as a function of ship velocity.

III. MODEL IMPLEMENTATION IN MATLAB/SIMULINK ENVIRONMENT

The mathematical modeling of the ship propulsion plant components, which was presented above in this text, was implemented in the MATLAB/Simulink environment, as shown in Fig. 3. The propulsion plant model consists of the following types of blocks: 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 modeling are categorized 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 receiver 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 non-linear algebraic system of the three variables are solved using the “fsolve” function of the Optimization Toolbox of MATLAB [33]. Having the air mass flow rate, the exhaust receiver gas pressure and temperature calculated, the remaining engine parameters as well as the time derivatives of the engine crankshaft 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 crankshaft 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.

For the case of simulating only the two-stroke marine Diesel engine, a modified version of the model shown in Fig. 3 was also implemented, in which eq. (40) is used for the calculation of propeller torque.

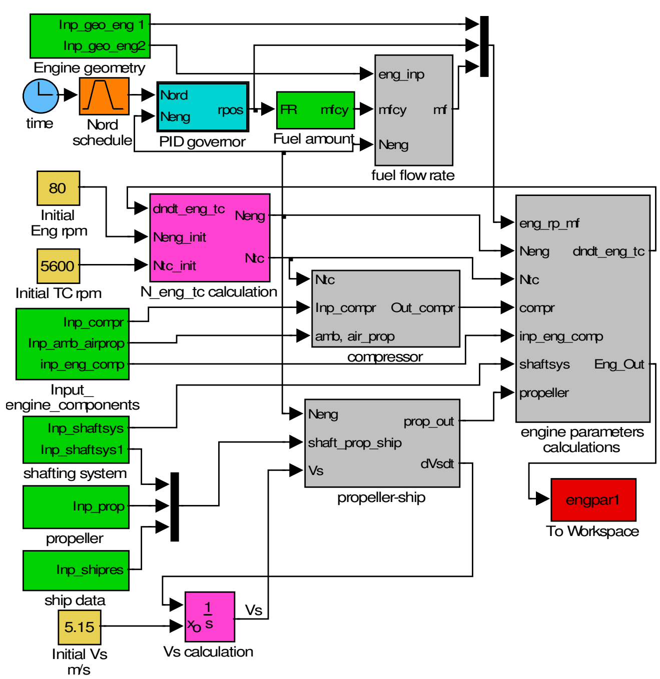

Fig. 3 Ship propulsion plant model implemented in MATLAB/Simulink

IV. RESULTS AND DISCUSSION

In order to examine the ability of the above described mean value engine model for predicting the engine operating parameters with sufficient accuracy, the MAN B&W 4L60MC engine was simulated. That engine is of the two-stroke marine Diesel type, turbocharged by one turbocharged unit working on constant pressure turbocharging system. For that type of engine, experimental data under steady state and transient operating conditions were previously published in [13]. Part of those data is also used in Fig. 4 and 5 presented below, indicated as “reference”. The main engine characteristics as well as the required model input data were extracted from the engine manufacturer project guide [34], whereas the required compressor and turbine input data were taken from [13]. The engine main parameters are given in Table I.

First, the model was set up providing the required input data. Then, simulation runs of the engine steady state operating conditions at 50%, 75%, 90% and 100% of the engine brake power were performed. Slight adjustments were performed in a number of the model variables, e.g. the engine cylinders discharge coefficient or turbine equivalent area, so that the model predictions are in acceptable agreement with the reference experimental data. The model predictions for a set of engine operating parameters including the brake mean effective pressure, the exhaust receiver gas temperature, the turbocharger shaft rotational speed and the scavenging receiver pressure are presented in Fig. 4. In the same figure, the respective measured data taken from [13] are also shown. As it can be inferred by comparing the data presented in Fig. 4,

TABLE I: ENGINE PARAMETERS

Engine type	4L60MC	
Number of cylinders	4	
Bore	600	mm
Stroke	1944	mm
Brake Power (MCR)	6720	kW
Engine speed (MCR)	111	rpm
bmep (MCR)	16.5	bar
Turbocharger units	1 MAN B&W NA57/TO	

the model predictions under engine steady state conditions exhibit adequate accuracy.

