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NUMERICAL MODELS FOR SHIP DYNAMIC POSITIONING

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Abstract. One of the best ways to design marine control systems before the construction of a ship is the use of simulation techniques. The paper presents two numerical models used to develop and test, in the preliminary design phase, a dynamic positioning (DP) system for marine vehicles. In particular, it refers to a surface vessel equipped with a conventional propulsion configuration, consisting of two controllable pitch propellers, two rudders and a single bow-thruster. For such a vessel, the DP system is required to manage the actuators in order to obtain a good dynamic positioning performance at zero-speed with moderate weather conditions. In order to verify control and allocation logics, two numerical models, with different degrees of details, have been developed and used in distinct steps. Several sub-systems as ship dynamics, propulsion plant, controller and environmental disturbances have been implemented making use of suitable mathematical models linked each other in order to take into account their mutual interactions. Eventually, simulation results are shown and critically compared in order to better understand the points of strength and weakness of the two proposed models.

1. INTRODUCTION

Dynamic Positioning has come a long way in the 50 years since its first installation, developing alongside the oil industry. Today its applications are as varied as the vessels it is installed upon and its technology has found its way into all aspects of the marine industry. Hundreds of scientific works related to this topic have been published. From the control point of view, relevant works [1-3] inspired the proposed methodology. The effectiveness of using simulation techniques to study the transient behaviour of the propulsion plant and the ship dynamics is proved in several works present in open literature [4-8]. Combining the state of the art in ship modelling with the controller design results to be the best ways to design DP system before the ship availability. This allows to reduce the time and the cost of full scale sea trial tests. In the present paper the methodology used by the authors to develop and test, in the preliminary design phase, a dynamic positioning system for marine vehicles is shown. The ship taken into account in this study is a patrol vessel which was designed with a conventional propulsion system (twin propeller-rudder configuration and a bow-thruster) but was requested to provide, as a retrofit, a certain dynamic positioning performance at zero-speed with moderate weather conditions. This kind of propulsion configuration is clearly a disadvantage as far as station-keeping and dynamic-positioning are concerned. Nevertheless, a conventional propulsion configuration could be requested for specific operations to offer some dynamic positioning capability, albeit limited.

The proposed methodology is subdivided in two distinct steps, the first one is the implementation of a simplified dynamical model, needed in the preliminary design of the controller in order to test the effectiveness of the implemented control and allocation logics. Such model has been specifically designed in order to assess both the regulation strategies and the station keeping capability of the vessel with more realistic results than those obtained by a purely static analysis. In this model, the interaction between rudder and propeller have been modelled in a simplified way and the actuators have been modelled through first order transfer functions based on manufactures data, omitting a more detailed description of the engines dynamics.

The second step envisages some simulation tests about the behaviour of the proposed regulator through a more detailed numerical model, to obtain a better assessment of the system performances. In this simulation platform the actuators, the rudders and the propellers have been modelled in a more physical way together with their mechanical constraints. The dynamics of the main engines and their governors have been implemented too. In both the steps, the same mathematical description have been adopted for wind, wave, and current forces, as well as for the controller. In the next sections the main equations, used in both the developed simulation platforms, are reported as well as the corresponding simulation results.

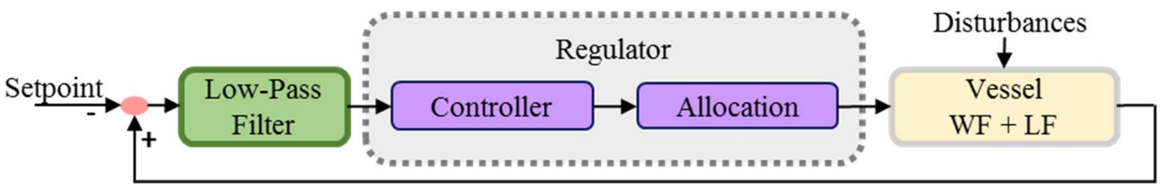

Figure 1. DP simulation model

2 SHIP MOTIONS

In this section, both low and high frequency motion models are presented: two different levels of model detail are compared for low frequency motions, while for high frequency motions the same model is used for both the plants.

