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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 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 (Dynamic Positioning) 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 (DP) 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 work related to this topic have been published. Regarding the control point of view, relevant works published by Sørensen et al. [1-3] inspired the proposed methodology.

The effectiveness of using simulation techniques to study the transient behaviour of the propulsion plant and to study the ship dynamics is shown in several works present in open

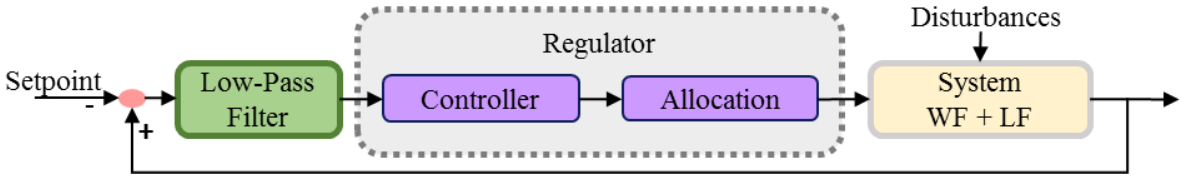
literature [4-8]. Combining the state of art in ship modelling with the controller design results to be the best ways to design DP system before the ship availability. This allows to reduce the time and the cost of full scale sea trial test.

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 take 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 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.

This proposed methodology is subdivided in two distinct step, the first step 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 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 hydrodynamic forces and moments have been considered linear and the interaction between rudder and propeller have been modelled in a simplified way. 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 developed simulation platforms, are reported as well as the simulation results.

Figure 1: DP simulation model



2 SHIP MOTION

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

2.1 Kinematic

Let, the array $\boldsymbol{\eta} := [\boldsymbol{x}, \boldsymbol{y}, \boldsymbol{\psi}]^{\boldsymbol{T}} \in \mathbb{R}^{\boldsymbol{3}}$ contains the positions and orientation with respect to the Earth-Fixed Frame, and $\boldsymbol{\nu} := [\boldsymbol{u}, \boldsymbol{v}, \boldsymbol{r}]^{\boldsymbol{T}} \in \mathbb{R}^{\boldsymbol{3}}$ contains the velocities and the rotation rate expressed with respect to the Body-Fixed Frame. Then the relation between the two basis is the following:

$$\dot{\boldsymbol{\eta}} = \boldsymbol{R}(\boldsymbol{\psi})\boldsymbol{\nu}$$

2.2 Dynamics

As far as dynamics is concerned, two models are presented. The first one is a simplified model where only added masses and cross-flow drag are taken into account. Let introduce the notation for the latter of to $\boldsymbol{\tau} \coloneqq [\boldsymbol{X}, \boldsymbol{Y}, \boldsymbol{N}]^{\boldsymbol{T}} \in \mathbb{R}^{\boldsymbol{3}}$as the array containing the longitudinal and lateral forces and their resultant moment as component, respectively.

$$\boldsymbol{M}\dot{\boldsymbol{\nu}} + \boldsymbol{D}(\boldsymbol{\nu})\boldsymbol{\nu} = \boldsymbol{\tau}_{\boldsymbol{D}} + \boldsymbol{\tau}_{\boldsymbol{E}}$$

The second …..

$$\boldsymbol{M}\dot{\boldsymbol{\nu}} + \boldsymbol{C}(\boldsymbol{\nu})\boldsymbol{\nu} + \boldsymbol{D}_{\boldsymbol{0}}\boldsymbol{\nu} + \boldsymbol{D}(\boldsymbol{\nu})\boldsymbol{\nu} = \boldsymbol{\tau}_{\boldsymbol{D}} + \boldsymbol{\tau}_{\boldsymbol{E}}$$

It will be referred to as $\boldsymbol{\Sigma}_{\boldsymbol{1}}$ for the simplified dynamic and to as $\boldsymbol{\Sigma}_{\boldsymbol{2}}$ for the detailed one.

3 ENVIROMENTAL FORCES

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

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

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

4 CONTROLLER LAYOUT

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 and the delivered forces and the required thrusts are the outputs.