The next step was to examine the model ability to capture the engine transient response. In that respect, the Run 11 presented in [13] was simulated. According to that, engine abrupt load changes from 97% to 67% at the 5th s and back to 97% at the 38th s were demanded. Those correspond to changes in engine speed from about 110 rpm to 95 rpm. Such load changes are considered very fast and are usually avoided in real engine operation, but engine transient response under similar conditions are used for the control system design.

The transient run was initially performed by using the constants derived from the steady state runs, and the actual value of the turbocharger rotating parts polar moment of inertia. The analysis of the derived simulation results, which are not presented in this text, showed good agreement between simulation and measurements for engine speed and power variations, whereas the predictions for turbocharger speed, scavenging receiver pressure and exhaust gas receiver temperature varied more rapidly than the respective measured data. That was attributed to the unmodeled dynamics of engine scavenging air and exhaust gas receivers and the turbocharging system piping volumes, as it is also reported in [10]. The transient run was repeated using for the turbocharger rotating part polar moment of inertia its actual value increased by 100%, so that the dynamics of the engine receivers and turbocharger system volume are taken into account. A set of the derived simulation results, including the engine speed and brake power, the turbocharger speed, the scavenging receiver pressure and the exhaust receiver gas temperature, as well as the respective measured data (taken from [13]) are presented in Fig. 5. As it is deduced by comparing the simulation results to the respective measured data, the presented engine operating parameters (except for exhaust receiver gas temperature) are predicted with very satisfactory accuracy. Although the exhaust receiver gas temperature is adequately predicted under steady state conditions as it is shown in Fig. 4 and Fig. 5 (time periods: from the 1st s the 5th s, from the 25th s to the 38th s and from the 60th s to the 80th s), the model is not capable to capture the exhaust gas receiver response during engine fast transients (Fig. 5 time periods: from the 5th s to the 20th s and from the 38th s to the 60th s). To adequately predict the exhaust gas temperature variation during fast transients, a more detailed model, which should include the engine exhaust gas receiver modeling by means of the open thermodynamic system concept, is required, as it is explained in [10]. However, in cases of slow engine transients, which are usual for the ship propulsion plant operation, the engine operating parameters are predicted with enough accuracy using the modeling approach presented above in this text [10].

Having validated the engine model, the propulsion plant of a typical merchant ship was simulated. The ship taken into consideration was of the bulk carrier type having deadweight 55000 t. The ship propulsion plant consists of a MAN B&W 6L60MC engine (the same engine type as presented in Table I, with two additional cylinders), which is directly connected to the ship propeller via the shafting system (i.e. gearbox is not installed). The main ship and propeller parameters are given in Table II.

