Investigation of asymmetrical shaft power increase during ship maneuvers by means of simulation techniques

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Abstract

Marine propulsion plants can experience large power fluctuations during tight maneuvers, with increases of shaft torque up to and over 100% of the steady values in straight course and considerable asymmetry between internal and external shafts during turning circle. This phenomenon (studied in Viviani et al 2007a and 2007b), can be of particular interest for twin screw ships propulsion systems with coupled shaftlines, in which asymmetrical loads can represent a challenge for the whole propulsion system (e.g. unique reduction gear, shaftlines, automation). A joint research has been set up in order to deeply investigate the phenomenon, by means of large scale model testing and related numerical simulations.

In the present work, preliminary simulation results with different simplified automation systems and with an automation system more similar to the real one are reported, allowing to get a better insight into this complex problem.

Keywords

Maneuverability; Propulsion plant; Simulation; Automation

Introduction

Marine propulsion plants can experience large power fluctuations during tight maneuvers. During these critical situations, dramatic increases of shaft torque are possible, up to and over 100% of the steady values in straight course. In the case of a twin-screw ship turning circle, the two shaft lines dynamics can be completely different in terms of required power and torque. In order to analyze this phenomenon, a preliminary work was performed in last years analyzing turning circle maneuvers at different speeds and rudder angles performed during sea trials for a series of twin screw naval ships. Results of this analysis allowed to underline a common trend for asymmetrical shaft power increase despite significant differences in ships considered in terms of dimensions, ship type and propulsion system (Viviani et al 2007a).

A simplified approach to the problem by means of the adoption of an asymmetrical variation of wake fraction during maneuvers was proposed. This approach seemed promising, despite still presenting a certain uncertainty and a not completely clear trend and correlation with ship characteristics. With this in mind, a parallel analysis by means of free running model tests was performed, in order to improve prediction accuracy for specific ships in preliminary design phases and to investigate possible scale effects for this phenomenon (Viviani et al 2007b).

On the basis of the outcomes of these preliminary analyses, it was clear that this phenomenon, if not correctly considered, may be potentially dangerous, mainly for propulsion plants with two shaft lines driven by a unique reduction gear, which can be subject to significant unbalances. This kind of propulsion plant, reported schematically in figure 1, despite not very common, has been recently proposed as a solution for particular applications, such as naval ships. In these cases the ship automation system has to be designed in order to prevent possible problems. From another point of view, effect of asymmetrical shaft power increase during maneuvers (and of different behavior of the automation plant) may affect maneuvering behavior of the ship, with effect on macroscopic parameters such as tactical diameter in turning circle.

In order to better analyze the physics related to this phenomenon, a new series of dedicated free running model tests (still under development at time of writing of present paper) has been planned, increasing the number of measurements with respect to usual set in this kind of tests and performing trials with different simplified automation behaviors (namely constant RPM, constant torque and constant power), as it will be presented in the paper.

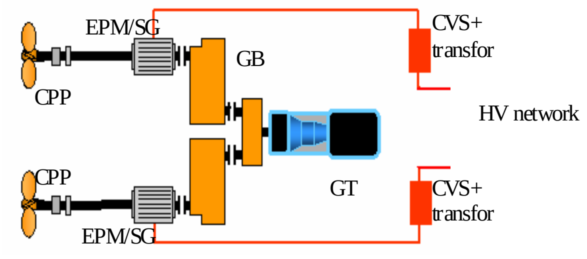

Fig. 1: Propulsion system configuration with coupled shaftlines

Reason for application of different simplified automation is related to the interest in analyzing its possible effect on global maneuverability characteristics (mainly due to different propeller loading in turn and resulting effect on rudder force). Moreover, it will be also possible to analyze effect of different configurations on asymmetrical power increase. From this point of view, asymmetrical wake fraction variation (and eventually thrust deduction factor variation) will be evaluated first from constant RPM tests and then validated (or modified) on the basis of results of other tests.

Results of these trials will be used for a fine tuning of ship propulsion system and maneuverability simulators, which in their turn may be used as a useful tool during ship propulsion system and automation design, being complementary to free running model tests, allowing to introduce elements which can hardly be represented in model scale (such as CPP, effective propulsion system functioning and automation effect, etc.). This approach of adopting hybrid simulators including propulsion system and ship maneuverability has been used at DINAEL for rather a long time (see Benvenuto et al 2003 and Altosole et al 2008), and is becoming a standard in complex propulsion system (and automation) design. For the particular problem of asymmetrical shaft loading, a preliminary work was presented in Viviani et al 2008, with promising outcomes.