The controller consists of a PID-controller. In this case of study is presented a PID controller independent for each axis. Then, by defining the controller input errors as: $\tilde{\boldsymbol{\eta}} \coloneqq \boldsymbol{\eta} - \boldsymbol{\eta}_{\boldsymbol{d}}$ and $\dot{\tilde{\boldsymbol{\eta}}} \equiv \tilde{\boldsymbol{\eta}} := \dot{\boldsymbol{\eta}} - \dot{\boldsymbol{\eta}}_{\boldsymbol{d}}$; where $\boldsymbol{\eta}_{\boldsymbol{d}} = [x_d, y_d, \psi_d]^{\boldsymbol{T}} \in \mathbb{R}^{\boldsymbol{3}}$ and $\dot{\boldsymbol{\eta}}_{\boldsymbol{d}} = \left[\dot{x}_d, \dot{y}_d, \dot{\psi}_d\right]^{\boldsymbol{T}} \in \mathbb{R}^{\boldsymbol{3}}$ are the desired setpoints. Then, the controller law could be the following:

$$\boldsymbol{\tau}'_{\boldsymbol{R}} = \boldsymbol{K}_{\boldsymbol{P}}\tilde{\boldsymbol{\eta}} + \boldsymbol{K}_{\boldsymbol{D}}\dot{\tilde{\boldsymbol{\eta}}} + \bar{\boldsymbol{\tau}}_{\boldsymbol{PD}} - \bar{\boldsymbol{\tau}}_{\boldsymbol{W}}$$

where $\boldsymbol{K_P}$ and $\boldsymbol{K_D}$ are constant diagonal matrices. In this case I-action was set "off" in order to avoid windup performance deterioration. It is substituted by an estimation of the mean disturbances values. In fact, in DP applications, such contribution is assigned to an estimation of the mean components of the environmental disturbances. In this study, disturbances are estimated by means of a moving average of the required forces from the PD controller, in a certain time interval ($\Delta \boldsymbol{t}$).

Then, disturbances are split into two actions: wind ($\bar{\boldsymbol{\tau}}_{\boldsymbol{W}}$) and sea ($\bar{\boldsymbol{\tau}}_{\boldsymbol{PD}}$) . Wind forces and moment are reconstructed from the wind sensors signals. Such sensors provide for both the main incoming direction and speed. On the other hand, the sea forces and moment are computed by means of the registration of the PD output signals in a certain $\Delta \boldsymbol{t}$. Practically, both mean wind and direction are falsified by a Gaussian White Noise and sent to the regulator where wind mean forces and moment are computed by the same procedure as shown in Chaprter…env.

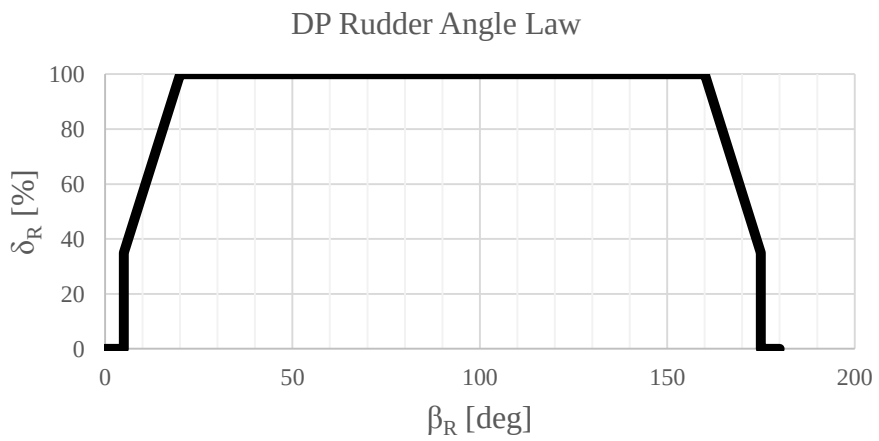
Then, during the first transient ($\Delta \boldsymbol{t}$), only wind action is compensated. As well as the transient is expired, the residual main components are added to the controller output. Summarizing, wind action is directly compensated; the rest is estimated by the controller and, then, compensated.