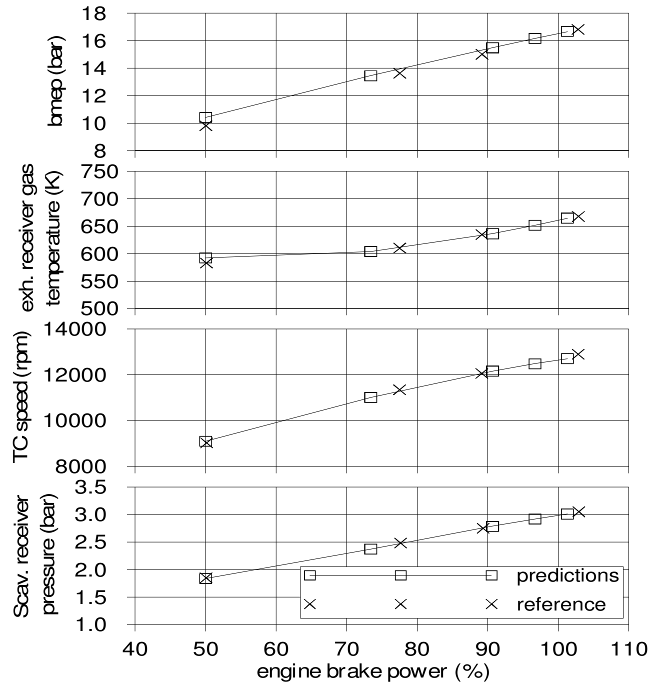

Fig. 4 Simulation results under steady state engine operating conditions and comparison to reference data taken from [13]

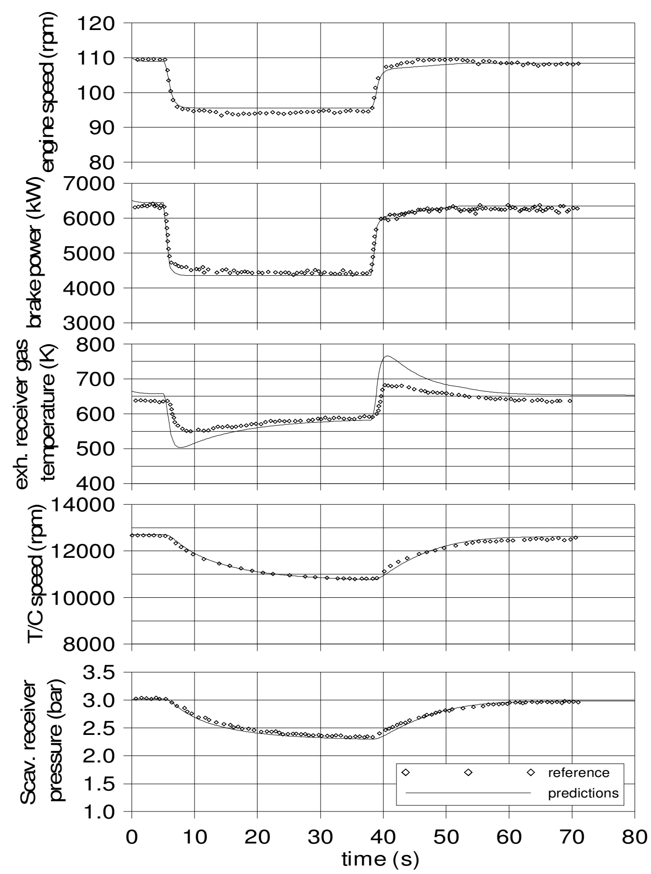

Fig. 5 Simulation under engine transient operating conditions and comparison to reference data taken from [13]

TABLE II: SHIP & PROPELLER PARAMETERS

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

Slight modifications in the engine model input data, which was already set up for the MAN B&W 4L60MC, were performed. 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 the operation of the above described ship propulsion plant during a time period of 600 s (10 min), are shown in Fig.6. 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. 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 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 60^{th} s to 13.1 knots at 280 s. As it is clearly seen from Fig. 6, 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 120^{th} 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 propeller non-dimensional torque and thrust coefficients and the real slip ratio.

The engine ordered speed is held constant at 105 rpm till the 360^{th} s. Then, a linear reduction in the engine ordered speed is