In the present work, a brief summary of automation system behavior in general and of previous analyses regarding asymmetrical shaft power increase are reported. Moreover, a description of different simulators developed (i.e simulators with simplified and with more realistic automation) and of some preliminary results already obtained is reported. Finally, the programmed experimental campaign is summarized.

Automation system behavior

In present paragraph, a brief review of some concepts related to propulsion system automation is reported.

In a ship propulsion regulation chain two different controllers are simultaneously in operation: the Engine Controller and the Propulsion System Controller. Generally the Propulsion System Controller is provided by the shipyard while the Engine Controller is responsibility of the engine manufacturer. The normal control philosophy is based on the set-point check of two of the main propulsion parameters: propeller pitch and shaft speed.

The set-point control is done through the definition of proper rules (combinator), one for each working condition and maneuvering mode.

The lever signal is somehow elaborated before entering the combinator block. This technique is used in order to avoid that too rapid changes of the lever position may overload the propulsion system.

The engine control is based on a closed loop of the shaft speed, normally included into a governor block. The principal control is a PID (proportional, integral, derivative), usually with zero derivative action.

The propeller pitch control is used to obtain the desired ship speed and, as overload protection, to limit the shaft torque.

If two engines are operating on the same shaftline, the governor has to balance the loads on the two engines. The balance loop reacts to the torque difference between the two engines. In this kind of applications the response time of the balance loop is an order of magnitude longer than the shaft speed loop.

A similar loop may be required if it is necessary to balance the load of the two shaft lines. This latter function may be of particular interest when a propulsion plant configuration like the one reported in figure 1 is adopted, since it may avoid significant unbalances on the reduction gear.

The governor normally contains ‘load control’ functions with the aim to prevent overloads on the propulsion system components. The load control functions can act on the propeller pitch as well as on the fuel flow.

Previous data from sea trials and model tests

As anticipated in the introduction paragraph, asymmetrical shaft power increase during turning circle maneuvers at different speed and rudder angle has been already considered in preliminary works, which provided a set of data from different naval ships (Viviani et al 2007a).

In the following, summary of results obtained is presented; in particular, stabilized power increases obtained for all ships are summarized in figures 2 and 3 for internal and external shafts respectively as a function of rudder angle, in correspondence to different ship speed (effect of ship speed proved to be rather limited).

In these figures, experimental data are reported together with best-fit curves (linear in correspondence to external shaft, quadratic in correspondence to internal shaft), and a band indicating a range of plus and minus 10%. As it can be seen, despite data present obviously a certain scatter, a rather clear tendency is found.

Stabilized power increase (recorded during stabilized part of the turn) at maximum rudder angle, excluding the most disperse data, ranges from about 85% to about 105%, with a mean value of about 95%, and peaks up to

120% for external shaft,, from 30% to 50%, having a mean value of about 40% and peaks up to 60% for internal shaft. Peak power increases, recorded in correspondence to some maneuvers, resulted about 10-15% higher than stabilized ones, for both shafts.

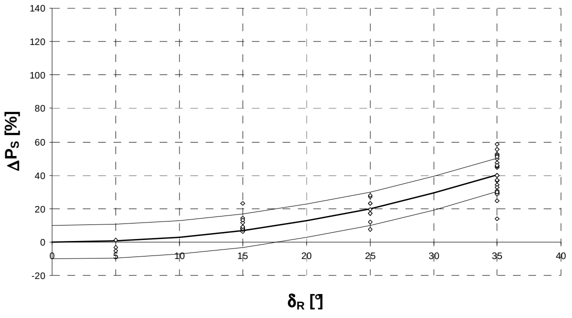

Fig. 2: Internal shaft – Stabilized power increase (Viviani et al 2007)

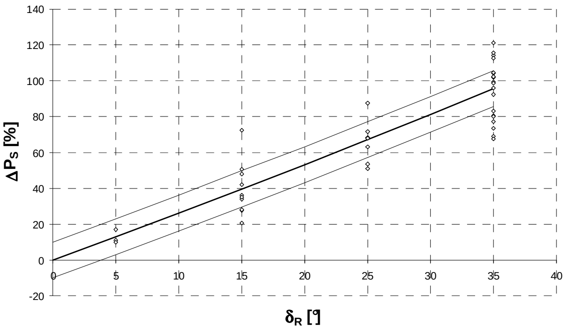

Fig. 3: External shaft – Stabilized power increase (Viviani et al 2007)

In past works, the approach used to analyze the shaft power increases was the "asymmetrical wake fraction variation" during turn. The process of asymmetrical loading is summarized in following figure 4.