4.2 Allocation

The following description of the thrust allocation logic has been conceived for the layout of thrusters and rudders available on board: 2 CPP skew propeller, 2 flap rudders, and 1 bow thruster. This layout is compatible with the DP1 class rules requirements. No redundancy is fore viewed then the failure of each thruster do not allow the vessel to keep maintaining his position. This logic envisages to adopt the rudders behind the propellers (uncoupled) in order to generate the longitudinal and traversal required forces and the moments. In particular, the allocation algorithm requires the utilization of one rudder (called active rudder or DP rudder) while the other is kept fixed hard over. The main TAL criteria are the following: the propeller working with the DP rudder has to be always in forward running, due to the low performances of the rudder if down-line of a reverse running skew propeller. The second rudder is hard over then the propeller is able to deliver the required backward force. Theoretically, position keeping could be done by one fixed rudder but, from a performing point of view, it is better to use both the rudder. In order to guarantee the assistance of the forward running propeller torque:

- $N_{W|_{t=t_0}} \geq 0 \Rightarrow \mathrm{DP-rudder\ at\ PT}$
- $N_{W|_{t=t_0}} < 0 \Rightarrow \mathrm{DP-rudder\ at\ SB}$

From an energetic point of view, the best solution is found with helm hard over or maximum rudder angle. DP-rudder angle sign is led by the solution of the forces allocation algorithm in order to guarantee DP- propeller to be always forward running. The main idea is to keep the helm hard over till the required transverse forces or moment restrained in a preset sensitive interval (dead-band); in order to avoid the well know rudder bang-bang disease, an opportune ramp has been introduced in the rudder angle law as shown below.

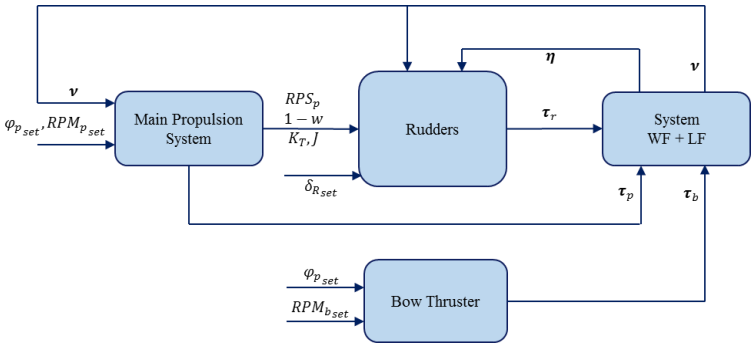


The prerequisite of the forces allocation logic is that the absolute value of the DP rudder angle is computed with respect to the required forces. This implies that the DP rudder angle (for less then its sign) is not an unknown of the thrust allocation algorithm. At this stage, the actuation magnitudes, a priori unknown, that the FAL has to determine are the three thrusts of the thrusters. Then, the forces and moments static balance is uniquely determined in an algebraic way except for some situations that require a special handling as shown in martech.

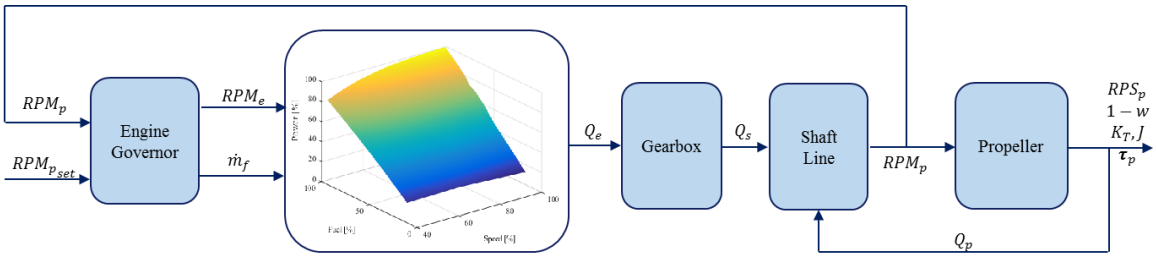
5 SYMPLIFIED PROPULSION MODEL

Although not thoroughly detailed, the simplified propulsion model provides a good approximation of response times measured by the shipyard. By collecting the portside (PT), starboard (SB) and bow thrusts into the array $\boldsymbol{T} \coloneqq \left[\boldsymbol{T}^{(PT)}, \boldsymbol{T}^{(SB)}, \boldsymbol{T}^{(BT)}\right]^{\boldsymbol{T}}$, the actuators model can be summarized in a saturation law for requested thrusts $\boldsymbol{T_R}$, with upper and lower bounds. Then, actuation speeds are taken into account by means of rate limiters. Finally, 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 $\boldsymbol{h} = \boldsymbol{diag}\{\boldsymbol{h}^{PT}, \boldsymbol{h}^{SB}, \boldsymbol{h}^{BT}\}$, where the time constants provided by the shipyard.

