L_m} \cdot \frac{m_m}{m} \cdot \left(\frac{v_m}{v}\right)^2 = \frac{\rho_m}{\rho} \cdot \frac{1}{\alpha^2} \cdot \left(\frac{v_m}{v}\right)^2$	$\frac{T_m}{T} = \frac{\rho}{\rho_m} \cdot \frac{1}{\alpha^3}$	$\frac{T_m}{T} = \frac{\rho}{\rho_m} \cdot \left(\frac{\mu_m}{\mu}\right)^2$
$\Pi_9 = \frac{T}{R}$	$\frac{R_m}{R} = \frac{T_m}{T}$		
$\Pi_{10} = Fn = \frac{v}{\sqrt{g \cdot L}}$	$\frac{v_m}{v} = \sqrt{\frac{g_m \cdot L_m}{g \cdot L}} = \frac{1}{\sqrt{\alpha}}$	$\frac{Fn_m}{Fn} = 1$	$\frac{Fn_m}{Fn} = \frac{\rho}{\rho_m} \cdot \frac{\mu_m}{\mu} \cdot \alpha^{1.5}$
$\Pi_{11} = Re = \frac{\rho \cdot v \cdot L}{\mu}$	$\frac{v_m}{v} = \frac{\rho \cdot L}{\rho_m \cdot L_m} \cdot \frac{\mu_m}{\mu} = \frac{\rho}{\rho_m} \cdot \frac{\mu_m}{\mu} \cdot \alpha$	$\frac{Re_m}{Re} = \frac{\rho_m}{\rho} \cdot \frac{\mu}{\mu_m} \cdot \frac{1}{\alpha^{1.5}}$	$\frac{Re_m}{Re} = 1$
$\Pi_{12} = \beta$	$\frac{\beta_m}{\beta} = 1$		
$\Pi_{13} = \frac{\omega \cdot L}{v}$	$\frac{\omega_m}{\omega} = \frac{L}{L_m} \cdot \frac{v_m}{v} = \alpha \cdot \frac{v_m}{v}$	$\frac{\omega_w}{\omega} = \sqrt{\alpha}$	$\frac{\omega_w}{\omega} = \frac{\rho}{\rho_m} \cdot \frac{\mu_m}{\mu} \cdot \alpha^2$
$\Pi_{14} = \frac{\gamma \cdot L}{a}$	$\frac{\gamma_m}{\gamma} = \frac{a_m}{a} \cdot \frac{L}{L_m} = \frac{t}{t_m} \cdot \frac{v_m}{v} \cdot \alpha$	$\frac{\gamma_m}{\gamma} = \alpha$	$\frac{\gamma_m}{\gamma} = \frac{\rho}{\rho_m} \cdot \frac{\mu_m}{\mu} \cdot \alpha^4$
$\Pi_{15} = \frac{I}{m \cdot L^2}$	$\frac{I_m}{I} = \frac{m_m \cdot L_m^2}{m \cdot L^2} = \frac{1}{\alpha^5} \cdot \frac{\rho_m}{\rho}$		
$\Pi_{16} = \frac{M}{T \cdot L}$	$\frac{M_m}{M} = \frac{T_m \cdot L_m}{T \cdot L} = \frac{1}{\alpha^3} \cdot \frac{\rho_m}{\rho} \cdot \left(\frac{v_m}{v}\right)^2$	$\frac{M_m}{M} = \frac{\rho_m}{\rho} \cdot \frac{1}{\alpha_4}$	$\frac{M_p}{M} = \frac{\rho}{\rho_m} \cdot \frac{\mu_m}{\mu} \cdot \frac{1}{\alpha}$

comparable to the above mentioned scale effect mitigating measures for the hull, incorporate some additional form of scale effect mitigation or compensation in the setup. Such a feature might be included by adding a small additional torque on the shaft, or perhaps by means of a realtime software algorithm that corrects the measured model scale operating point, such that the corrected operating point has the correct position in the engine envelope. Note that the shift in operating point shown in Figure 2 is conceptual. At this point nothing is said about the size and direction of the shift. Another

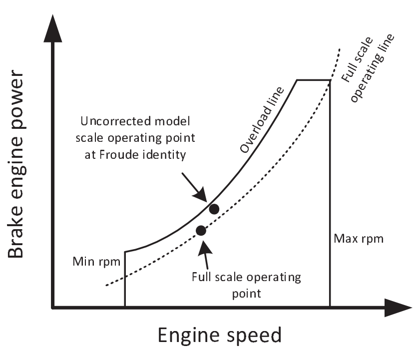

Fig. 2. Visualisation of scale effect on static operating point in the engine envelope.

