A Modelling Approach for the Overall Ship Propulsion Plant Simulation

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Abstract: - In the present paper, a modelling approach for the simulation of the overall ship propulsion plant is presented. A cycle mean value model with differential equations for the calculation of the engine crankshaft and turbocharger shaft speeds is used for the modelling of the main engine of the vessel. The ship shafting system is modelled using the power balance and its efficiency. In order to calculate the propeller thrust and torque, the polynomials for the propellers of the Wageningen B type are used. In addition, the ship velocity and the movement along its longitudinal axis are also calculated using the differential equation describing the ship surge dynamics. The mathematical equations of the ship propulsion plant model are implemented and solved using the computational environment MATLAB Simulink®. The model was used for the simulation of the propulsion plant of a merchant ship under various operating conditions and the derived results are presented.

Key-Words: - ship propulsion plant, marine Diesel engine, simulation, MATLAB Simulink®

1 Introduction

Nowadays, simulation of the various components of the ship propulsion plant plays important role on the better understanding of the physical processes occurring, as well as the interaction between the involved subsystems. In addition, simulation can be used to facilitate the development and optimization procedure of the ship powerplant components by initially testing alternative design options, or to evaluate propulsion plant subsystems control schemes.

The simulation tools used throughout the design procedure of the propulsion plant subsystems can be of varying complexity. Cycle mean value models are used for fast transient powerplant performance estimation, evaluation of the interaction between the involved components as well as for the engine control system development [1,2]. Zero-dimensional and one-dimensional models are used for more detailed modelling of the thermodynamic and flow dynamic processes inside the engine components [3-5]. Three-dimensional (CFD or finite elements) models [6] are used for the development, investigation or optimization of powerplant components design where the evolved processes take place in three dimensions (e.g. engine combustion chamber design, propeller design, etc.).

The objective of this work is to present a comprehensive but easily handled tool for the simulation of the overall ship propulsion plant, so that the initial tests of various design options, the interaction between the various components and control scenarios for the main engine as well as propeller pitch control can be investigated. In that respect, the mathematical modelling of the overall ship propulsion system implemented in the computational environment MATLAB Simulink® is accomplished. The main engine of the ship is modelled using a cycle mean value model approach in conjunction with differential equations for the calculation of the engine crankshaft speed and the turbocharger shaft speed. The thermodynamic and flow dynamic processes in the engine components are taken into consideration in order for the various engine parameters to be calculated. The angular momentum conservation and the energy balance in the propulsion plant shafting system in combination with the shafting system efficiency is also taken into account for calculating the engine shaft speed. The polynomials for the Wageningen B propeller type are used for the calculation of the propeller toque and thrust. The differential equation derived using the ship surge dynamics is used for calculating the ship longitudinal movement and velocity. The developed tool was applied for the simulation of the propulsion plant of a merchant ship and the derived ship and propulsion plant operating parameters are presented.

2 Propulsion Plant Description

The typical propulsion plant installation of a modern merchant vessel, shown in Fig.1, consists of the main marine Diesel engine, the shafting system and the propeller.

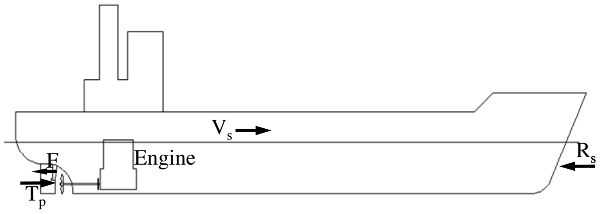

Fig. 1 Ship propulsion schematic

The main engine can be of the two-stroke type for installations of high power or of the four-stroke type for installations of lower power [7,8]. In both cases, the engine is turbocharged by one or more turbocharger units so that the power produced by the engine is maximized. The engine turbocharging system consists of the turbocharger unit in combination with the engine intake and exhaust manifolds [9]. The turbocharger unit consists of a centrifugal compressor and a centripetal or an axial flow turbine attached on a common shaft. The engine exhaust gas is expanded in the turbine, thus producing work, which in turn is utilized to the compressor for compressing the air. In consequence, the pressure, and as a result, the density of the air contained in the engine intake manifold is increased. Thus, the amount of air entering the engine cylinders is greater, allowing more fuel to be burnt; as a consequence, the engine produces more power without increasing the engine size.