6 DETAILED PROPULSION MODEL



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



The engine governor compute the fuel flow requirement with respect to the propeller speed error.Practically, it is a PI controller with the following law:

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

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 coefficient of the regulator. Then, fuel flow is saturated in accordance with eq…..

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

The engine is modeled by the plant map shown in fig…. the output is the engine torque. Required propeller torque is transformed in required shaft torque through the reduction gear. The shaft line dynamic is described by

$$\omega = \int_0^t \frac{Q_p(\xi)\eta_s - Q_s(\xi)}{I_e + I_g + I_s + I_p} \mathrm{d}\xi$$

In the literature, several numerical methods based, for instance, on the potential approach or on R.A.N.S.E. solver, 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 blade position φ:

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$K_Q = \frac{Q}{\rho n^2 D^5}$$

For the same reason of the evaluation of the propeller forces also in this case the rudder forces have been evaluated through a quasi-static-methodology using lift and drag coefficient.

$$C_D$$
$$C_L$$

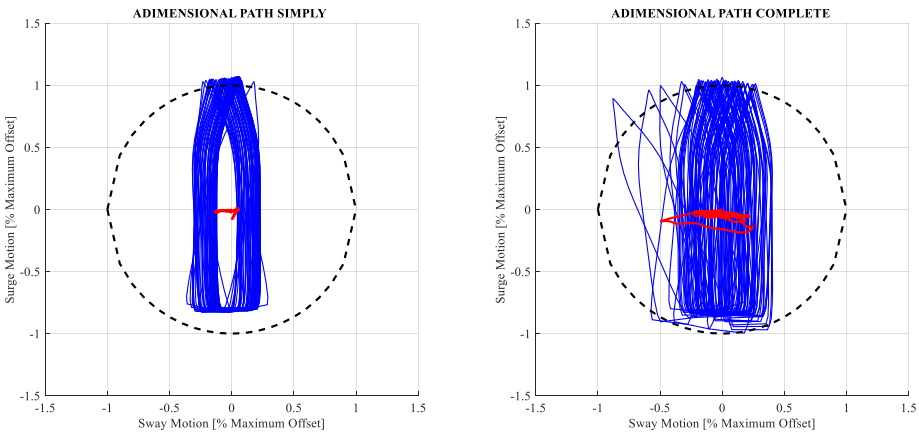
Also the complex interaction between rudder and propeller have been taken into accout using methodology described in….ref.

7 SIMULATIONS AND COMPARISON

In this section the comparisons between the two proposed ship models are shown. In particular, only the most significant simulation output will be compared. In all figures dotted line represent the results of the simplified model instead the continuous ones referes to the detailed model.

Results are shown for both hold position and heading maneuver where all the external forces are aligned and coming from 45° from North, their intensity corresponds to the requirements that correspond to SS4. The vessel is required to keep its position and heading when the wind has a 21 kn speed and the current is 1 kn speed. Initial conditions for the simulation are representative of a real DP maneuver where the system is activated when the vessel has already reached the desired position with the corresponding desired heading. The vessel is supposed to start such maneuver while it is in the origin of the Earth-Fixed Frame with the bow aligned with North direction. Then, its relative angle between the bow and disturbances is still 45°. This is a critical maneuver due to the particular propulsion plant, not conceived for DP application, where lateral force and moment should be independently deliverable. Quartering seas, where longitudinal and lateral forces are comparable and moment reach its highest values, should be, then, difficult to be kept for such vessel.

