Digital twin simulation model of hull-propeller-engine interactions for ship condition monitoring in irregular sea navigation

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Synopsis

Faults in the prime mover of a ship would lead to unpleasant consequences during navigation, especially in the event of bad weather. To avoid troublesome situations, any faults in the propulsive chain should be promptly recognized, by observing the initial degradation of sensitive mechanical components through condition monitoring techniques. However, the effects of such degradations need to be distinguished from the offset induced by the wave actions on a healthy system. This paper concerns the analysis of the consequences of several types of degradations on a ship sailing in rough weather. The study is carried out by means of a numerical simulation model that accounts for the hull, propeller and engine interactions in irregular head seas. The degradation levels of some relevant engine components (mechanical parts) are systematically modeled and simulated. The performances of hull dynamics, propeller actions, and engine performances are monitored (monitored variables) and compared with the healthy propulsion system, aiming at providing a correlation between cause and effect. To this end, a twin screw Ro-pax vessel, powered by two marine diesel engines, is considered.

Keywords: Ship dynamics, hull-propeller-engine interactions, marine systems, condition monitoring

1. Introduction

During navigation, it is of primary importance that the main engine of a ship operates healthy and efficiently, in order to avoid power loss and dangerous consequences. The degradation of main engine components should be identified in advance and appropriate replacements of deteriorated parts should be carried out together with an ad-hoc maintenance strategy (Zhao et al. 2022). This would increase the reliability of the prime mover, minimize the troublesome situations in open water navigation and thus increase the ship safety (Luo & Shin 2015).

The fault analysis of a system can: indicate that something is going wrong in the monitored system (fault detection), find the location of the fault (fault isolation) and quantify the magnitude of the fault (identification procedure) (Tleis 2019) (Frank & Seliger 1991). Among the available fault analysis, some of them were applied to marine diesel engines, which is the commonly adopted type of prime mover onboard ships. In (Laskowski 2015; Knežević et al. 2020), the failure of marine engines is studied by fault tree analysis, that uses logical relations for the assessment of possible faults in a system. In (Khelil et al. 2012; Bukovac et al. 2015; Campora et al. 2015; Zaccone et al. 2015), neural network methods are used at the scope of the fault analysis on different types of marine diesel engines. In (Altosole et al. 2022), the fault analysis is carried out by means of parameter estimation approach using a numerical simulation model of the marine engine, where the parameters resemble the magnitude of the faults. In (Orhan & Celik 2023), a literature review of the fault analysis methods for marine engines is carried out, covering a period on 20 years up to the 2022. Only a limited number of papers attempts to address in an explicit way the performances of the main engine operating in presence of rough weather i.e. connecting hull, propeller and engine behaviours in waves (Ghaemi & Zeraatgar 2021). It is immediately observable that a healthy engine operating in waves will have a different working status, resulting in a lower ship speed (depending on the additional wave resistance); moreover, this situation will have effects also on the propeller loading. A good simulator for a marine propulsion chain in rough sea is made up of several numerical sub-models representing the elements involved in the problem that are ship dynamic in waves model, propeller model, engine model. Usually, the dynamics of a ship is usually studied by 6DOF numerical models in the time domain based on the equations of the rigid body motions (Mortola et al. 2012; Matusiak 2021). However, these numerical approaches assume constant propeller revolutions, thus disregarding propeller interaction with the main engine.

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The engine behaviour can be modelled with different levels of accuracy, according to the different purposes and, at the scope, there exist specific programming languages (Altosole et al. 2017; Mrzljak et al. 2017; Altosole et al. 2019) or commercial software (Theotokatos et al. 2016; Mocerino et al. 2021; Ceglie et al. 2023).

In the current paper, we adopt the numerical model for ship dynamics, based on the so-called hybrid or blended non-linear approach (Acanfora & Rizzuto 2019); whereas the engine model refers to a 0D model based on a filling and emptying approach (Benvenuto et al. 2017). The engine modelling choice, despite somewhat simplified compared to commercial software, has the advantage of being implemented in Matlab/Simulink environment, similarly to the numerical model of hull dynamics. Moreover, the sole engine model has been already used for condition monitoring analysis with degraded mechanical components in a previous research (Altosole et al. 2022). Besides, a preliminary coupling of these two numerical models has been presented in (Acanfora et al. 2022), in absence of any degradation of the engine and disregarding added resistance from hull fouling, where two irregular sea states were investigated.