The shafting system comprises the connecting shafts and the bearings. For the cases where the main engine is four-stroke, one gear box is connected between the engine crankshaft and the propeller shaft. In high power installations, a shaft generator is often installed in order to produce the required by the ship electric power during ship voyages, where the engine is operating at relatively high load.

In the case of merchant vessels propulsion plant installation, the propeller is usually of fixed pitch type, but at the last years designs with controllable pitch propellers have also been used [10].

2.1 Engine modelling

The engine of the ship is modelled by considering the processes occurring in its parts. The engine parts that have been taken into consideration, shown in Fig.2, are the engine cylinders, which are connected between the engine inlet receiver and the engine exhaust receiver, the engine turbocharger, which consists of the compressor and the turbine connected in common shaft, and the engine air cooler, which is located after the compressor in order to cool the hot air exiting the compressor.

For two stroke marine Diesel engines, the air mass flow rate through the engine cylinders is calculated considering that the cylinders are equivalent to a system comprising two orifices connected in series [11]. Each one of the orifices corresponds to the intake ports and the exhaust valve, respectively. The above described layout can be simplified by combining the two orifices in one equivalent orifice with pressure difference equal to the cylinder pressure difference and flow area calculated using the following equation [11]:

$$A_{eq} = \frac{z_c}{2\pi}\int_0^{2\pi} \frac{A_i(\phi)A_e(\phi)}{\sqrt{A_i^2(\phi)A_e^2(\phi)}}\mathrm{d}\phi \qquad (1)$$

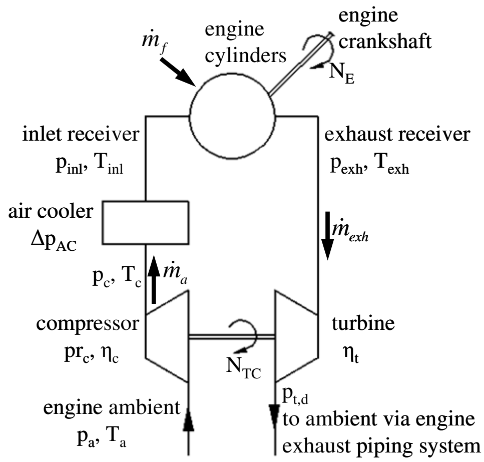

Fig. 2: Engine components and operating parameters

Thus, the air mass flow rate is calculated using the following equation, which has been derived according to the quasi-one dimensional consideration in an orifice with subsonic flow [12]:

$$\dot{m}_a = c_d A_{eq} \frac{p_{inl}}{\sqrt{R_a T_{inl}}} f\left(p_{exh} / p_{inl}, \gamma_\alpha\right) \qquad (2)$$

where $f\left(\frac{p_{exh}}{p_{inl}}, \gamma_\alpha\right) = \sqrt{\frac{2\gamma_\alpha}{\gamma_\alpha - 1}\left[\left(\frac{p_{exh}}{p_{inl}}\right)^{\frac{2}{\gamma_\alpha}} - \left(\frac{p_{exh}}{p_{inl}}\right)^{\frac{\gamma_\alpha+1}{\gamma_\alpha}}\right]}$

In the case of four-stroke engines, the following amount, which accounts for the mass flow rate due to the air pumping of the engine, must be added to the respective one derived by applying equation (2):

$$\dot{m}_{pump} = \eta_{vol}\rho_{inl}V_D N_E / 120 \qquad (3)$$

The energy balance applied on engine cylinders gives:

$$(\dot{m}_a c_{p,a} T_{inl} + \eta_{comb}\dot{m}_f H_L \zeta)\eta_{exh} = (\dot{m}_a + \dot{m}_f) c_{p,e} T_{exh} \qquad (4)$$