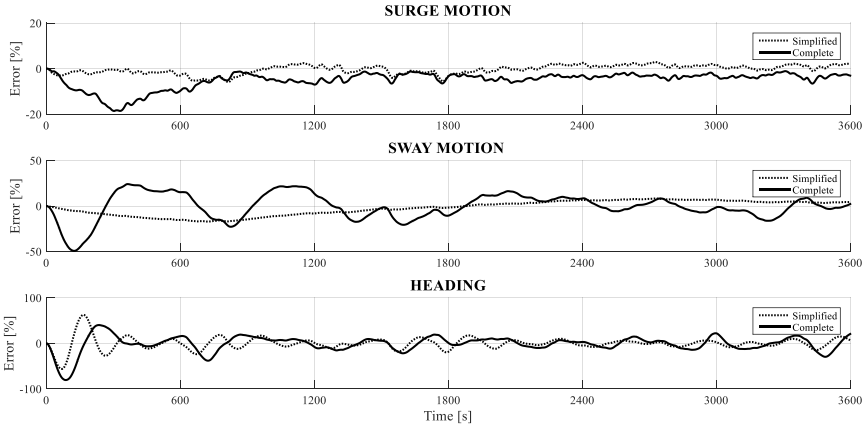
Time history of the path of vessel mid-ship, red line, is shown in Fig…. where on the left hand side results can be found for the simplified model, while on the other side detailed model results are reported. In motion plots, all quantities are represented as percentage of the maximum allowable errors. As it can be seen both the models gives similar results regarding macro-behavior of the vessel during the maneuver, e.g. mean motion 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. Motion can be throughout analyzed in Fig…where the time records of the surge, sway and yaw motion are reported. Here, also yaw amplitude can be seen. Also in the transients, errors never exceed their maximum values, and are almost always comparable by means of errors amplitude. During the transient, only wind

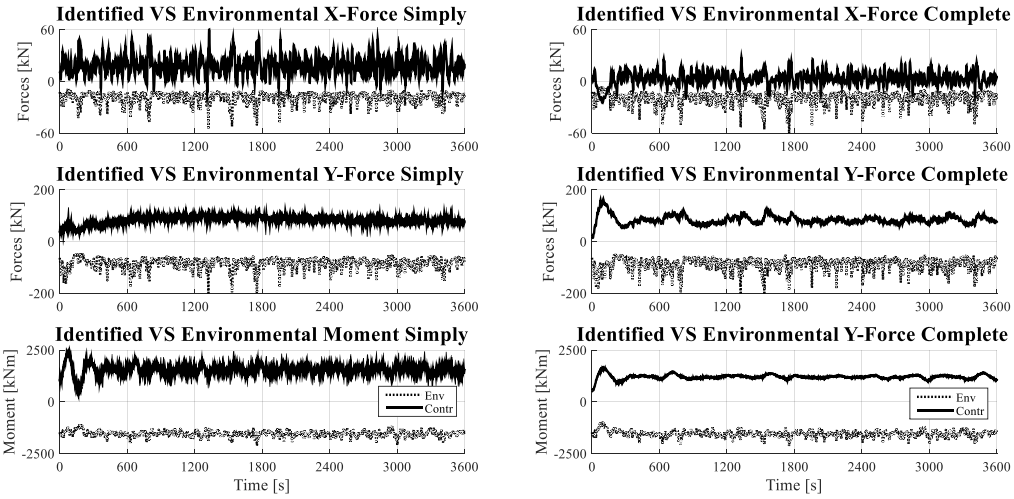


forces and moment are compensated and sea action is identified from the PID. In Fig…the controller output is shown compared with the disturbances forces. For readability purpose, disturbances are the dotted line with their sign; it can be see that in both the models the controller estimates correctly the mean value of the disturbances. An important difference in the models is the computation of the hull force, as shown in eqs…, reported in fig…. Probably, due to the low velocities, not appreciable differences are arguable. On the other hand, the difference in the moment is important because it shows how, when yaw rates are comparably different from zero, within the validity of the zero speed models, the presence of more detailed damping components give their contribution.