Therefore, in this research, we further develop the numerical model including possible degradations in the propulsion chain. The hull under investigation is a Ro-pax ferry, powered by two four-stroke medium diesel engines. Given a reference irregular sea state, the effects on ship dynamics and on engine variables, when each degradation is set in the propulsion system, are simulated and analysed. Within the same framework, also hull fouling is treated as a source of degradation.

2. Hull, propeller, engine model

The time domain numerical simulation model is implemented in the MATLAB/Simulink environment, and it has a modular arrangement. The sub-models of ship dynamics in irregular seas includes all pertinent non-linearities regarding: the non-linear coupling terms of the rigid body dynamics; the non-linearities of hull geometry in the calculation of Froude–Krylov and restoring actions (subscript "FK" in (1)); the radiation forces and moments, implemented by means of the convolution integral (in (1), the terms a_{ij} and k_{ij}, with i and j from 1 to 6, are, respectively, the added mass coefficients corresponding to the infinite frequency, and elements of the memory function). Differently, diffraction forces and moments (subscript "diff" in (1)) were obtained by linear superposition of regular wave components by potential strip theory calculations. The term X_{res} in eq. 1 is the ship resistance in calm sea, function of the Froude number, while, the added wave resistance accounts only for ship inertia and for Froude–Krylov and restoring actions, that are predominant in case of long waves (i.e. prevailing on diffraction force in the advance direction). Therefore, in order to improve the accuracy of the numerical outcomes, it is recommended working with sea states characterized by wave lengths longer than ship length, thus with large characteristic periods.

The propeller characteristics in wave, i.e. thrust (T_{prop}) and torque (Q_{prop}), are corrected accounting for propeller emersion and propeller loading by means of the technique presented by (Smogeli 2006). The advance velocity at the propeller depends on the variable ship speed in wave (due to surge motions) and by the wake of the hull. For irregular sea problems, the implemented wake factor w, and the thrust deduction factor t_p, equal the values referred to calm sea state. The equation (1) is expressed in the body fixed reference frame centered at the ship center of gravity.

$$\begin{cases}(m+a_{11})\dot{u}+m(qw-rv)+a_{15}\dot{q}=-mgsin\theta+X_{FK}+X_{diff}-k_{11}-k_{15}+X_{prop}+X_{res}\\(m+a_{22})\dot{v}+m(ru-pw)+a_{24}\dot{p}+a_{26}\dot{r}=mgcos\theta sin\phi+Y_{FK}+Y_{diff}-k_{22}-k_{24}-k_{26}\\(m+a_{33})\dot{w}+m(pv-qu)+a_{35}\dot{q}=mgcos\theta cos\phi+Z_{FK}+Z_{diff}-k_{33}-k_{35}\\(I_x+a_{44})\dot{p}+(I_z-I_y)qr+a_{42}\dot{v}+a_{46}\dot{r}=K_{FK}+K_{diff}-k_{44}-k_{42}-k_{46}\\(I_y+a_{55})\dot{q}+(I_x-I_z)rp+a_{51}\dot{u}+a_{53}\dot{w}=M_{FK}+M_{diff}-k_{55}-k_{53}-k_{51}\\(I_z+a_{66})\dot{r}+(I_y-I_x)pq+a_{62}\dot{v}+a_{64}\dot{p}=N_{FK}+N_{diff}-k_{66}-k_{62}-k_{64}\end{cases}\tag{1}$$

$$X_{prop}=(1-t_p)T_{prop}\tag{2}$$

A PI (proportional integral) controller is used for modelling the engine governor. The function of the engine governor is to keep the rotation speed of the diesel engine constant despite variations of propeller load because of wave actions and fouling by allowing the engine to increase or decrease the torque generated.

The governor determines the torque supplied by the engine Q_e acting on the amount of fuel injected at each engine cycle. The engine revolutions N_e are the result of the solution of the dynamic equation of the shaft line

(Taskar et Al. 2016), depending on the rotating inertia of the whole propulsive chain J, where the shaft is assumed to be rigid.

$$\eta_m Q_e - Q_{prop} = J\frac{dN_e}{dt} \tag{3}$$

In (3) the term η_m is a mechanical efficiency of the engine crank system and the propulsion shaft, that, in the current simulations, was assumed equal to 1.