challenge is to ensure that the dynamic behaviour of the drive train (and thus of the operating point in the engine diagram) reflects reality. The importance of this is visualised in Figure 3, which shows the full scale operating cloud (for instance due to sailing in irregular waves). The uncorrected model scale operating cloud lies at a different location, similar to Figure 2. Due to non-representative dynamics of the model scale setup the shape and orientation of the operating cloud is different than on full scale, which in this particular case would lead to activating the protective features of the engine control system

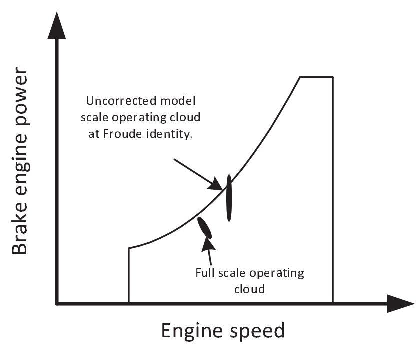

Fig. 3. Visualisation of effect of non representative model scale dynamics on the operating cloud in the engine envelope

too often, again leading to erroneous impression of system behaviour in waves or during manoeuvres. It therefore is deemed important to ensure realistic dynamic behaviour on model scale, although this is not a trivial task. The challenge is visualised in Figure 4, showing a schematic model of the propulsion train of both the full scale ship and of the model scale ship. The top figure shows the general ship propulsion block diagram as introduced by [14]. It describes the (non-linear) dynamics of a ship propulsion plant, including the propulsion machine and the propulsion control system. On the right hand side the ship translation loop is shown, based on a force balance between propeller thrust and ship resistance. When those two forces are out of balance, a net force will result in an acceleration or deceleration of the ship via:

$$v_s(t) = \int_0^t \frac{(F_{prop} - F_{ship})}{m_{ship}} dt + v_{s,0} \tag{1}$$

in which m_{ship} is the total ship mass including added water mass. Left of that the shaft rotation loop is shown, dealing with the balance between propeller and shaft torque. In the

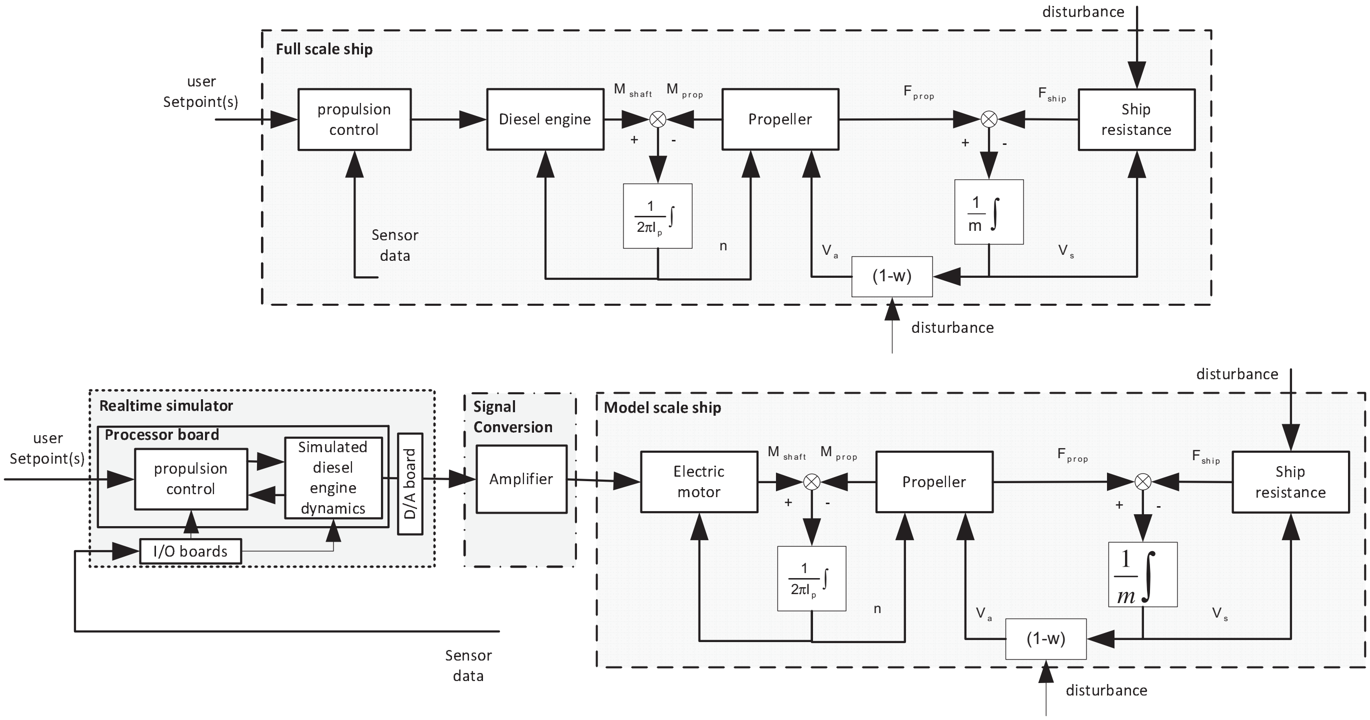

Fig. 4. Schematic visualisation of full scale and model scale system.