2.1 Kinematic

Let $\boldsymbol{\eta} := [x, y, \psi]^{\boldsymbol{T}} \in \mathbb{R}^3$ be the array expressing the position and orientation of the vessel with respect to the Earth-Fixed Frame, and let $\boldsymbol{\nu} := [u, v, r]^{\boldsymbol{T}} \in \mathbb{R}^3$ be the array containing the linear and angular velocity components with respect to the Body-Fixed basis. The kinematic relation between $\boldsymbol{\eta}$ and $\boldsymbol{\nu}$ is given by:

$$\dot{\boldsymbol{\eta}} = \boldsymbol{R}(\psi)\boldsymbol{\nu} \qquad , \qquad \mathbf{R}(\psi) = \begin{pmatrix} \cos\psi & -\sin\psi & 0 \\ \sin\psi & \cos\psi & 0 \\ 0 & 0 & 1 \end{pmatrix} \tag{1}$$

2.2 Dynamics

As far as dynamics is concerned, two models are considered. The first one is a simplified model where only added masses and cross-flow drag are taken into account. The notation $\boldsymbol{\tau} := [\boldsymbol{X}, \boldsymbol{Y}, \boldsymbol{N}]^{\boldsymbol{T}} \in \mathbb{R}^3$is introduced for the array containing the components of longitudinal and lateral forces and their resultant moment. The resulting ship motion equations are:

$$\boldsymbol{M}\dot{\boldsymbol{v}} + \boldsymbol{D}(\boldsymbol{v})\boldsymbol{v} = \boldsymbol{\tau_D} + \boldsymbol{\tau_E} \tag{2}$$

The second and more detailed model is given by:

$$\boldsymbol{M}\dot{\boldsymbol{v}} + \boldsymbol{C}(\boldsymbol{v})\boldsymbol{v} + \boldsymbol{D_0}\boldsymbol{v} + \boldsymbol{D}(\boldsymbol{v})\boldsymbol{v} = \boldsymbol{\tau_D} + \boldsymbol{\tau_E} \tag{3}$$

Where $\boldsymbol{M}$, $\boldsymbol{C}$ and $\boldsymbol{D}$ represent the mass-inertia and added mass, Coriolis and damping matrices respectively. $\boldsymbol{\tau_D}$ are the delivered forces and moments and $\boldsymbol{\tau_E}$ represent the environmental forces and moments.

3 ENVIROMENTAL FORCES

The environment action is modelled as the superposition of the low-frequency sea and wind effects. In particular, the disturbances array $\boldsymbol{\tau}_{\mathbf{E}}$ is expressed as the sum of three addends:

$$\boldsymbol{\tau}_{\mathbf{E}} = \boldsymbol{\tau}_{\mathbf{current}}(\gamma_r) + \boldsymbol{\tau}_{\mathbf{waves}}(\gamma_r, H_s) + \boldsymbol{\tau}_{\mathbf{wind}}(\gamma_r, v_G) \tag{4}$$

Forces are usually written in terms of non-dimensional coefficients $C_X(\gamma_r)$, $C_Y(\gamma_r)$, and $C_N(\gamma_r)$, where γ_r is the angle between the vessel bow and the main incoming force direction. In particular, the current velocity is assumed constant; the wind has both mean speed and gusts generated in accordance to Davenport spectrum; wave drift forces are modelled as proportional to the square of the significant height H_s, such height being originated from the envelope of the wave elevation time history generated in accordance to JONSWAP spectrum.