The diesel engine model is based on a 0D filling and emptying approach, aiming at a compromise between obtaining a fairly accurate engine behaviour during transient stages and limiting the computational work of the simulator. All main engine components are arranged in blocks and modelled by algebraic and/or differential equations according to the principles of mass and energy conservation, whereas the fluid modelling is based on the assumption of ideal gas which composition varies through the engine components. The engine blocks are: cylinder, compressor, turbine, intercooler, turbocharger shaft (shaft TG). The 0D engine model consists in simplified modeling of the thermodynamic behavior of a single cylinder, and considers all cylinders working in phase.

At the scope of the application, a four-stroke turbocharger engine model is implemented, where the turbine and compressor are modelled using steady state maps. Engine and turbocharger speeds are calculated by dynamic equilibrium equations. A classical double zone Wiebe function is implemented for modelling the heat release during the combustion in each cylinder. The inlet valves timing can be varied to control the air flow at different engine speed and loads. Fresh air temperature, and fresh cooling water temperature are kept constant during the simulation.

For the sake of synthesis, the detailed description of the ship dynamics model and engine model are available in (Benvenuto et al. 2017; Acanfora & Rizzuto 2019), respectively.

3. Modelling of hull and engine degradations

The hull-propeller-engine model, implemented in the Simulink toolbox, allows for the possibility of introducing a degradation level in pertinent mechanical components, and in the hull surface smoothness, using degradation coefficients. All coefficients have unitary value when the engine works in a healthy state and there is no fouling on the hull. Each alteration in the coefficient values can reduce the efficiency or the working feature of a specific engine element (in the case of the hull fouling, increase the drag), with consequences on the whole propulsion chain. Among the numerous mechanical engine parts susceptible of failure, we selected the most representative, based on the experience of the co-authors and referring to the technical literature (Benvenuto & Campora 2007; Ceglie et al. 2023).

Table 1 reports the list of the degradations under investigation, i.e. the degradation coefficients, implemented in the pertinent engine simulation block; whereas the hull degradation coefficient is applied to the evaluation of Xres in the ship dynamics model. Once a degradation level is set in an engine part or in the hull, it is possible simulating within the numerical model the deviations between hull propeller engine interactions compared to the simulation outcomes referring to the health state of the engine.

Among the numerous state variables charactering the hull, propeller, engine behaviour a selection is made. At the scope, we focused our attention only on meaningful variables that can be reasonably monitored on board (Ceglie et al. 2023) (Altosole et al. 2022). These are listed in Table 2 and Table 3.

Table 1: List of hull and engine degradation coefficients

Simulation block	**Degradation (Coefficient name)**
Intercooler	Intercooler fouling (d_in_p)
Intercooler	Efficiency reduction (d_in_eff)
Compressor	Dirty air filter (d_co_eff)
Compressor	Efficiency reduction (d_co_re)
Compressor	Mass flow reduction (d_co_m)
Shaft TG	Bearing deterioration (d_tg_cu)
Turbine	Efficiency reduction (d_t_re)
Turbine	Fouling of the blades (d_t_pa)
Cylinder	Fuel flow reduction (d_c_co)
Ship dynamics	Hull fouling (d_h_f)

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Table 2: List of monitored hull and propeller state variables

Hull	
Ship velocity	V
Heave	ζ
Pitch	θ
Propeller	
Torque	Q
Revolutions	N

Table 3: List of monitored engine state variables

Pressure (outlet)	
Compressor	p_c
Cylinder (outlet)	p_{cl}
Temperature (outlet)	
Compressor	t_c
Turbine	t_t
Cylinder	t_{cl}
Other	
Specific fuel consumption	sfc
Engine torque	Q_e
Engine Speed	N_e

4. Case study

The ship under investigation is a Ro-pax ferry named Seatech-D equipped with two diesel engines of 12MW each and a twin-screw propeller configuration. The hull has been used for previous researches concerning the applications and validations of the numerical model for ship dynamics in wave (Acanfora & Rizzuto 2019).

At the scope of the current study, the maximum speed is set as 24 knots. Two Wageningen B-series propellers are assumed, for which KT and KQ coefficients are available from (Barnitsas et al. 1981). The calibration and the validation of the engine model was carried out in (Altosole et al. 2022).