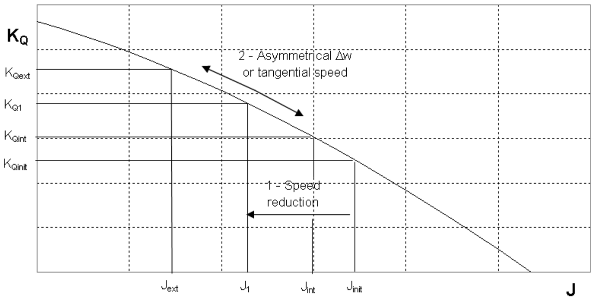

Fig. 4: Asymmetrical variation of advance coefficient J during manoeuvres (Viviani et al 2007)

In particular, two effects are superimposed during maneuvers, i.e.:

- a first symmetrical variation of advance coefficient due to speed reduction in the turn
- an asymmetrical variation of advance coefficient, which results in asymmetrical loading of shaftlines

The second effect might be attributed to different causes, i.e. longitudinal and/or tangential speed variation. After some analyses, it was found more convenient to consider only an equivalent longitudinal speed variation, by means of an asymmetrical wake fraction variation. This approach was applied for a ship and validated against different full scale trial results (including different maneuvers such as ZigZag maneuver), allowing to conclude that asymmetrical wake fraction variation is function of drift angle rather than of rudder angles (Viviani et al 2008). This difference is not evident when analyzing turning circle maneuver, but becomes clear when more unsteady maneuvers are considered, such as ZigZag.

It has to be remarked that trials planned in present research project will provide more data (and specifically thrust and torque time histories), thus allowing in principle to analyze other effects (such as possible asymmetrical thrust deduction factor).

Another important issue is represented by scale effects; in Viviani et al 2007b a comparison of results from free running model tests and sea trials, for a ship whose type and configuration are the same of the ship object of present study, was reported, allowing to have a first insight into this problem.

In figures 5 and 6 results are reported, showing that, at least for the ship analyzed, power increases tend to be underestimated during free running model tests, with values lower by about 10-15% in correspondence to maximum rudder angle for both external and internal shafts.

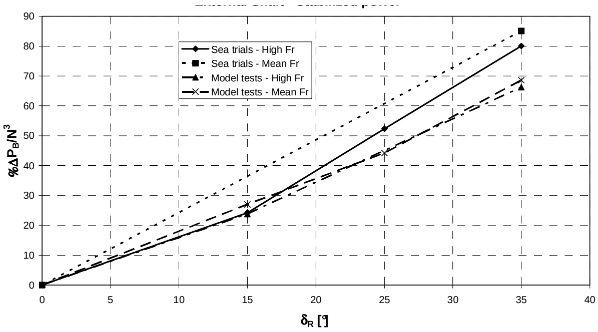

Fig. 5: Sea Trials and Model Tests results comparison – External Shaft – Ship 6 (Viviani et al 2007)

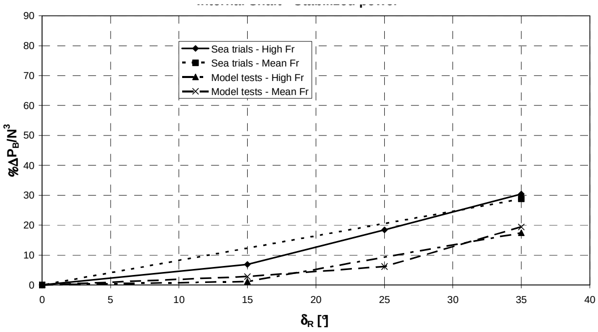

Fig. 6: Sea Trials and Model Tests results comparison– Internal Shaft – Ship 6 (Viviani et al 2007)

Unfortunately, this result is the only one available at the moment, thus a more comprehensive analysis will be needed in future to confirm it. Experimental results in model scale of present research project, together with future sea trials of the ship, when available, will represent a first validation of this trend, and this will allow also to evaluate possible scale factors in both wake fraction and thrust deduction factor (and in drift angle during manoeuvre, which in its turn affects them) , which are likely to be the reason for differences shown in figures 5 and 6.