By applying the mass balance in the engine, as well as by using the quasi-one dimensional consideration [9,12] for calculating the exhaust gas mass flow rate through the turbine, the following equations are derived:

$$\dot{m}_{exh} = \dot{m}_a + \dot{m}_f = \begin{cases} A_{t,eff} \dfrac{p_{exh}}{\sqrt{R_e T_{exh}}} f\left(p_{t,d} / p_{exh}, \gamma_e\right) \\ \qquad \text{for subsonic flow} \\ A_{t,eff} \dfrac{p_{exh}}{\sqrt{R_e T_{exh}}} \sqrt{\dfrac{2\gamma_e}{(\gamma_e - 1)} \left(\dfrac{2}{\gamma_e + 1}\right)^{\frac{\gamma_e+1}{\gamma_e-1}}} \\ \qquad \text{for sonic flow} \end{cases} \quad (5)$$

The turbine effective flow area is derived from the turbine geometric area and the turbine flow coefficient, which, in turn, is considered function of the turbine pressure ratio, i.e.:

$$A_{t,eff} = \alpha_t A_{t,geo} \;\; with \;\; \alpha_t = k_{t0} + k_{t1} pr_t + k_{t2} pr_t^2 \quad (6)$$

Equations (2), (4) and (5) forms a non-linear algebraic system of three equations with three independent variables, the air mass flow rate, $\dot{m}_a$, the exhaust receiver pressure, p_{exh}, and the exhaust receiver temperature, T_{exh}. The rest of the parameters involved in the above equations are calculated as follows.

The air and gas properties, the fuel lower heating value, the discharge coefficient and the correction factor for the exhaust gas temperature are considered to be constant.

The pressure and temperature of the air contained in the inlet receiver are calculated by modelling the compressor and the air cooler. The compressor is usually modelled using its performance map. However, in marine applications, where it can be assumed that the engine is loaded according to the propeller law, the compressor operating points lay on a single curve [9,13], which can be represented by the following equation:

$$pr_c = k_c N_{TC}^2 + 1 \quad (7)$$

In that case, the compressor efficiency can also be assumed constant, or for greater accuracy, its variation with turbocharger shaft speed can be taken into account. Thus, the compressor exiting air temperature is calculated using the following equation, which has been derived using the compressor efficiency definition [9]:

$$T_c = T_a \left(1 + \left(pr_c^{(\gamma_a - 1)/\gamma_a} - 1\right) / \eta_c\right) \quad (8)$$

The pressure drop in the air cooler is calculated by:

$$\Delta p_{AC} = f_{AC} \frac{\rho}{2} V^2 = \frac{f_{AC} R_a T_c \dot{m}_a^2}{2 p_c A_{AC}^2} \quad (9)$$

The temperature of air exiting air cooler, which is equal to the inlet receiver temperature, is calculated using the air cooler effectiveness and the temperature of the air cooler coolant medium:

$$T_{inl} = T_c - \varepsilon\left(T_c - T_w\right) \quad (10)$$

The air cooler effectiveness is assumed to be a function of the air mass flow rate:

$$\varepsilon = k_{AC0} + k_{AC1} \dot{m}_a + k_{AC2} \dot{m}_a^2 \quad (11)$$

The pressure of the air contained in the inlet receiver is calculated by:

$$p_{inl} = p_c - \Delta p_{AC} = pr_c p_a - \Delta p_{AC} \quad (12)$$

The proportion of the chemical energy of the fuel contained in the exhaust gas is considered linear function of the engine mean effective pressure [11]:

$$\zeta = k_{z0} + k_{z1} \overline{p}_b \quad (13)$$

The engine brake mean effective pressure is calculated by subtracting the friction mean effective pressure from the indicated mean effective pressure. The indicated mean effective pressure is calculated using the rack position, the maximum indicated mean effective pressure of the engine and the combustion efficiency, which in turn is regarded as function of air to fuel ratio [9,12]:

$$\overline{p}_i = x_r \overline{p}_{i,\max} \eta_{comb} \quad (14)$$

The friction mean effective pressure is considered function of the indicated mean effective pressure and the engine crankshaft speed:

$$\overline{p}_f = k_{f0} + k_{f1} N_E + k_{f2} \overline{p}_i \quad (15)$$

For the calculation of the fuel mass flow rate, the variation of the mass of injected fuel per cylinder and per cycle vs. fuel rack position must be provided as input. Thus, for each value of rack position, the fuel mass flow rate is calculated by:

$$\dot{m}_f = z_c m_{f,cy} N_E / \left(60\, rev_{cy}\right) \quad (16)$$

In order to calculate the engine shaft and the turbocharger shaft speeds, required for the calculation procedure eq. (3), (7), (15), (16), the following equations are used, which are derived by applying the angular momentum conservation in the propulsion plant shafting system and the turbocharger shaft, respectively:

$$\frac{dN_E}{dt} = \frac{\eta_{sh} \eta_{GB} i_{GB} Q_{eng} - Q_p}{I_E + I_{GB} + I_{sh} + I_p} \quad (17)$$

$$\frac{dN_{TC}}{dt} = \frac{Q_t - Q_c}{I_{TC}} \quad (18)$$

In the case where a gear box is not installed in the propulsion plant, then the respective terms in eq. (17) become: $I_{GB}=0$, $\eta_{GB}=1$, $i_{GB}=1$.

The engine torque is derived using the engine brake mean effective pressure, the engine displacement volume as given below:

$$Q_E = \overline{p}_b V_D / \left(2\pi\, rev_{cy}\right) \quad (19)$$

The engine brake power, brake specific fuel consumption and efficiency are calculated by the following equations, respectively:

$$P_b = Q_E \pi N_E / 30,\ bsfc = \dot{m}_f / P_b,\ \eta_b = P_b / \left(\dot{m}_f H_L\right) \quad (20)$$

The compressor impeller absorbed torque is calculated by:

$$Q_c = P_c / \omega_{TC} = \dot{m}_a c_{pa} \left(T_c - T_a\right) / \left(\pi N_{TC} / 30\right) \quad (21)$$

The turbine wheel delivered torque is given by:

$$Q_t = \dot{m}_{exh} c_{pe} \left(T_{exh} - T_{t,d}\right) / \left(\pi N_{TC} / 30\right) \quad (22)$$

Using the turbine efficiency definition equation [9] and after some manipulation, the following equation is derived for the calculation of the temperature of the exhaust gas exiting turbine:

$$T_{t,d} = T_{exh} \left(1 - \eta_t \left[1 - \left(p_{t,d} / p_{exh}\right)^{(\gamma_e - 1)/\gamma_e}\right]\right) \quad (23)$$

The turbine efficiency is taken as function of the turbine pressure ratio, or alternatively of the turbine velocity ratio [9].

The pressure after the turbine is calculated using the pressure increase of the exhaust piping system:

$$p_{t,d} = p_a + \Delta p_{ep} = p_a + \frac{f_{ep} \dot{m}_{exh}^2}{2 \rho_{ep} A_{ep}^2} \quad (24)$$

2.2 Engine governor modelling

A PID (proportional-integral-differential) engine governor model was chosen and implemented in MATLAB Simulink® environment. According to that, the following equation is used for the calculation of the engine rack position:

$$x_r = x_{r,o} + k_p \Delta N + k_i \int_0^t \Delta N dt + k_d \frac{d\left(\Delta N\right)}{dt} \quad (25)$$

where $\Delta N = N_{ord} - N_E$ is the difference between the ordered engine speed and the actual engine speed. The engine governor model is depicted in Fig.3. As it is shown in that figure, an engine torque limiter has also been incorporated in the engine governor model. Its function is to limit the engine governor rack position to a predetermined maximum value depending on the engine speed, in order to protect the engine integrity during fast transients.

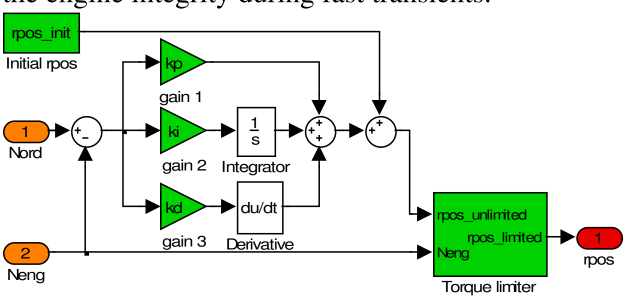

Fig. 3: Engine governor model

2.3 Propeller modelling

For the ship propeller, either of fixed or controllable pitch type, its torque and thrust are calculated using the non-dimensional coefficients, as follows:

$$Q_p = k_Q \rho_{sw} N_p^2 D_p^5 \quad (26)$$

$$T_p = k_T \rho_{sw} N_p^2 D_p^4 \quad (27)$$

The non-dimensional coefficients are calculated from polynomial equations for Wageningen B propeller series [14]:

$$k_Q = \sum CQ_{s,t,u,v} J^s \left(p / D_p\right)^t \left(A_E / A_o\right)^u z_p^v \quad (28)$$

$$k_T = \sum CT_{s,t,u,v} J^s \left(p / D_p\right)^t \left(A_E / A_o\right)^u z_p^v \quad (29)$$