same way as in the translation loop a net torque will cause an acceleration or deceleration of the shaft system via:

$$n(t) = \frac{1}{2 \cdot \pi} \int_0^t \frac{(M_{shaft} - M_{prop})}{I_p} dt + n_0 \qquad (2)$$

in which I_p is the total polar moment of inertia of the rotating shaft system including propeller and entrained water. Note that for clarity no gearbox is included in the block diagram.
In the middle of the block diagram the propeller thrust and torque are modelled based on the open water diagram of the propeller under consideration.

$$\begin{aligned} F_{prop} &= \rho \cdot n^2 \cdot D^4 \cdot K_T(J,\theta) \\ M_{prop} &= \rho \cdot n^2 \cdot D^5 \cdot K_Q(J,\theta) \cdot \eta_R^{-1} \end{aligned} \qquad (3)$$

On the left the propulsion control system is shown, including sensor and user input.
The bottom part of the figure shows the foreseen HIL setup, where the model scale drive train partially consists out of hardware (a controllable amplifier, an electric motor, a shaft and a propeller) and partially out of time domain simulation software which simulates the non-linear dynamic behaviour of the diesel engine, gasturbine or other prime mover(s). The challenge is to ensure that the combined hardware and software of the HIL setup result in a system which is dynamically equivalent to the (non-linear) full scale system. Based on the foregoing some qualitative points of attention are given for design and setting up of the simulation model.
The characteristics of the time domain simulation model are in the sphere of control of the model maker. This means that, although it might be difficult to create a representative simulation model, there is no fundamental limitation here. The HIL simulator and its IO cards are however to be selected for high sampling speed, sufficient computing power and high signal resolution (to prevent resolution problems). The latter goes hand in hand with selection of suitable sensors. The commercially available amplifiers nowadays have many different features among which current control (and thus motor torque control [15]) and have a fast response compared to the mechanical time constant of the setup. The expectation therefore is that the amplifier related dynamics can be neglected, although this needs to be checked. The electrical time constant of the electric DC motor (typically in the order of milliseconds) is assumed to be negligible as well, while its inertia can be taken into account in the design of the model scale shaft. A potential concern is whether the relation between current and motor torque is sufficiently constant or at least predictable over the operating range. If this is not the case, additional measures need to be taken such as for instance feedback of torque measurement, while ensuring that this feedback loop has no negative effect on system dynamics in the frequency band of importance. With regards to the model scale ship and propeller the scaling laws (Table I) are to be followed.

V. Conclusions and Recommendations

From the qualitative analysis it is concluded that it should be possible to create a meaningful HIL setup in the towing tank or model basin. An important question is how important the hydrodynamic scale effects are and how to deal with them. The answer might be dependent on the case study under consideration. A possible option could lie in application of a correction torque, in a physical way or perhaps via a software algorithm that corrects measured torque in real-time. Ideas on how to do this need to develop further. Another challenge is to ensure equivalent propulsion plant dynamics on model scale compared to full scale. It is expected that the part of the model scale propulsion train that requires most attention is the combination of amplifier and electric motor. The requirement is that those components together are able to (almost instantaneously) produce the torque as calculated by the simulation model of the prime mover. To this end an investigation into the detailed characteristics and options of commercially available amplifiers and (DC) motors is recommended.

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NOMENCLATURE

Ar	Archimedes
a	acceleration
B	beam
C_B	block coefficient
D	ship draught
D	propeller diameter
F	force
Fn	Froude number
I	moment of inertia
J	advance ratio
K_T	thrust coefficient
K_Q	torque coefficient
L	length
M	moment
m	mass
n	shaft speed
Q	open water propeller torque
R	resistance force
Re	Reynolds number
T	thrust force
t	time
v	velocity
x	position
α	scale factor
β	heading
γ	angular acceleration
η_R	relative rotative efficiency
λ_B	beam/ length ratio
λ_D	draugth/ beam ratio
μ	dynamic viscosity
θ	propeller pitch angle
ρ	density
ω	angular velocity
∇	volume underwater ship