Maneuverability and Propulsion Simulator

Ship selected for present analysis is a twin screw naval ship, similar to those analyzed in previous studies. In following table 1, main ship characteristics are reported, where L is ship length, B is ship beam, T is draft, C_B is block coefficient and A_R is total longitudinal projected rudder area.

L/B	7.531
B/T	3.286
C_B	0.51
A_R/LT	3.2%

Table 1: Main ship characteristics

In the following, a brief overview of the simulator developed at DINAEL is reported.

Brief Overview

This simulator consists of a set of differential equations, algebraic equations and tables that represent the various elements of the propulsion system and the ship maneuverability behavior, namely automation, engines, propellers, shaft lines, rudders, hull forces and interactions between different elements.

Solving numerically the differential equations allows to obtain time histories of propulsion system behavior (power, torque, RPM, etc.) and of maneuverability (in particular, the three degrees of freedom considered are surge, sway and yaw). The implementation of the numerical code has been made in MATLAB-SIMULINK® software environment, a wide used platform for the dynamic systems simulation.

Detailed information about the entire structure of the ship simulation model can be found in Benvenuto et al 2003 and Altosole et al 2008, while in Viviani et al.2008 a first modification of the model in order to consider separated shaftlines was described.

In the present work, two simulators have been developed, with different propulsion system characteristics, keeping on the contrary equal the maneuverability part.

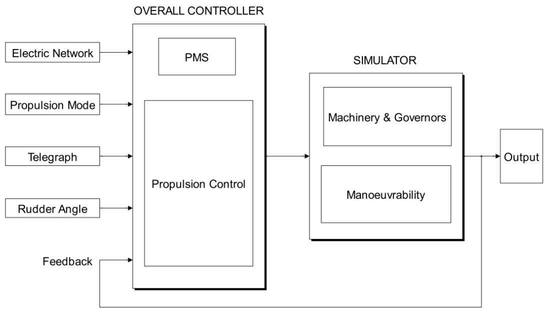

Fig. 7: Simulator functional scheme – free running model (model 1)

In particular, the first simulator (Figure 7) represents the free running model which will be used for experimental campaign (with FPP, electrical motors and a simplified automation system).

The second simulator (figure 8) includes characteristics of the real ship (with CPP, effective propulsion configuration and an automation system more similar to real); in the figure, only an overall view is provided.

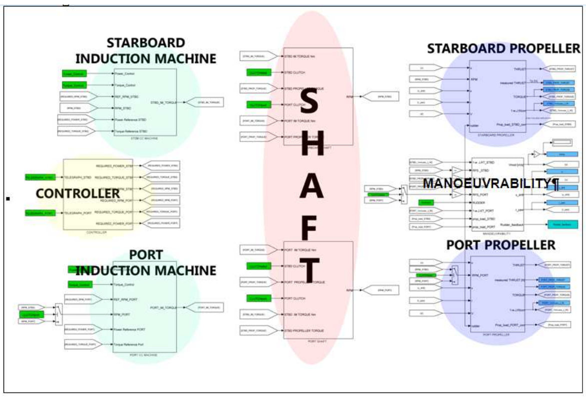

Fig. 8: Simulator functional scheme – full scale ship (model 2)

For each element illustrated in Figures 7 and 8, numerical models with different level of accuracy have been developed, taking into account the general objective of a good balance between the reliability of the simulation results and the code performance.

Propulsion system part

Propulsion plant dynamics is considered in a simplified way in both simulators; in particular, each shaftline dynamics is represented by the differential equation:

$$2\pi J_p \frac{dn(t)}{dt} = Q_e(t) - Q_p(t) \qquad (1)$$

J_p= polar moment of inertia;
Q_e= engine torque;
Q_p= propeller torque;
n =shaft speed;

Propellers are FPP (as usual for model tests) in model 1 and CPP in model 2; in both cases open water characteristics are given.

Regarding prime movers, in model 1 electric motors are considered only from the point of view of their possible different controls, i.e. constant RPM (thus following torque from propellers), constant torque or constant power.

In order to achieve this, a PID controller is used, where the controlled parameter is engine torque and the error monitored is alternatively RPM, torque or power, depending on the setup chosen.

In model 2, electric motors are modeled considering the maximum torque and the different control strategies, i.e. constant speed or constant power.

Gas Turbine is modeled considering the maximum torque and the fuel consumption map over the entire working range; also the Turbine Control System is modeled in order to allow the speed reference control and the different protections (i.e. overtorque, etc.).