Table 2: Principal particulars of Seatech-D

Hull Seatech-D	
Length between perpendiculars, L (m)	158.00
Breadth, B (m)	25.00
Depth, D (m)	15.00
Draft forward, T_F (m)	6.10
Draft aft, T_A (m)	6.10
Displacement, Δ (tons)	13,766
Center of gravity above keel, KG (m)	11.834
Long. coordinate of the center of gravity from aft perpendicular, LCG (m)	74.77
Transv. radius of gyration in air, k_{XX} (m)	10.06
Long. radius of gyration in air, k_{YY} (m)	39.36
Propeller Wageningen B-series	
Number of blades Z	4
A_e/A_0	0.750
D_{prop} (m)	4.8
$P/_{Drop}$	1.2
Engine	
Number of cylinders	12
Bore (m)	0.51
Stroke (m)	0.60

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Engine revolution N_{eng} (rpm)	514
Engine power P_B (MW)	12

Table 4 summarizes the main hull, engine and propeller features.

The applications are carried out for the same irregular sea realization (see Figure 1) obtained by the technique described in (Acanfora & Rizzuto 2019). The chosen sea state is described by the JONSWAP spectrum and it is characterized by a significant wave height Hs=5.5 m and a zero crossing period Tz=11 s. The ship sails in head sea. Prior to introduce any degradation in the numerical model, the state variables are simulated for the still water condition and for the irregular sea condition, given the unitary values of all degradation coefficients (i.e. engine healthy state) and a fixed propeller revolution Neng. Then, all degradations listed in Table 1, are induced, assuming a constant degradation level of 10 %, corresponding to a degradation coefficient value of 0.9, except for the hull fouling coefficient. In this case, the coefficient is meant to increase the drag of 5% and thus it is set equal to 1.05.

5. Results: no degradation

This section includes the still water and wave results for the engine working without any degradation. Each simulation lasts 10 minutes. Figure 1 shows a comparison between the hull state variables in absence and in presence of waves, whereas Figure 2 provides the same comparison but focusing on engine state variables.