In the above relations, the number of propeller blades, z_p, the disk area coefficient, A_E/A_o, the pitch to diameter ratio, p/D_p, and the propeller advance coefficient, J, are required as input. Provided that the propeller of the ship has been selected, the first two parameters are constant. In addition, for a fixed pitch type propeller, the pitch to diameter ratio takes also constant value. For the case of a controllable pitch propeller, the p/D_p values can vary according to a predetermined schedule or by the propeller pitch control system. The propeller advance number, J, depends on the speed of advance (velocity of the water arriving in the propeller), V_A, the propeller rotational speed and the propeller diameter, i.e.:

$$J = V_A / \left(N_p D_p / 60\right) \quad (30)$$

The speed of advance is calculated using the ship velocity and the ship wake fraction, w, which is considered constant taking values in the range from 0.20 to 0.45 for ships with a single propeller [14,15]:

$$V_A = \left(1 - w\right) V_s \quad (31)$$

The propeller open water efficiency is defined by the following relation:

$$\eta_p = \frac{k_T J}{k_Q 2\pi} \quad (32)$$

In order to take into account the inertia of the entrained water by the propeller, the entrained water coefficient is used. The total propeller inertia is calculated by:

$$I_p = \left(1 + \eta_{ew}\right) I_{p,a} \quad (33)$$

The entrained water coefficient variation with the propeller blades angle of attack is presumed [16]. The propeller blades angle of attack is derived as the difference between the blades geometric pitch angle and the angle of relative velocity at propeller blades leading edge (advance angle):

$$\alpha = \arctan\left(\left(p / D_p\right) / \pi\right) - \arctan\left(V_A / V_u\right) \quad (34)$$

The circumferential blade velocity is given by:

$$V_u = 0.7 D_p \left(\pi N_p / 60\right) \quad (35)$$

The propeller real slip ratio [14], which is a parameter used in order to indicate the propeller loading under various operating conditions, is calculated by:

$$SR_R = 1 - V_A / \left(p\, N_p / 60 \right) \qquad (36)$$

2.4 Ship longitudinal movement modelling

By applying the ship surge dynamics, the following differential equation is derived for the calculation of the longitudinal ship velocity:

$$\left(m_s + m_{hydro} \right) \frac{dV_s}{dt} = T_p - F - R_s \qquad (37)$$

The ship resistance, R_s, is considered a second order function of the ship velocity, V_s [8]. The mass, m_s, is the mass of the ship, which is calculated by multiplying the ship displacement with the sea water density. The term, m_{hydro}, is an added virtual mass, which is used in order to take into account the hydrodynamic force arising due to the acceleration of a body in a fluid [14].

The thrust deduction, F, is calculated using the thrust deduction coefficient, t, as given below:

$$F = t\, T_p \qquad (38)$$

The thrust deduction coefficient can be taken as constant with typical values in the range from 0.12 to 0.30 for ships with a single propeller [15], or alternatively it can be presumed as a function of ship velocity.

3 Propulsion Plant Model Implementation

The mathematical modelling of the ship propulsion plant components, which was presented above in this text, was implemented in the MATLAB Simulink® environment, as shown in Fig.4. The propulsion plant model consists of the following types of blocks designated with different colours: a) the blocks where the inputs of the submodels must be inserted, b) the blocks where the initial values of the engine crankshaft speed, the turbocharger shaft speed and the ship velocity have to be set, c) the blocks where the integration of the model differential equations of the model is performed, d) the blocks where the calculation of the parameters of the ship propulsion plant components is carried out, e) the ordered engine speed schedule block, f) the engine governor block and g) the output block of the model.

The required input data for the propulsion plant modelling are categorised in the following groups: the geometric data of the components, the properties of working media (air, exhaust gas, sea water) and the constants of the submodels.