4 CONTROLLER LAYOUT

The overall controller layout is shown in Figure 2. It consist in four main: Proportional and Derivative (PD) controller, wind forces reconstruction (Wind), sea force estimation ($\overline{\text{PD}}$) and allocation. The detail of the blocks modeling are explained in the next paragraph.

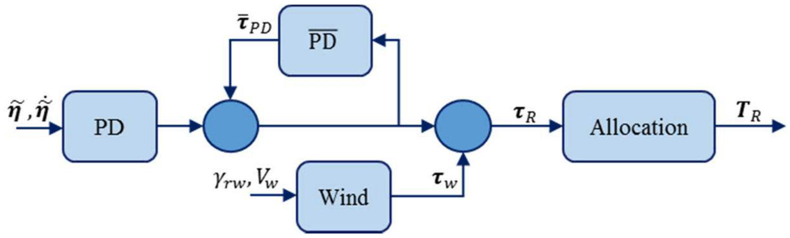

Figure 2. Controller Logic

4.1 Controller

This block is actually the kernel of the simulator, because it contains both the control logic and the allocation algorithm. Inputs are the errors of position and velocity as well as the delivered forces, while the required thrusts are the outputs.

The controller consists of a PD-controller. The controller input errors are defined as: $\tilde{\boldsymbol{\eta}} := \boldsymbol{\eta} - \boldsymbol{\eta_d}$ and $\dot{\tilde{\boldsymbol{\eta}}} \equiv: = \dot{\boldsymbol{\eta}} - \dot{\boldsymbol{\eta}}_{\boldsymbol{d}}$, where $\boldsymbol{\eta_d} = [x_d, y_d, \psi_d]^{\boldsymbol{T}} \in \mathbb{R}^3$ and $\dot{\boldsymbol{\eta}}_{\boldsymbol{d}} = [\dot{x}_d, \dot{y}_d, \dot{\psi}_d]^{\boldsymbol{T}} \in \mathbb{R}^3$ represent the desired position and velocity (setpoints). The controller law is then the following:

$$\boldsymbol{\tau_R} = \boldsymbol{K_P}\tilde{\boldsymbol{\eta}} + \boldsymbol{K_D}\dot{\tilde{\boldsymbol{\eta}}} + \bar{\boldsymbol{\tau}}_{\boldsymbol{PD}} - \boldsymbol{\tau_W} \tag{5}$$

Where $\boldsymbol{K_P}$ and $\boldsymbol{K_D}$ are constant diagonal matrices. The integral term, usually present in conventional PID-controllers, is here replaced by an estimation of the mean values of the environmental disturbances, performed by means of a moving average of the required forces, every predetermined time interval Δt.

Disturbances are split into two contributions: wind $\boldsymbol{\tau_W}$ and sea $\bar{\boldsymbol{\tau}}_{\boldsymbol{PD}}$. On board, wind forces and moments are reconstructed from the wind sensors signals. Such sensors provide for both the main incoming wind direction and speed. In the simulator, a Gaussian White Noise falsifies both mean wind speed and direction, computed as briefly outlined in Section 3. The sea force and moment are computed by means of an average procedure of the registered PD output signals in a certain time interval Δt. During the first transient time interval Δt, only wind action is compensated. As well as the transient is expired, the residual main components are added to the controller output. Summarizing, wind action is directly compensated; the rest of disturbances is estimated by the controller and, then, compensated.

4.2 Allocation

The thrust allocation logic has been conceived for the layout of thrusters and rudders available on board, compatible with the DP1 class rules requirements. This logic envisages adopting the rudders behind the propellers (uncoupled) in order to generate the required forces and moments. In particular, the allocation algorithm requires the utilization of one rudder (called DP rudder) while the other is kept fixed hard over. The DP rudder angle is computed based on the required forces. This implies that the DP rudder angle is not an unknown of the thrust allocation algorithm.

Then, the forces and moments balance is uniquely determined in an algebraic way except for some situations that require a special handling as shown in [9].