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

Finally, the complete model allows to deeply study the behavior of the motor, by means of fuel consumption, efficiency, working points…[mitch se vuoi aggiungere qualche considerazione in merito] i.e. Fig….shows how the engine works…


SURGE MOTION
SWAY MOTION
HEADING
Error [%]
Simplified
Complete
Time [s]


Identified VS Environmental X-Force Simply
Identified VS Environmental X-Force Complete
Identified VS Environmental Y-Force Simply
Identified VS Environmental Y-Force Complete
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Identified VS Environmental Y-Force Complete
Forces [kN]
Moment [kNm]
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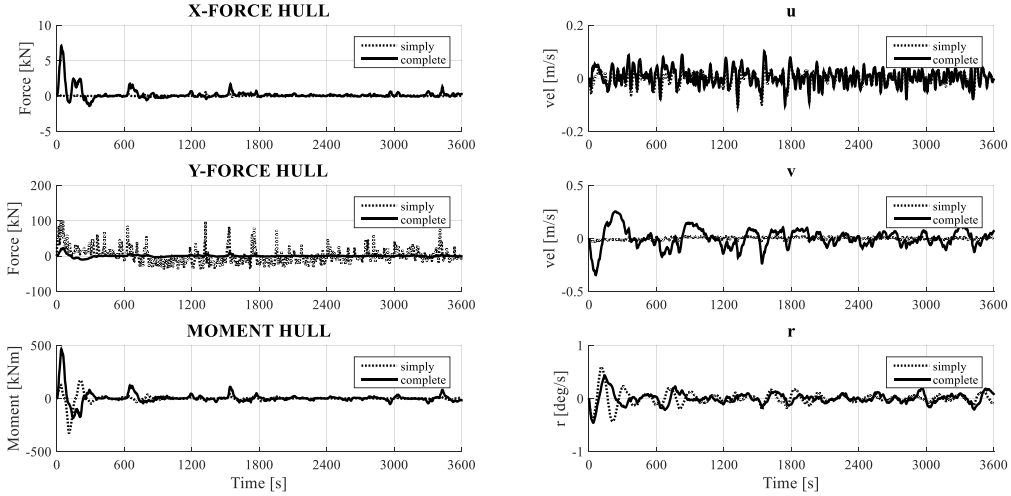
Contr
Time [s]