Gearbox is taken into account only by the reduction ratio and the inertia, couplings are considered in order to model all the possible propulsion configurations.

Moreover, a complete automation system is also included in the model, with ‘high level’ propulsion control and subsystems controllers.

In order to be able to consider also configurations with coupled shaftlines, both models are modified, resulting in one unique differential equation with two driving torques and two propeller torques (plus frictional losses due to shaftlines mechanical coupling and bearings).

Maneuverability part

Maneuverability equations adopted in the simulator are the usual ones reported in the following:

$$\text{Surge: } \sum F_x = m(\dot{u} - vr) \qquad (2)$$

$$\text{Sway: } \sum F_y = m(\dot{v} + ur)$$

$$\text{Yaw: } \sum M_z = I_{zz}\dot{r}$$

u,v = ship speed in surge and sway directions;
r =ship rotation speed ;
m= ship mass;
I_{zz} = ship inertia moment about z-axis;
F_x =forces acting on the ship in x-axis direction;
F_y = forces acting on the ship in y-axis direction;
M_z = moments acting on the ship about z-axis;

Regarding hull forces and moments, a comprehensive description of them is reported in Viviani et al. 2009. In particular, regression formulae dedicated to twin screw vessels were obtained starting from Ankudinov model (Ankudinov 1996) and correcting it in order to consider appendages effect. Regarding rudder forces, model described in Viviani et al. 2009 is adopted, with further corrections on the basis of Molland and Turnock 2006.

Asymmetrical behaviour of shaftlines

Asymmetrical behavior of shaftlines is taken into account, as anticipated, by means of introduction of asymmetrical variations of wake fraction during maneuvers for the two shaftlines, as already introduced in Viviani et al. 2008. The model is also developed in order to consider a second asymmetry, i.e. thrust deduction factor, since model test data will allow to evaluate it in addition. In particular, during maneuvers, values of coefficients Δw and Δt may be computed for each shaftline, as functions of ship speed and ship drift angle, and then they may be added to values in straight motion as obtained from usual self propulsion tests.

Effective J value for each shaft is given as:

$$J = \frac{u(1-w-\Delta w)}{nD} \qquad (3)$$

As a consequence, J value is different for internal and external shaft, and thus different K_T and K_Q values result.

Furthermore, a second correction of computed thrust is obtained by means of the asymmetrical thrust deduction factor.

$$T_{eff} = T(1-t-\Delta t) \qquad (4)$$

Values of Δw and Δt may be obtained analyzing results of model tests with a process similar to self propulsion tests analysis. In present work results have been computed considering only a first set of asymmetrical wake fraction values Δw obtained on the basis of free running model tests carried out with a preliminary (and different scale) model, not dedicated to the analysis of asymmetrical shaft behavior. No asymmetrical thrust deduction factor is applied in this case, since it was not possible to compute it.

Simulations with different automation control – Model 1

In present paragraph, preliminary results obtained with model 1 simulator in correspondence to different propulsion system behaviors (constant RPM, constant torque, constant power) are reported.

In following figures 9-11, internal and external engine torque, power and RPM time histories during 35° turning circle maneuvers from model speed equivalent to a Froude number of 0.26 are reported. It is worth mentioning that in this case separated shaftlines are considered, thus propeller torque is, apart short transients, equal to engine torque.

As it can be seen, moving from constant RPM to constant power and constant torque results in a progressively reducing value of shaft revolutions during maneuvers, and contemporarily in a reduction of power and torque increases.

In correspondence to constant RPM control, asymmetrical shaft power increases are about 60% and 30% for external and internal shafts respectively. These values are in the lower range of those obtained with sea trials analysis, and more similar to those obtained with previous model tests, even if with a lower asymmetry between external and internal shaft. This result seem to confirm the tendency of model tests (on the basis of which the simulator model was preliminarily calibrated) to underestimate shaft power increase.