The calculation procedure takes place as follows. At the start of the simulation time, the values for the three independent variables (air mass flow rate, exhaust gas pressure and temperature) are estimated. For each time step, taking into account the values for the engine speed, turbocharger shaft speed and the ship speed (their initial values are taken into consideration at the start of the simulation time), the required parameters of the submodels are calculated and the no-linear algebraic system of the three equations (2), (4) and (5) are solved using the "fsolve" function of the Optimization Toolbox of MATLAB [17]. Having the air mass flow rate, the exhaust gas pressure and temperature calculated, the remaining engine parameters as well as the time derivatives of the engine speed and the turbocharger shaft speed are also derived. The last two parameters and the time derivative of the ship velocity are fed to the integration blocks, where the engine speed, the turbocharger shaft speed and the ship velocity, are calculated, respectively, using a fourth order Runge-Kutta integration method with fixed time step. The above described procedure is repeated for every time step till the end of the simulation time.

For each time step, a set of parameters of the ship propulsion plant is stored in a variable available to the workspace of MATLAB, so that the plots of the parameters variation can be easily constructed.

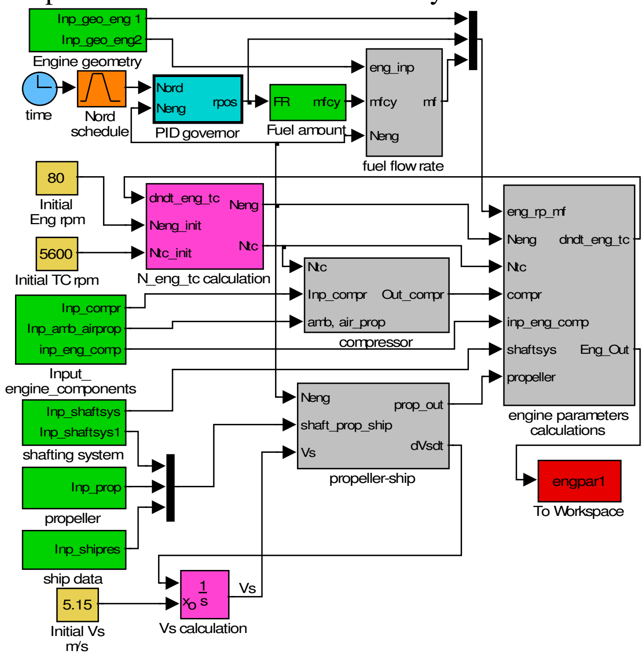

Fig. 4: Overall ship propulsion plant model implemented in MATLAB Simulink®

4 Numerical Example

The propulsion plant of a typical merchant ship was simulated using the modelling approach described above. The ship taken into consideration was of the bulk carrier type having deadweight 55000 t. The main ship particulars are shown in

Table 1. The ship propulsion plant consists of a MAN B&W 6L60 engine [18]. The engine is of the two-stroke marine Diesel type, turbocharged by one turbocharged unit working on constant pressure turbocharging system and is directly connected to the ship propeller via the shafting system (i.e. gearbox is not installed). The engine and propeller main characteristics are given in Tables 2 and 3, respectively.

Table 1: Ship Parameters

Size (at scantling draught)	55000	dwt
Length overall	190	m
Length between perpendiculars	183	m
Breadth	32.26	m
Draught (scantling)	12.7	m
Draught (design)	11.5	m
Mass	$6.358\ 10^6$	kg

Table 2: Engine Parameters

Bore	600	mm
Stroke	1944	mm
Number of cylinders	6	
Brake Power (L1)	11520	kW
Engine speed (L1)	123	rpm
bmep (L1)	17	bar
bsfc (L1)	171	gr/kWh
Turbocharger units	1	

Table 3: Propeller Parameters

Diameter	6.5	m
Number of blades	5	
Pitch to diameter ratio	0.665	
Area ratio	0.57	

Initially, the engine model was set up and the model constants were calibrated so that the simulation results for various values of the engine governor rack position are in good agreement with the respective ones given by the engine manufacturer in [18]. Then, the calibrated engine model in conjunction to the rest of the propulsion system geometric data was used to set up the overall propulsion plant model.