X-FORCE HULL
Force [kN]
simply
complete
u
vel [m/s]
Y-FORCE HULL
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MOMENT HULL
Moment [kNm]
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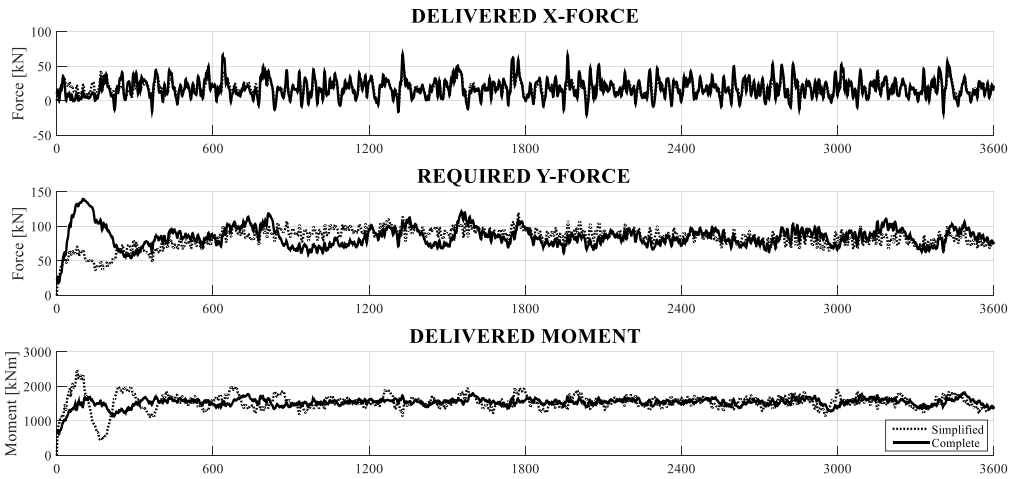
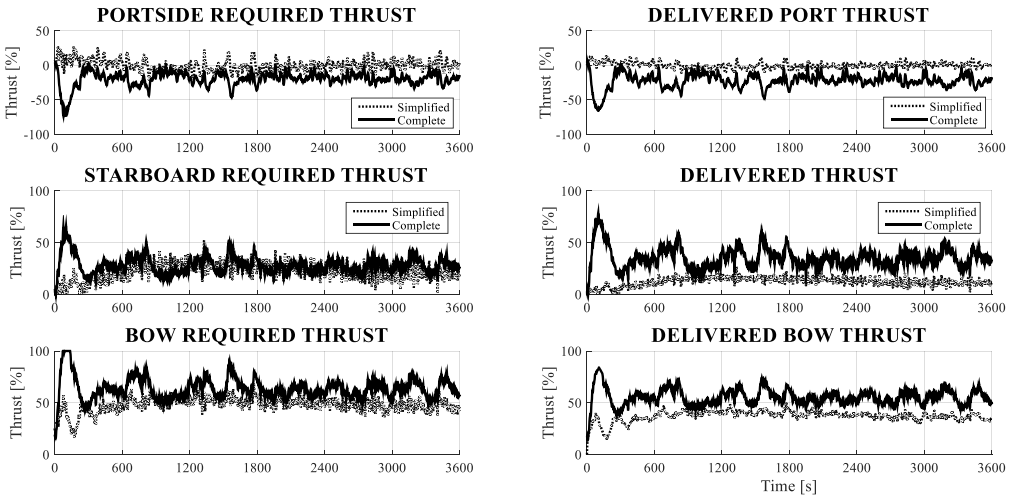
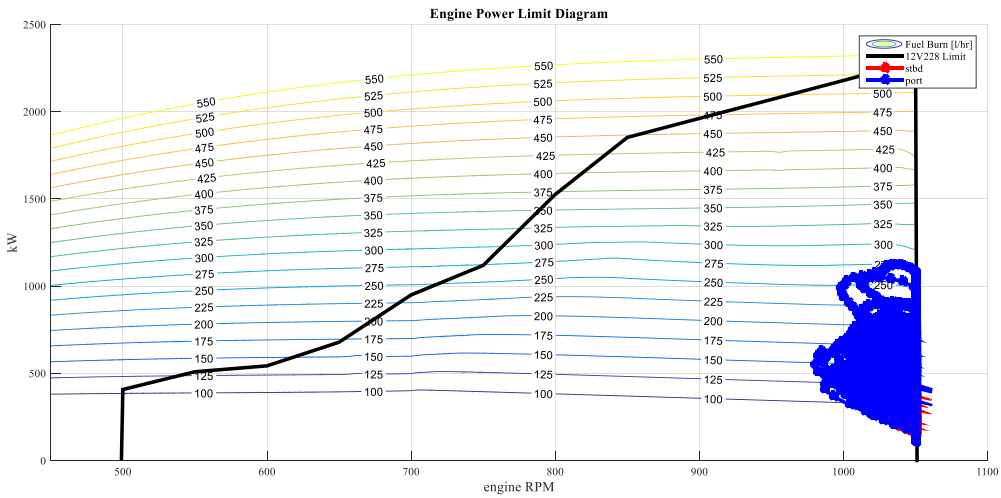

DELIVERED X-FORCE
Force [kN]
REQUIRED Y-FORCE
DELIVERED MOMENT
Moment [kNm]
Simplified
Complete


PORTSIDE REQUIRED THRUST
DELIVERED PORT THRUST
STARBOARD REQUIRED THRUST
DELIVERED THRUST
BOW REQUIRED THRUST
DELIVERED BOW THRUST
Thrust [%]
Time [s]
Simplified
Complete


Engine Power Limit Diagram
Fuel Burn [l/hr]
12V228 Limit
stbd
port
kW
engine RPM

8 CONCLUSIONS

We have presented two different simulation models for testing the robustness of DP regulator, devised to retrofit with a limited DP capability an existing vessel equipped with a conventional propulsion configuration. Benefits from the adoption of a simplified model for testing control logic are summarized in the equivalence in DP performances in both cases, of course also tanks to the accuracy of the saturation values provided by the shipyard. This case of study is particular for the vessel propulsion plant that gives no degrees of freedom for the DP propulsion assessment. Surely, a detailed model allows a more precise reconstruction of the behavior of the whole propulsion line, the possibility of detection of failures and their forecast. DP maneuver is always a stressful condition where a vessel use to work. Moreover, in order to take into account the optimization of fuel consumption and emissions, such a model is a due. Future work will concern comparison with on-board calibration tests on the target ship after installation of the prototype.

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