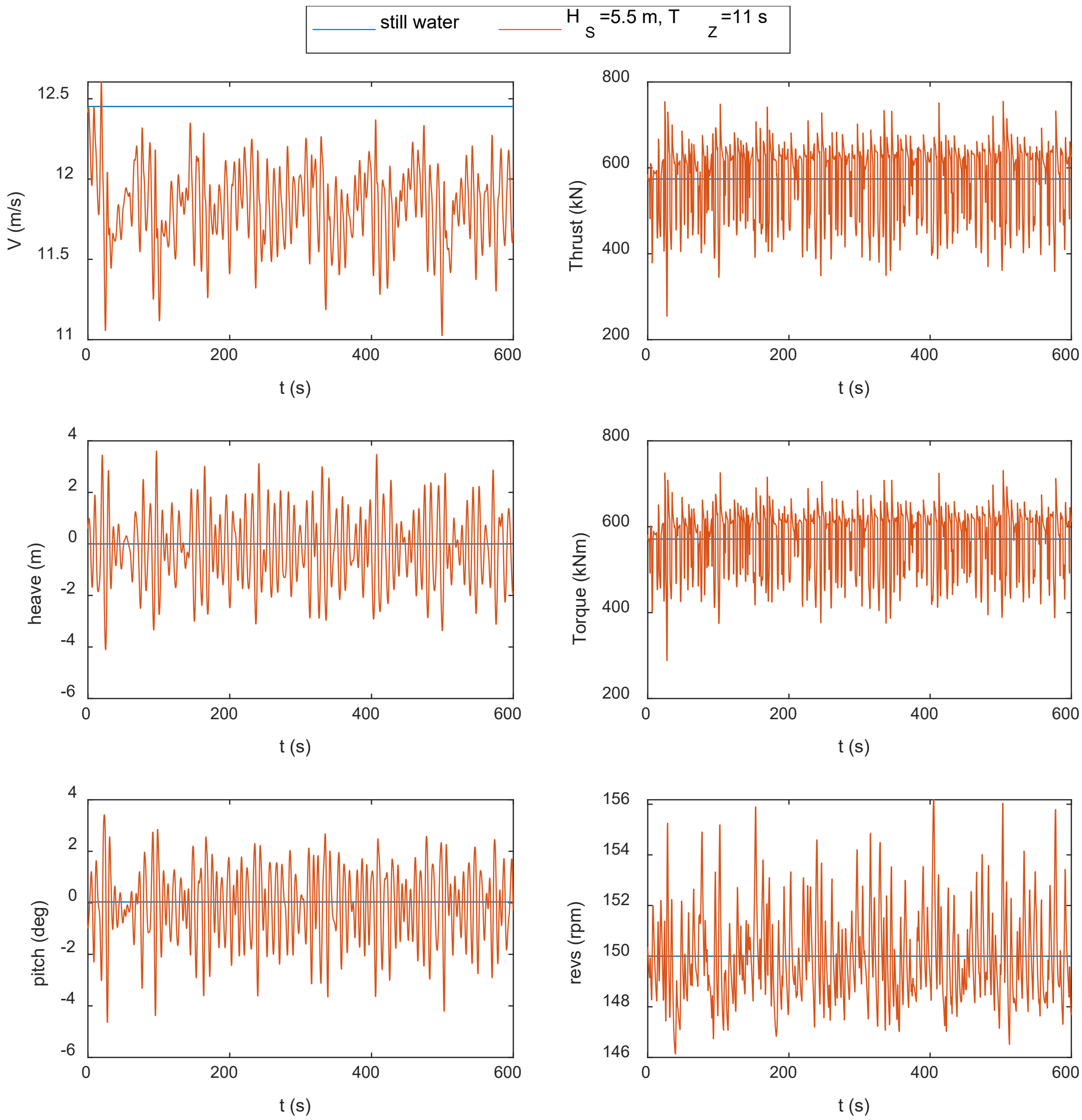

Figure 1: Hull and propeller behaviours in still water and in waves, no degradation

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The steady behavior of the engine variable in still water condition presented in Figure 2, could be attributed to the 0D engine model assumptions.

Due to the added wave resistance, a speed reduction of almost 1.2 knots is observed in waves, compared to the still water case. However, the averaged propeller behaviour remains almost equal in the two cases, in terms of torque, thrust and revolutions (there is an increase of only 0.25% in thrust and torque), although in waves they show an irregular oscillatory trend. Although the propeller thrust is not a monitored variable, it is presented in Figure 1 for a more complete description of the simulated propeller behaviour. Regarding the monitored engine variables, it is possible observing that from the numerical simulation there are no significant oscillatory pressure variations in waves, that almost overlaps still water outcomes. Instead, for the remaining variables i.e. temperatures, engine torque, revolutions, and specific fuel consumption, the results exhibit an appreciable irregular oscillatory behavior in waves, but with average values almost equal to the corresponding still water outcomes.

This preliminary analysis suggests that for the chosen operational condition, there is no substantial change for irregular sea navigation in the mean values of the monitored engine and propeller variables. The only appreciable change is observed in the ship speed.