A set of results including operating parameters of engine, propeller and ship, which was derived from the simulation of a period of 600 s (10 min) of the above described ship propulsion plant operation, are shown in Fig.5. During the first 60 s of the simulation run, the engine is operating at 80 rpm, which corresponds to a rack position value of 0.42 and ship velocity of 10 knots. It must be noted the rack position value of 1.0 corresponds to the maximum continuous rating of the engine (point L1 in [18]). After the 60^{th} s, there is a linear increase in the ordered engine speed from 80 rpm to 105 rpm at the 180^{th} s. This causes the action of engine

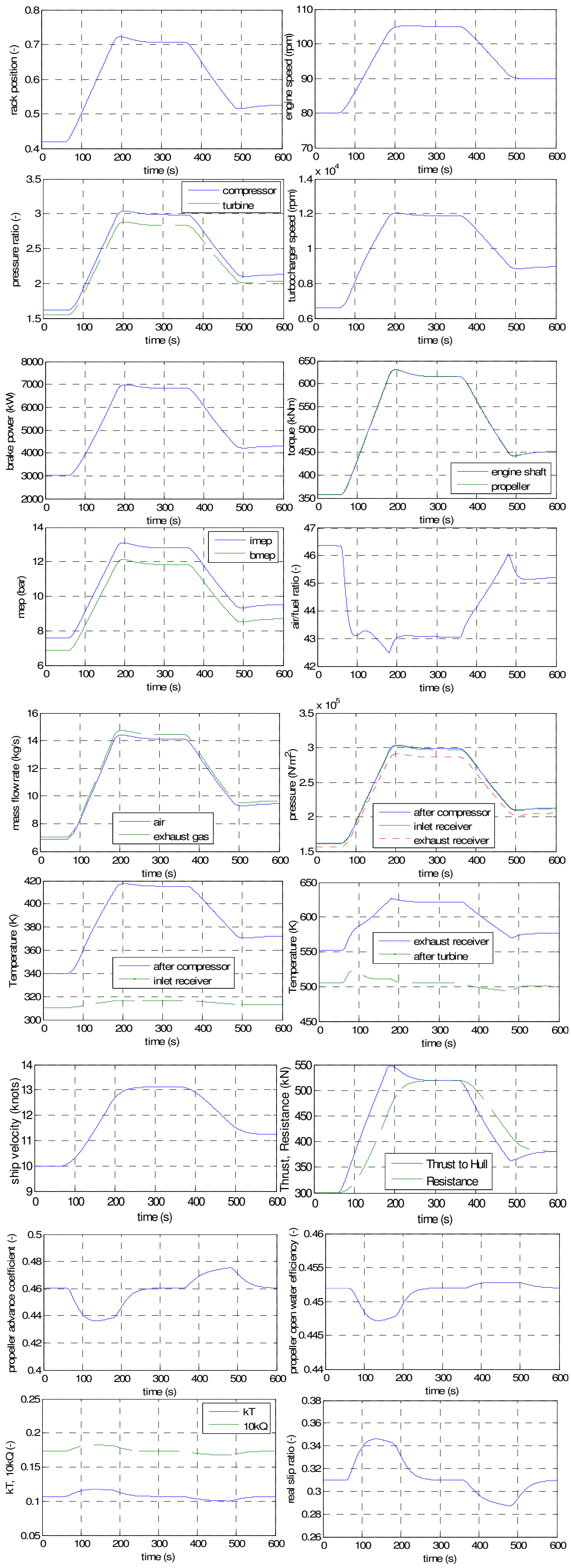

Fig. 5: Simulation results including engine, propeller and ship operating parameters

governor, which increases the rack position resulting in more fuel to be injected and burnt into the engine cylinders, thus producing more engine torque. That, in turn, increases the engine shaft speed faster than the ship velocity rises, and as a consequence, the propeller produced thrust exceeds the ship resistance, thus accelerating the ship from 10 knots at 60th s to 13.1 knots at 280 s. As it is clearly seen from Fig.5, during the ship acceleration period, the engine air to fuel ratio reduces, resulting in the increase of exhaust receiver temperature, which is more pronounced at the initial part of the ship acceleration period. That is owing to the slower response of the turbocharger system and the delivery of inadequate amount of air to the engine cylinders. However, after the 120th s, where the turbocharger speed increases, more air mass flow is available and the slope of the exhaust receiver gas temperature exhibits an obvious reduction. During the ship acceleration period, the propeller advance coefficient decreases due to the fact that the propeller speed increases faster than the ship velocity does, resulting in the reduction of the engine propeller open water efficiency. In addition, the propeller is more heavily loaded, which is also indicated by the observed increase of the non-dimensional coefficients and the real slip ratio.