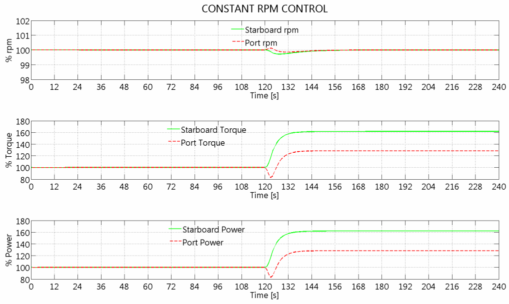

Fig. 9: Constant RPM control (Model 1)

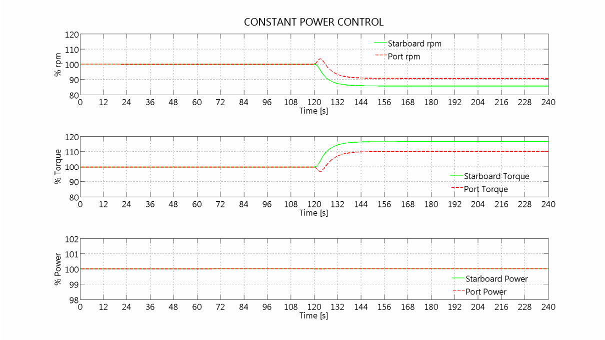

Fig. 10: Constant torque control (Model 1)

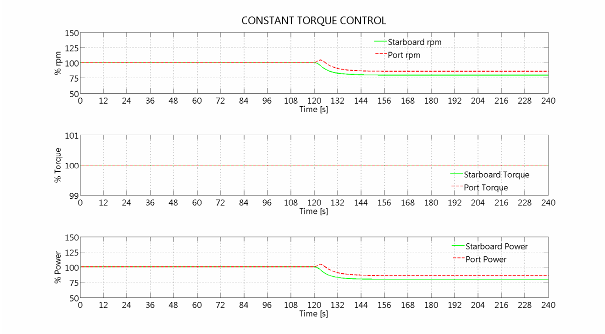

Fig. 11: Constant power control (Model 1)

From the point of view of ship maneuverability, in following figures 12-14 time histories of ship speed, sway velocity and angular velocity respectively are reported for all cases considered, while in figure 16 ship trajectories are compared.

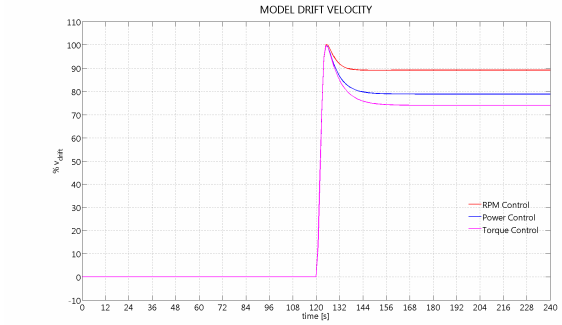

Fig. 12: Ship speed time histories (Model 1)

As it can be seen, main difference between various cases considered is stabilized ship speed during maneuvers, which, correspondingly to shaft revolutions, tends to reduce moving from constant RPM control to constant power and constant torque configurations.

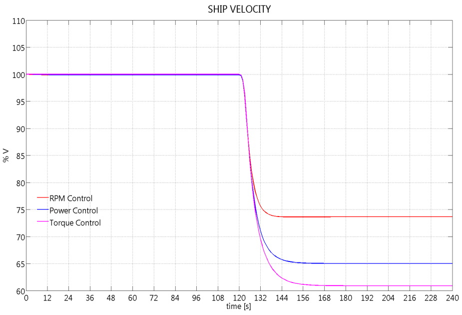

Fig. 13: Sway speed time histories (Model 1)

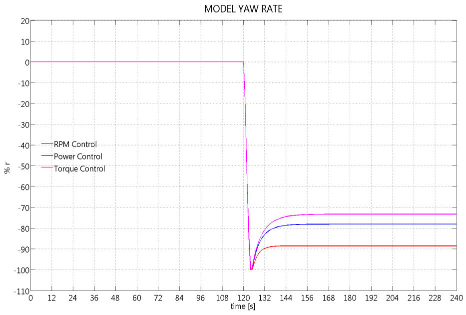

Fig. 14: Angular speed time histories (Model 1)

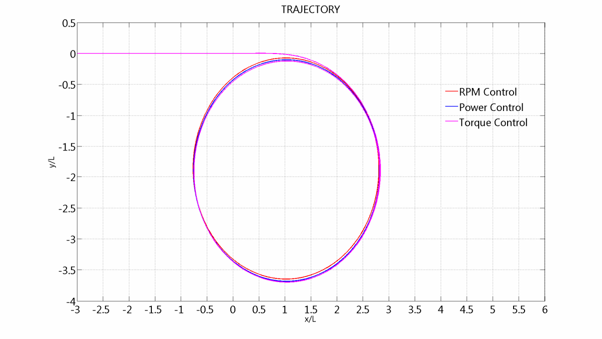

Fig. 15: Trajectories (Model 1)

Correspondingly, also sway velocity and angular velocity are reduced, almost proportionally to ship speed. This uniform reduction, in its turn, results in very small variation of ship trajectory, which results in a slightly reduced turning circle for constant RPM setting, even if differences are negligible.