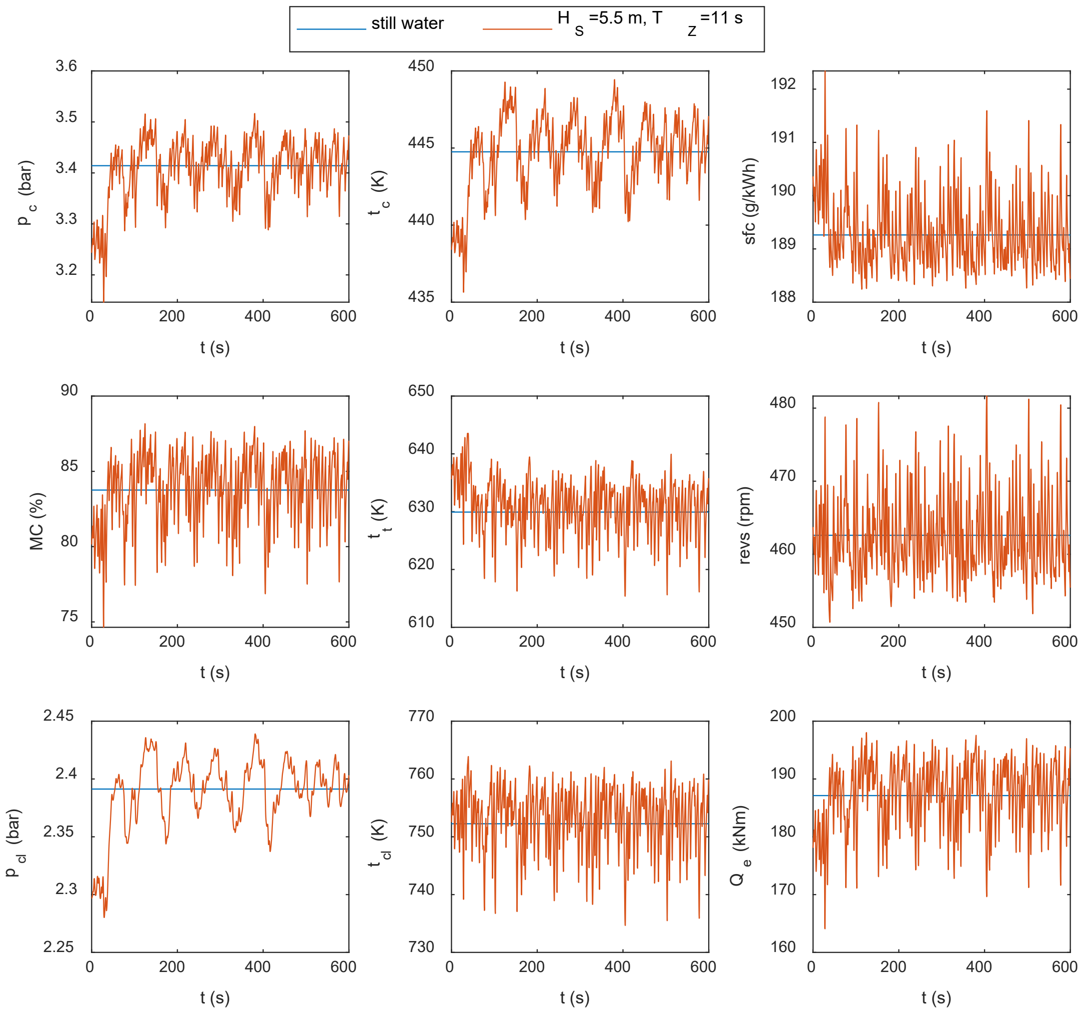

Figure 2: Engine behavior in still water and in waves, no degradation

Due to the absence of data from the field, validation of numerical simulation regarding the engine behavior in transient conditions was not possible. However, the obtained results appear reasonable, especially because they depend on the 0D engine model (i.e. neglecting the effective dynamics of the combustion).

Nevertheless, we can deduce that the ship sailing the chosen irregular sea experiences a greater fuel consumption than in calm sea navigation, mainly because of the speed reduction (increase in the navigation time), whereas the specific fuel consumption of the engine remains somewhat unaltered.

It is worth underlying that although the coupling between the presented hull dynamics model and engine model was not validated, the standalone models were properly calibrated and validated on the hull (Acanfora & Rizzuto 2019) and on the engine (Altosole et al. 2022) used at the scope of the case study.

6. Results: hull and engine degradations

This analysis aims at comparing the values of the monitored ship and engine variables between the nominal condition (i.e. no degradation) and the cases with an induced degradation. At the scope, ten numerical simulations are carried out, each of them corresponding to a specific altered degradation coefficient, in the starboard engine. Given the twin-screw arrangement of the ship, it was possible inducing degradation to the starboard engine and maintain the port engine operating at the nominal condition as a reference.

For each degradation case, the monitored variables of the degraded starboard engine (altered values) and the monitored variables of the non-degraded port engine (nominal values), were simulated and collected. Following, we analysed the offset between degraded vs non-degraded engine variables by calculating the root mean square of the error (rmse) between the monitored degraded vs non-degraded engine variables. The obtained results are arranged in Table 5, in order to investigate on the variations induced by the several degradation types. In Table 5, for each degradation scenario, the rmse between nominal and altered variables are shown as percentage of the nominal condition. As an example, setting in the simulation a fouling level of 10% in the intercooler (d_in_p) of the starboard engine, this degradation type causes an average variation in the cylinder temperature t_{cl} of 6% compared to the non-degraded engine (i,e, with no fouling at the intercooler).

In Table 5, it is possible observing how the degradation level of 10% in the engine components has a modest influence on ship speed compared to the hull fouling effects, for the sea state under investigation. The outlet pressure at the compressor has been resulted the most sensitive parameter, especially with degradation types belonging to the turbocharger system. Indeed, there are similar trends in the monitored variables given the mechanical element interested by the degradations, such as those referring to the intercooler (d_in_p e d_in_eff) or to the compressor (d_co_eff, d_co_re, d_co_m). The fouling on turbine blade degradation (d_t_pa) exhibits the largest observed deviation, in the outlet pressure at the cylinder, associated to a modest variation of the specific fuel consumption. Actually, this variable varies of almost 1% in all degradations associated to turbocharger and intercooler (that could be reasonable considering the modest degradation levels under investigation) and it remains unaltered for hull fouling degradation. Indeed, this is consistent with the fact that the engine is working with no degradation at its nominal condition, thus in case of hull fouling only a speed reduction of 1% is observed.