5 SYMPLIFIED PROPULSION MODEL

Although not thoroughly detailed, the simplified propulsion model provides a good approximation of the actuators response times measured by the shipyard.

By collecting the portside (PT), starboard (SB) and bow (BT) thrusts into the array $\boldsymbol{T} := [T^{(PT)}, T^{(SB)}, T^{(BT)}]^{\boldsymbol{T}}$, the actuators dynamics is modelled by the differential equation $\dot{\boldsymbol{T}}_{\boldsymbol{D}} = -\boldsymbol{h}(\boldsymbol{T_D} - \boldsymbol{T_R})$, being $\boldsymbol{T_D}$ the delivered thrusts array and $\boldsymbol{h} = diag\{h^{PT}, h^{SB}, h^{BT}\}$ the time constants provided by the shipyard.

In order to take the mechanical constraints on the actuators into account, a saturation law for requested thrusts $\boldsymbol{T_R}$, with upper and lower bounds, and rate limiters for the actuation speed are used.

6 DETAILED PROPULSION MODEL

The whole simulation platform representing the detailed system is shown in Figure 3.

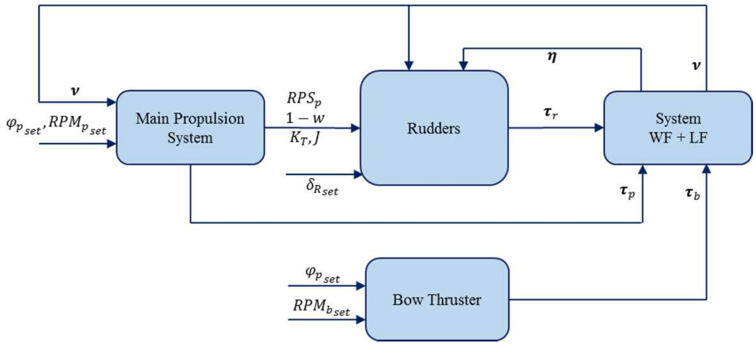

Figure 3. Detaild simulation model

The main elements of the propulsion plant modeled are two main engines, two gearboxes, and two controllable pitch propellers and one controllable pitch bow thruster.

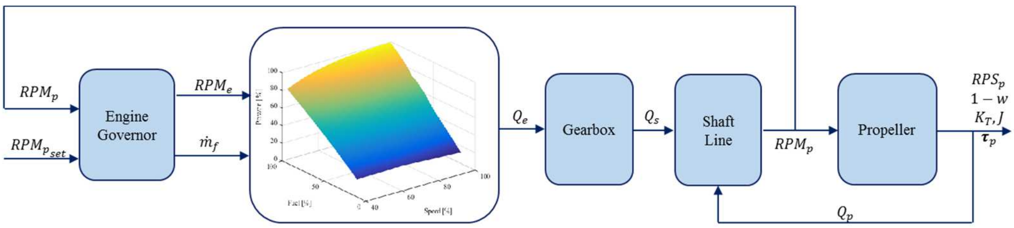

Figure 4. Propulsion plant model

6.1 Engine

The engine governor computes the fuel flow requirement on the basis of the propeller speed error. It is a PI controller with the following law
:

$$\dot{m}_f^r = K_{P_{RPM}} e_{RPM} + \int_0^t K_{I_{RPM}}\, e_{RPM}(\xi) d\xi \tag{6}$$

Where $e_{RPM} = \frac{100}{N_{MAX}}(N_e^r - RN_s^r)$ is the error from the setpoint coming from the controller and $K_{P_{RPM}}$ and $K_{I_{RPM}}$ are the constant coefficients of the regulator. Then, the fuel flow is saturated in accordance with the maximum engine torque limits:

$$\dot{m}_f = \begin{cases} \dot{m}_f^r & \dot{m}_f^r < \dot{m}_f^{(max)} \\ \dot{m}_f^{(max)} & otherwise \end{cases} \tag{7}$$

The engine is modeled by a response surface shown in Figure 4, the final output is the engine torque. The last is transformed in shaft torque through the reduction gear.