Possible effect of shaft coupling has also been considered. In particular, same simulations have been carried out considering coupled shafts. In general, shaft coupling results in an equal behavior of the two electrical motors and in asymmetrical behavior of propellers. This behavior is due to the fact that propeller torque is not forced anymore to be in equilibrium with the correspondent engine, since the two shaftlines behave as a unique one with two driving motors and two propellers.

As an example, in following figure 16 difference between coupled and separated shaftlines in terms of torque in correspondence to constant RPM setting is reported. In this case, being prime mover and propeller torque and power different, they are both plotted.

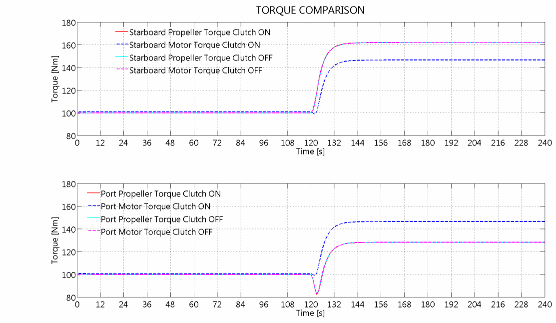

Fig. 16: Constant RPM control:.separated vs coupled shaft configurations: torques (Model 1)

As it can be seen, motor torques (dotted blue lines) vary in the two cases with respect to other values (being the mean value when shaftlines are coupled).

Propeller torques, on the contrary, are not modified in the two cases. As a consequence (see figure 17 with ship speed) maneuverability behavior is not modified significantly. Similar conclusions can be drawn also in correspondence to other cases.

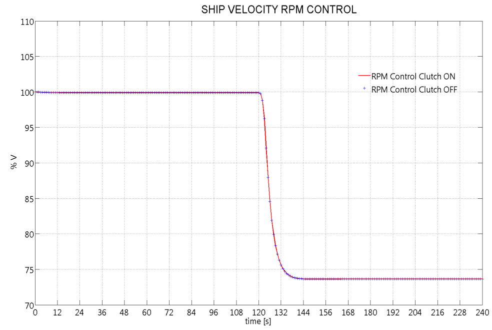

Fig. 17: Constant RPM control: separated vs coupled shaft configurations: ship speed (Model 1)

Simulations in full scale (Model 2) – Influence of automation system

In present paragraph, preliminary results obtained with model 2 simulator are reported, showing some of the possible differences which can be encountered when moving to full scale ship. In particular, two different controls are considered, i.e. constant RPM control without asymmetrical load compensation (as in one of the cases in model scale) and control with asymmetrical load compensation.

The first configuration allows to show possible differences with model tests due to different functioning point of propellers (no compensation for higher propeller load is included in the model), CPP instead of FPP, different combinatory settings, etc.

Second configuration, moreover, shows the possible differences in terms of shaft loading with different (and more complex) automation strategy, and how these can affect maneuverability parameters.

In following figure 18, a comparison of torque in the two configurations considered is reported.

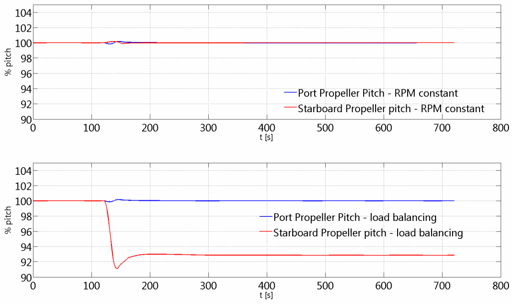

Fig. 18: Constant RPM control vs load balancing: comparison of torque time histories (Model 2)

As it can be seen, in the case of constant RPM control, behavior is very similar to the one obtained with model tests apart small differences; this is due to the fact that ship speed considered presents a certain margin with respect to MCR (and other limits), thus, also in full scale, the ship is capable of sustaining torque increases. On the contrary, influence of automation when load balancing is present is evident, with smaller (and almost equal on the two shafts) increase of torque during maneuvers.

This results in an asymmetrical variation of propeller pitches, as reported in following figure 19.