The specific fuel consumption increases of 11.13% for the degradation case involving the fuel flow reduction (d_c_co) i.e. associated to a valve leakage, that leaves somewhat unaltered the remaining engine monitored variables. The values in Table 5 cannot directly provide the state of the engine, especially dealing with small degradation levels and small alterations of the variables. However, this analysis suggests that each degradation scenario could have a peculiar set of sensitive variables to be used as an indicator of the healthy state of the system, using optimization techniques for condition monitoring (Altosole et al. 2022), (Ceglie et al. 2023).

Table 3: Offset of the engine monitored variables, with degradation vs no degradation

	t_{cl}	p_{cl}	t_c	p_c	t_t	sfc	N	Q_e	Q	V
d_in_p	6.02%	5.17%	0.11%	1.23%	7.33%	1.08%	0.03%	0.10%	0.10%	0.01%
d_in_eff	2.13%	0.69%	0.47%	2.02%	2.32%	0.32%	0.01%	0.05%	0.05%	0.00%
d_co_eff	6.78%	5.77%	0.17%	9.97%	8.26%	1.22%	0.03%	0.11%	0.11%	0.01%
d_co_re	6.66%	5.66%	0.28%	9.80%	8.10%	1.20%	0.03%	0.10%	0.10%	0.01%
d_co_m	6.49%	6.98%	3.67%	12.84%	7.93%	1.15%	0.03%	0.10%	0.10%	0.01%
d_tg_cu	6.30%	5.56%	3.32%	10.05%	7.71%	1.13%	0.03%	0.10%	0.10%	0.01%
d_t_re	6.36%	5.58%	3.33%	10.11%	9.68%	1.15%	0.03%	0.10%	0.10%	0.01%
d_t_pa	4.60%	19.55%	4.15%	13.15%	8.07%	0.51%	0.01%	0.06%	0.06%	0.01%
d_c_co	0.18%	1.03%	0.41%	0.99%	0.27%	11.13%	0.26%	0.87%	0.87%	0.11%
d_h_f	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	1.03%

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7. Conclusions

This work reported the development of a numerical model for the hull-propeller-engine interactions in rough sea conditions.

A brief description of ship dynamics in 6DOF was provided in the paper. The effects of manoeuvring on the propeller-engine matching were neglected. The case study accounted only for head sea navigation. The engine 0D model was implemented, including the possibility of simulating partial ineffectiveness of mechanical components by using degradation coefficients. Additionally, the hull degradation coefficient was introduced, regarding the effects of hull fouling on ship resistance.

A Ro-pax ferry, characterized by a twin-screw arrangement was chosen for the case study. Among all variables available within the numerical simulation, only a reduced number of engine and hull variables were presented in this study. Such variables were judged pertinent for monitoring the engine and hull state. A sample wave train realization was assumed and kept for all numerical simulations. It was observed that the sea state had influence on hull behaviour and that the average engine state depended only on the average engine power (no degradation case). Additionally, we observed that the mean value of specific engine consumption did not depend on hull dynamics in waves nor on hull fouling. Obviously, the total fuel consumption within a planned navigation would increase because of the increase in sailing time due to speed reduction.

The analysis of the monitored variables, induced by each degradation, disclosed that the most sensitive item for assessing turbocharger degradation was the outlet pressure at the compressor.

Based on the outcomes of the present study, although restricted to a limited case study, we could foresee that diagnostic techniques performed on the ship engine operating in rough sea navigation could still be successful by working on the mean values of the variables. Future studies are expected to address this topic.

In particular, in future studies, after a validation of the presented model (that will serve as a digital twin of the engine operating on a sailing ship), we will aim at employing techniques for condition monitoring of the engine state, by comparing the measured variables onboard with the simulated ones.

It is worth underlying that, although the presented outcomes refer only to the hull and engine under investigation, the proposed approach can be extended to different engine types and to different ships, given the availability of pertinent engine models and hull data.

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