The engine ordered speed is held constant at 105 rpm till the 360th s. Then, a linear reduction in the engine ordered speed is commanded from 105 rpm to 90 rpm at the 480th s. After the action of engine governor, the engine speed is reduced and the ship gradually decelerates to 11.2 knots. The engine air to fuel ratio increases as the fuel is reduced and at the same time due to the turbocharging system slower response, adequate amount of air is still available. However, after the 520th s, the turbocharger speed and the compressor pressure ratio have been reduced enough so that the engine air mass flow rate also decreases, resulting in the reduction of the air to fuel ratio. The decrease of the mass of fuel injected into the engine cylinders leads to the engine speed reduction since for that period of the engine operation, the propeller absorbs more torque than the torque delivered by the engine. In addition, due to the faster deceleration of the propeller speed than the respective one of the ship, the ship resistance exceeds the thrust delivered to the ship hull, thus causing the reduction of the ship velocity. During the ship deceleration period, the propeller advance coefficient increases resulting in a slight rise of the propeller open water efficiency. The propeller operates under lighter loading, as can be also deduced by the fall of real slip ratio and the non-dimensional coefficients.

5 Conclusion

The mathematical modelling of the overall ship propulsion plant installation and the ship longitudinal movement, implemented in the computational environment MATLAB Simulink®, was presented. Typical operating cases of a bulk carrier were simulated and a set of results derived from the simulation, containing selected engine propeller and ship operating parameters were presented.

The presented model is capable for investigating the steady state and transient performance of the overall propulsion plant installation, initial testing of various design options of the ship propulsion plant subsystems (e.g. engine, propeller etc) as well as for the development and validation of selected control scenarios. In addition, due to its modular implementation, its components can be easily modified and extended, augmenting the effectiveness of its usage.

Finally, it can also be used for educational purposes, in order for the students to understand the interaction between the ship and its propulsion plant subsystems.

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Nomenclature

A :	area
a_t :	turbine flow coefficient
bsfc:	brake specific fuel consumption
c_p :	specific heat at constant pressure
D :	diameter
F :	thrust deduction
f :	friction factor
I :	polar moment of inertia
i :	gear ratio
J :	propeller advance coefficient
H_L:	fuel lower heating value
h :	specific enthalpy
k :	non-dimensional coefficients, constants
L :	length
m :	mass
$\dot{m}$:	mass flow rate
N :	rotational speed
P :	power
p :	pressure, pitch
pr :	pressure ratio
$\overline{p}$:	mean effective pressure
Q :	torque
R :	gas constant, resistance
rev_{cy}:	revolutions per cycle
SR:	slip ratio
T :	temperature, thrust
t :	time, thrust deduction coefficient
V :	velocity
V_D:	displacement volume
x_r :	rack position
z_c :	number of engine cylinders
γ :	ratio of specific heats
Δp:	pressure drop
ε :	effectiveness
η :	efficiency
η_{exh}:	correction factor for the temperature of the exhaust receiver
η_{vol}:	volumetric efficiency
ρ :	density
ζ :	proportion of the chemical energy of the fuel contained in the exhaust gas
ϕ :	crank angle
ω :	angular velocity

Subscripts

a :	ambient
AC:	air cooler
b :	brake
c :	compressor
comb:	combustion
cy :	cycle
d :	downstream
E :	engine
eff :	effective
ep :	exhaust pipe
eq :	equivalent
ew :	entrained water
exh:	exhaust receiver
f :	fuel, friction
GB:	gear box
i :	indicated
inl :	inlet receiver
p :	propeller
s :	ship
sh :	shaft
sw :	sea water
TC:	turbocharger
t :	turbine
u :	circumferential
w :	coolant water