6.2 Shaft Line

Eq. (9) describes the shaft line dynamics:

$$\frac{d\omega}{dt} = \frac{Q_E(t) - Q_p(t) - Q_f(t)}{I_e + I_g + I_s + I_p} \tag{8}$$

Where: $I_e + I_g + I_s + I_p$ represent the engine, gear, shaft and propeller rotational inertia; Q_E , Q_P and Q_f are the engine, propeller and fictional torque, respectively.

6.3 Propeller

In the literature, several numerical methods based have been proposed to predict propeller hydrodynamic loads. Unfortunately, due to their long computational times, these methods are not suitable to be applied in the context of a time domain simulator like the one described in the present work. Therefore, in the proposed work the hydrodynamic forces, both for propeller and bow thruster,have been evaluated through a quasi-steady-state methodology based on the propeller open water tests. These tests provide an open water diagram, which allows the evaluation of the thrust coefficient K_T and torque coefficient K_Q.

For controllable pitch propellers, these coefficients depend on the advance coefficient J and from the blade angle φ:

$$K_T = \frac{T}{\rho n^2 D^4} \qquad\qquad K_Q = \frac{Q}{\rho n^2 D^5} \tag{9}$$

6.4 Rudder

For the same reasons as before, also in this case the rudder forces have been evaluated through a quasi-static-methodology, using lift and drag coefficients $C_L$and C_D. In addition, the complex interaction between rudder and propeller has been taken into account using the approach described in [5].

7 SIMULATIONS AND COMPARISON

In this section a comparison between the two proposed simulation models is discussed . For shortness reasons, only the most significant simulation outputs will be compared. In all figures, dotted lines represent the results of the simplified model, while the continuous ones refer to the detailed model.

Results are shown for both hold position and heading maneuvers, where all the external forces are aligned and coming from 45° with respect to North, and their intensity corresponds to Sea State 4. The vessel is required to keep its position and heading when the wind and current speeds are respectively 21 and 1 kn. Initial conditions for the simulation are representative of a real DP maneuver, where the system is thought activated when the vessel has approximately already reached the desired position and heading. The vessel is supposed to start such maneuver while it is at the origin of the reference Earth-Fixed Frame with the bow aligned with North direction. This is a critical maneuver because of the particular propulsion configuration, not conceived for DP applications. Indeed, for such propulsion plant, quartering seas, where longitudinal and lateral forces are comparable and the moment reaches its highest values, represent the worst situation to perform DP maneuvers.

Time history of the path of vessel mid-ship, red line, is shown in Figure 5 where simulation results for the simplified and the detailed models, are respectively shown on the left and the right hand sides. In motion plots, all quantities are represented as percentage of the maximum allowable errors. As it can be seen, both the models gives similar results regarding macro-behavior of the vessel during the maneuver, e.g. mean motions are comparable. Mid-ship path is kept with half of the maximum error for the whole maneuver and not so much higher oscillations can be detected in the detailed model.

The vessel motion can be throughout analyzed in Figure 6 where the time records of the surge, sway and yaw motions are reported. Also in the transients, errors never exceed their maximum values, and are usually comparable by means of errors amplitude. During the transient, only wind forces and moment are compensated and sea action is identified by the controller .

In Figure 7 the controller output about disturbances estimation is compared with the simulated disturbances forces. For readability purposes, disturbances are the dotted line; in both the models the controller estimates correctly the mean value of the disturbances.

An important difference between the two models is the computation of the hull forces and moments, as reported in Figure 8. While, probably due to the low velocities, not appreciable differences concerning the forces are arguable, the difference in the moment evaluation is significant and shows how, when yaw rates are different from zero, but still within the validity of the zero speed models, more detailed damping components give their contribution.