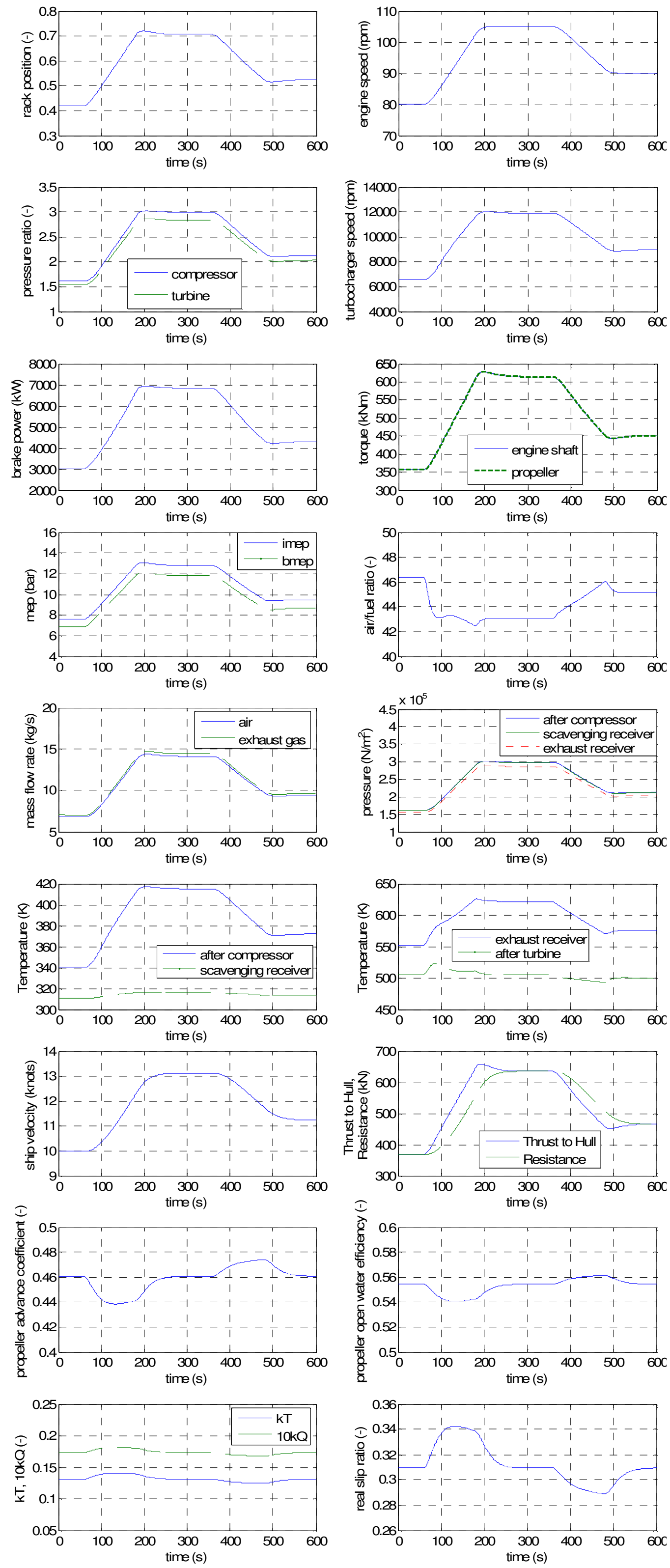

Fig. 6 Simulation results including engine, ship and propeller operating parameters

commanded from 105 rpm to 90 rpm at the 480^{th} 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 500^{th} 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 greater amount of torque than the one 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 torque and thrust coefficients.

As it is clearly deduced for the above described analysis of the transient simulation results, the overall propulsion plant modeling can be used in order to better understand the dynamic behavior of the propulsion plant as well as the engine-propeller-ship interaction.

V. Conclusion

The dynamic behavior of a bulk carrier propulsion plant was investigated using a model implemented in the computational environment MATLAB/Simulink. The main engine of the vessel propulsion plant, which is considered of the two-stroke marine Diesel type, is modeled using a quasi steady cycle mean value approach. First, the validation of the two stroke marine Diesel engine model under steady state and transient operating conditions was performed. Then, typical operating cases of the bulk carrier were simulated and the results containing engine, propeller and ship operating parameters were discussed.

The main findings derived from this work are summarized as follows.