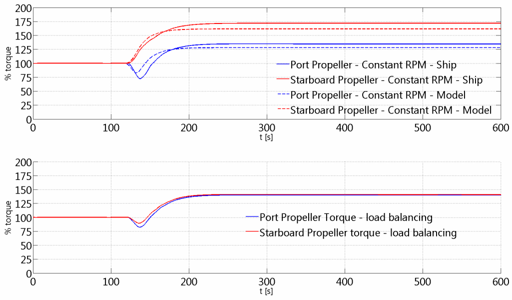

Fig. 19: Constant RPM control vs load balancing: propeller pithces time histories (Model 2)

Regarding trajectories, also in this case differences are very limited., probably because no scale effect on maneuvering coefficients (and wake fraction variations, even if less important for global trajectories) is assumed; further analyses should be carried on full scale trials, when available, in order to analyze this phenomenon. Since many differences are already present in the model, it can be expected that system identification techniques may provide interesting information, without being affected by large model approximations.

Finally, results in terms of ship speed in different configurations are reported in following figure 20.

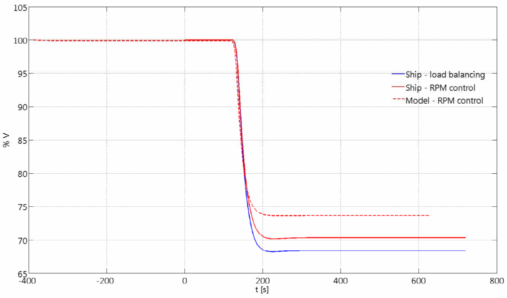

Fig. 20: Constant RPM control vs load balancing: ship speed

More significant differences are visible in this case, similarly to what already obtained with model 1 in correspondence to different control strategies. In particular, in model scale a smaller speed reduction is computed, due to the higher resistance given by scale effect, which results in a comparatively lower added resistance in turn. Considering the two control setting in full scale, load balancing, reducing pitch on one shaft, results in a lower ship speed.

Future work: experimental test matrix

As anticipated in previous paragraphs, in present research project a systematic series of free running model tests (under development at time of writing) has been scheduled, testing three different control settings (constant RPM, constant torque, constant power).

Main characteristics of the model used in present work are reported in the following table 2, showing considerable size of model adopted in the experimental campaign.

Dimensions	
L (model scale)	abt 7.2 m
Δ (model scale)	abt 1100 kg
Propulsion / control	
Propellers	2 FPP
Electrical power generation	1 Fischer Panda PMS 12000NE
Main drives	2 Mavilor BLS-143
Rudders	Twin spade rudders

Table 2: Model characteristics

Following free running model tests are planned:

- 3 propulsion system simplified automations (constant RPM, constant torque, constant power)
- 2 ship speeds for each configuration (namely cruising speed and high speed)
- Maneuvers at each speed / automation:
    1. Turning circle maneuvers (±35°, ±25°, ±15° rudder angle)
    2. ZigZag maneuvers (10°/10° and 20°/20°)
    3. Dieudonnè spiral maneuver

Moreover, in correspondence to previous configurations, shaftlines may be totally independent or connected with each other and forced to maintain same RPM during maneuver.

Conclusions

In the present work, developed in the context of a research project including both simulations and model tests, the problem of asymmetrical shaft loading during maneuvers has been considered.

In particular, simulation results in correspondence to different possible configurations (model scale and full scale, with different propulsion system characteristics and different automation control) have been presented, showing possible differences.

From results obtained, it is clear that different configurations do not result in large differences in terms of maneuverability macroscopic characteristics (turning circle trajectories), neither considering different automation controls, nor considering differences between model and full scale. Main differences obtained are related to ship speed, with reduces differently depending on control system adopted.

These results will be verified by means of model test campaign under development, which will allow to confirm asymmetrical shaft loading model, and to improve it by means of the introduction of asymmetrical thrust deduction factor. Moreover, influence of possible different settings of automation on shaft overloading will be further analyzed, in order to confirm the “generality “ of asymmetrical shaft loading coefficients.

This activity will allow to have a further improved model to check automation system in full scale, whose effect has been preliminarily tested and presented in this paper. In particular, load balancing mode seems to act properly, and to be able to avoid unwanted asymmetrical loading on the reduction gear.

Further development of this work will be obtained when results from sea trials will be available, allowing to check the possible scale factors on asymmetrical shaft loading, for which currently a significant lack of data exist.

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