Regarding the differences in the propulsion plants, something important is shown comparing Figure 9, where delivered forces are plotted, and Figure 10 where delivered and required thrusts are reported. In fact, delivered forces are equal because they have been calibrated in both the models through on board tests. In the simplified model the way to deliver such forces and moments is missed. In particular, in this case of study a detailed model for the rudder makes the difference. Indeed, requirements are the same but delivered thrusts, by each thruster (portside propeller, starboard propeller, and bow thruster), are greater in the case of the detailed model.

Finally, the complete model allows to deeply study the behavior of the engines. In particular, the information regarding engines behavior could be used in order to study proper optimization logics in terms of fuel consumption. Figure 11 shows the working point in the engine load diagram for both starboard and portside shafts.

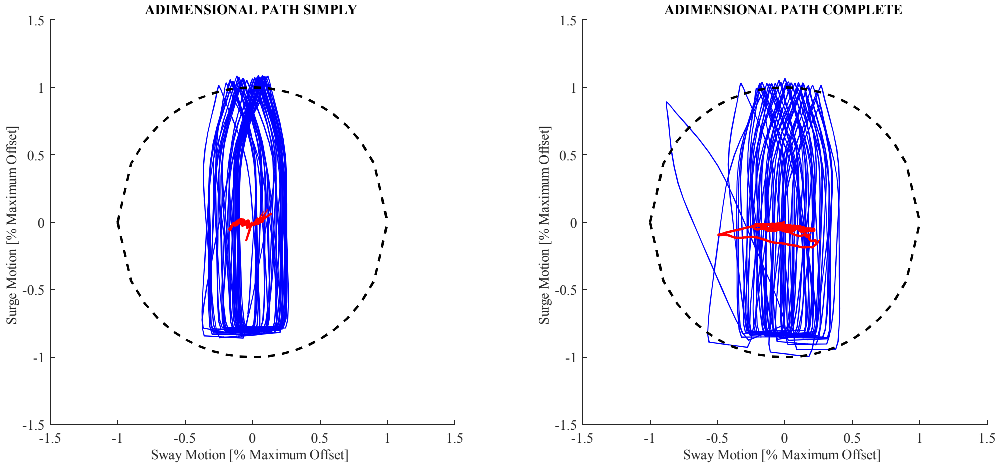

Figure 5. Ship trajectory

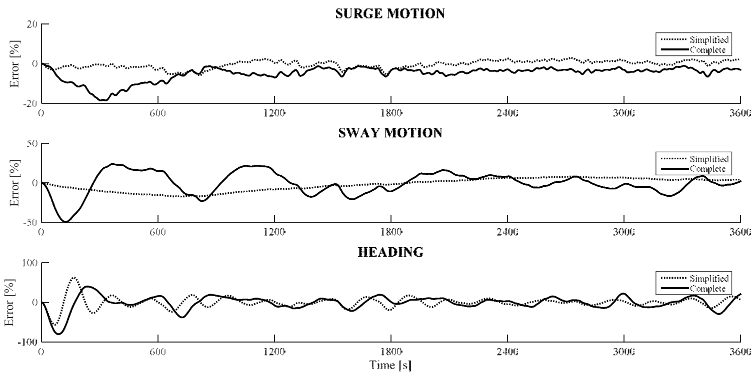

Figure 6. Ship motions

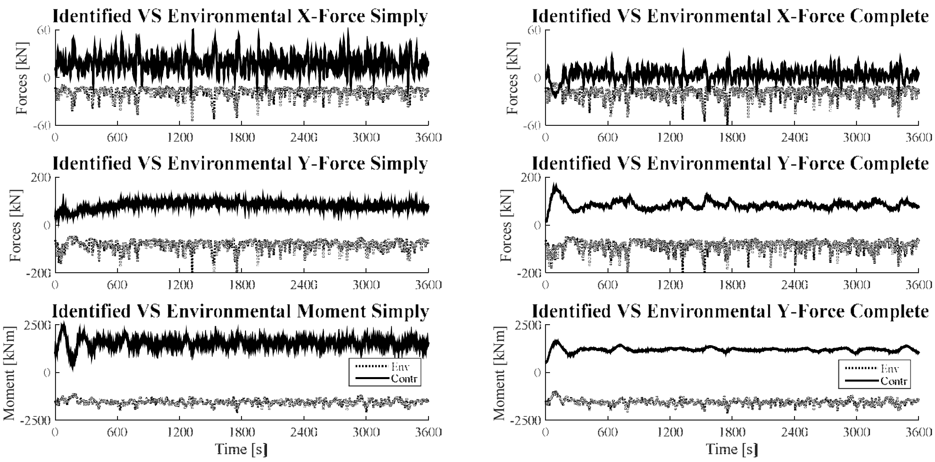

Figure 7. Enviromental forces and moment

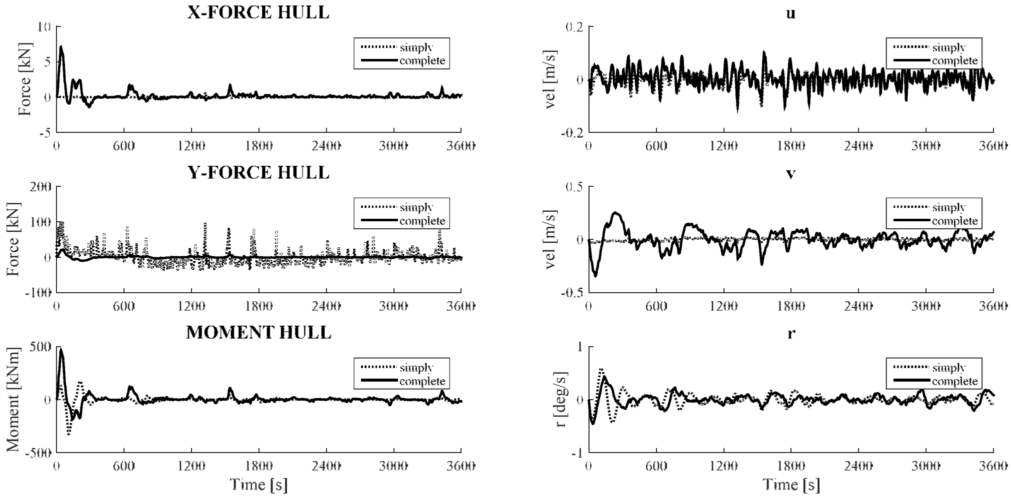

Figure 8. Hull forces and moment

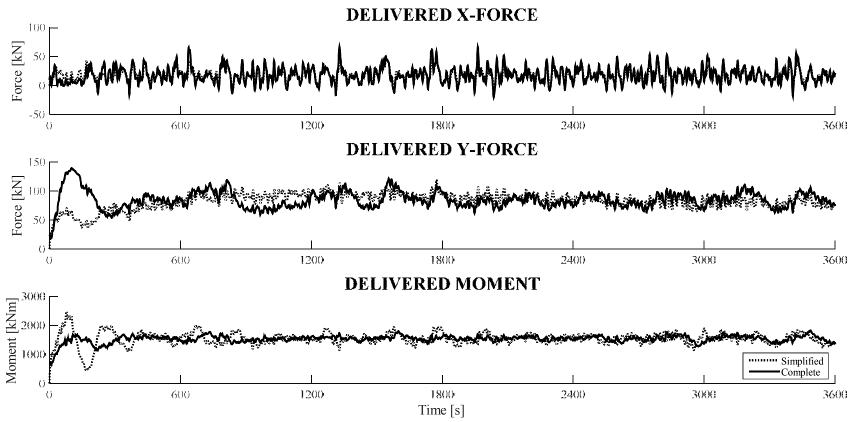

Figure 9. Delivered forces and moment

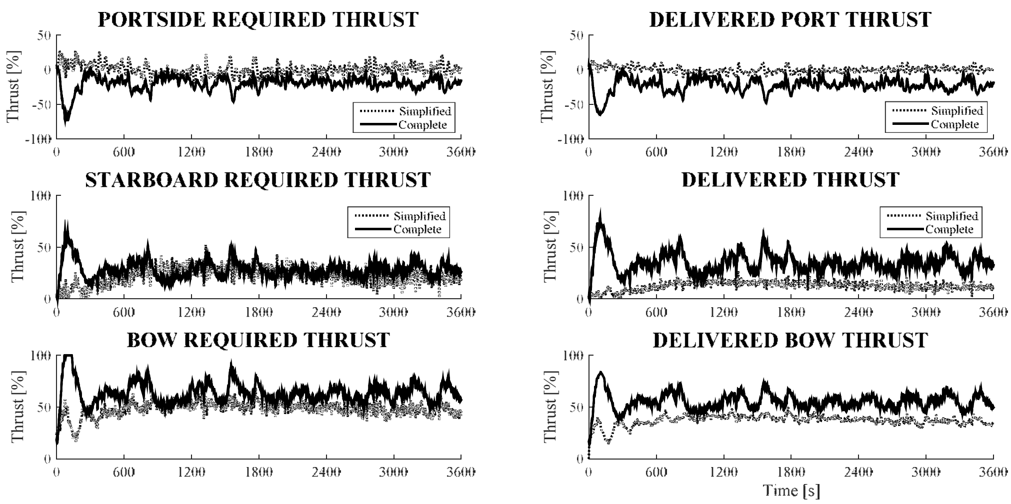

Figure 10. Required forces and moment

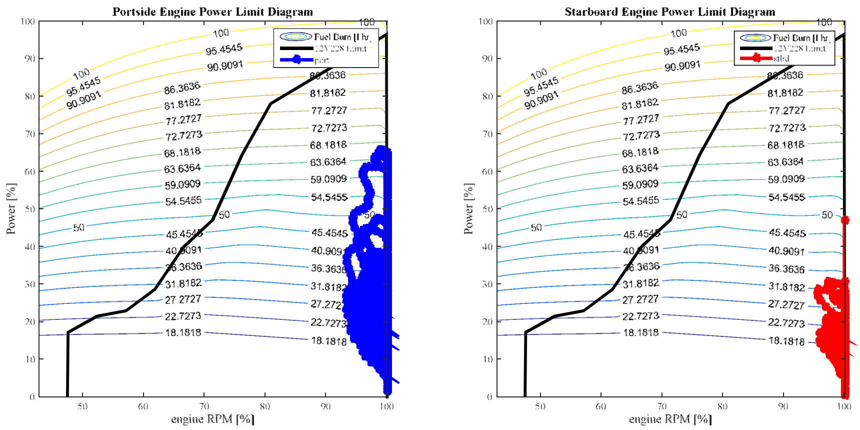

Figure 11. Engines working points

8 CONCLUSIONS

We have presented two different simulation models for testing the robustness of a DP regulator, devised to retrofit with a limited DP capability an existing vessel equipped with a conventional propulsion configuration. This case of study is particular because of the vessel propulsion plant that gives no degrees of freedom for the DP propulsion assessment. Benefits deriving from the adoption of a simplified model for testing control logics are pointed out by the substantial equivalence of the simulated DP performances yielded by both the models; of course also thanks to the accuracy of the saturation values provided by the shipyard.

Surely, a detailed model gives more precise information about the dynamic behavior of the whole propulsion line. This opens new challenges, for example to explore new control logics for fuel oil consumption optimization and to detect and forecast possible failures. Future work will concern a comparison with on-board calibration tests on the target ship after installation of the real controller.

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