The quasi steady mean value engine model is relatively simple compared to the more detailed zero or one-dimensional codes, requires small amount of input data and takes less computing time for execution. It adequately predicts the engine speed response. In order to represent with acceptable accuracy the turbocharger shaft speed and the scavenging receiver pressure responses during fast transients, the value of the turbocharger rotating parts polar moment of inertia, provided to the model as input, must be increased by an additional amount for taking into account the effect of the volume inertia of the engine scavenging and exhaust receivers and the turbocharging system piping. In such a case, additional data (experimental data or simulation results from more detailed models) are required in order to adjust the value of turbocharger inertia. Although the model can predict the exhaust receiver gas temperature under steady state conditions and during slow transients with adequate accuracy, it exhibits inefficiency to predict the exhaust receiver gas temperature variation during fast transients. In such cases, more detailed modeling of exhaust receiver, e.g. using open thermodynamic system approach, should be required.

The model can predict the dynamic behavior of the components of the ship propulsion plant and can be used to produce data that otherwise obtained using costly procedures. In that way, by analyzing the derived by the simulation ship-propeller-engine operating parameters response, the better understanding of the complex interactions between the subsystems can be obtained.

Nomenclature

A	area (m^2)
bsfc	brake specific fuel consumption (gr/kWh)
CQ	polynomial coefficients for k_Q calculation (-)
CT	polynomial coefficients for k_T calculation (-)
c_d	discharge coefficient (-)
c_p	specific heat at constant pressure (J/kgK)
c_v	specific heat at constant volume (J/kgK)
D	diameter (m)
F	thrust deduction (N)
f	friction factor (-)
I	polar moment of inertia (kgm^2)
J	propeller advance coefficient (-)
H_L	fuel lower heating value (J/kg)
h	specific enthalpy (J/kg)
k	coefficients, constants
k_p, k_i	engine governor proportional and integral constants
k_Q, k_T	propeller non-dimensional torque and thrust coefficients
m	mass (kg)
$m_{f,cy}$	mass of injected fuel per cylinder and per cycle (kg)
$\dot{m}$	mass flow rate (kg/s)
N	rotational speed (rpm)
P	power (W)
p	pressure (N/m^2)
pr	pressure ratio (-)
$\bar{p}$	mean effective pressure (bar)
Q	torque (Nm)
R	gas constant (J/kgK), resistance (N)
rev_{cy}	revolutions per cycle (-)
SR_R	propeller real slip ratio (-)
T	temperature (K), thrust (N)
t	time (s), thrust deduction coefficient (-)
u	specific internal energy (J/kg)
v	velocity (m/s)
V_A	propeller speed of advance (m/s)
V_D	displacement volume (m^3)
V_S	ship velocity (m/s)
V_u	circumferential propeller velocity (m/s)
w	ship wake fraction (-)
x_r	rack position (-)
z_{cyl}	number of engine cylinders (-)
z_P	number of propeller blades (-)

Greeks

α	propeller advance angle (rad)
α_T	turbine flow coefficient (-)
γ	ratio of specific heats (-)
Δp	pressure drop (N/m^2)
ε	effectiveness (-)
ζ	proportion of the chemical energy of the fuel contained in the exhaust gas (-)
η	efficiency (-)
η_{ex}	correction factor for the temperature of the exhaust receiver (-)

ρ	density (kg/m^3)
ϕ	crank angle (deg)

Subscripts

AC	air cooler
atm	ambient
b	brake
C	compressor
comb	combustion
cy	cycle
cyl	cylinder
d	downstream
E	engine
ER	exhaust receiver
e	exhaust gas, exhaust valve
eff	effective
ep	exhaust pipe
eq	equivalent
ew	entrained water
f	fuel, friction
geo	geometric
hydro	hydrodynamic
i	indicated, intake ports
MCR	maximum continuous rating
max	maximum
o	initial conditions
P	propeller
S	ship
SC	scavenging receiver
Sh	shaft
sw	sea water
TC	turbocharger
T	turbine
w	cooling water
α	air

Abbreviations

MCR	maximum continuous rating

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