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# Development of a combined mean value–zero dimensional model and application for a large marine four-stroke Diesel engine simulation

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## HIGHLIGHTS

- Development of a combined mean value–zero dimensional engine model.
- Application for simulating a large marine Diesel four-stroke engine.
- Results comparable to the respective ones of the mean value model.
- Enhancement of mean value models predictive ability with adequate accuracy.
- Appropriate where the mean value approach exceeds its limitations.

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## ABSTRACT

In this article, a combined mean value–zero dimensional model is developed using a modular approach in the computational environment of Matlab/Simulink. According to that, only the closed cycle of one engine cylinder is modelled by following the zero-dimensional approach, whereas the cylinder open cycle as well as the other engine components are modelled according to the mean value concept. The proposed model combines the advantages of the mean value and zero-dimensional models allowing for the calculation of engine performance parameters including the in-cylinder ones in relatively short execution time and therefore, it can be used in cases where the mean value model exceeds its limitations. A large marine four-stroke Diesel engine steady state operation at constant speed was simulated and the results were validated against the engine shop trials data. The model provided results comparable to the respective ones obtained by using a mean value model. Then, a number of simulation runs were performed, so that the mapping of the brake specific fuel consumption for the whole operating envelope was derived. In addition, runs with varying turbocharger turbine geometric area were carried out and the influence of variable turbine geometry on the engine performance was evaluated. Finally, the developed model was used to investigated the propulsion system behaviour of a handymax size product carrier for constant and variable engine speed operation. The results are presented and discussed enlightening the most efficient strategies for the ship operation and quantifying the expected fuel savings.

## 1. Introduction

The shipping industry has been facing a number of challenges due to the unprecedented rise of fuel prices [1–3], the increasing international concern and released regulations for limiting ship emissions and their impact on the environment [4] as well as the reduction of charter rates [5]. This combination of conditions has brought the subject of energy efficiency to the agenda of the maritime industry and of the corresponding academic research.

Improvements in energy efficiency can be obtained in several areas of ship operations and design [6,7]. Among the different components responsible for energy losses on-board a ship, however, it has been widely shown that the main engine(s), and in a less extent the auxiliary engines, occupy a crucial role, as they are responsible for the conversion of the fuel chemical energy to mechanical, electrical or thermal energy for covering the respective ship demands [8]. In this respect, engine manufacturers have developed a number of measures for improving engine efficiency and reducing pollutant emissions. In electronically controlled

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## Nomenclature

*Symbols*

| | |
|---|---|
| $A$ | area (m$^2$) |
| $BMEP$ | brake mean effective pressure (bar) |
| $bsfc$ | brake specific fuel consumption (g/kW h) |
| $c_d$ | discharge coefficient |
| $c_V$ | specific heat at constant volume (J/kg K) |
| $d$ | Cylinder bore (m) |
| $h$ | specific enthalpy (J/kg); heat transfer coefficient (W/m$^2$ K) |
| $HR$ | Heat release rate (J/°CA) |
| $\dot{H}$ | Energy flow (W) |
| $I$ | Polar moment of Inertia (kg m$^2$) |
| $LHV$ | fuel power heating value (J/kg) |
| $k$ | Coefficients; revolutions per cycle |
| $m$ | mass (kg) |
| $\dot{m}$ | mass flow rate (kg/s) |
| $N$ | rotational speed (r/min) |
| $p$ | pressure (Pa) |
| $pr$ | pressure ratio |
| $P$ | power (W) |
| $Q$ | heat transfer (J) |
| $\dot{Q}$ | heat transfer rate (W) |
| $R$ | gas constant (J/kg K) |
| $r_c$ | compression ratio |
| $t$ | time (s) |
| $T$ | temperature (K) |
| $u$ | specific internal energy (J/kg) |
| $V$ | Volume |
| $V_D$ | Engine displacement volume (m$^3$) |
| $w$ | Velocity (m/s);weight factors (–) |
| $W$ | Work (J) |
| $z_{cyl}$ | number of engine cylinders |

*Greek symbols*

| | |
|---|---|
| $\gamma$ | ratio of specific heats |
| $\Delta$ | difference |
| $\Delta\phi$ | Crank angle difference (°) |
| $\Delta_{cy}$ | engine cycle duration (°) |
| $\varepsilon$ | Air cooler effectiveness |
| $\eta$ | Efficiency |
| $\lambda$ | Air–fuel equivalence ratio (–) |
| $\rho$ | density (kg/m$^3$) |
| $\phi$ | crank angle (°) |
| $\tau$ | torque (Nm) |

*Subscripts*

| | |
|---|---|
| $a$ | air |
| $amb$ | ambient |
| $AC$ | air cooler |
| $AE$ | Auxiliary engines |
| $AF$ | air filter |
| $comb$ | combustion |
| $cor$ | corrected |
| $cy$ | cycle |
| $cyl$ | cylinder |
| $C$ | compressor |
| $d$ | downstream |
| $E$ | engine |
| $e$ | exhaust gas |
| $el$ | electrical |
| $ep$ | exhaust pipe |
| $eq$ | equivalent |
| $ER$ | exhaust receiver |
| $EV$ | exhaust valve |
| $EVO$ | Exhaust valve open |
| $f$ | fuel |
| $GB$ | gearbox |
| $ht$ | Heat transfer |
| $id$ | ignition delay |
| $in$ | inlet |
| $IR$ | inlet receiver |
| $IV$ | inlet valve |
| $IVC$ | inlet valve closing |
| $ME$ | Main engines |
| $out$ | outlet |
| $pump$ | pumping |
| $P$ | propeller |
| $ref$ | reference |
| $scav$ | scavenging |
| $SG$ | shaft generator |
| $Sh$ | shafting system |
| $SOC$ | start of combustion |
| $T$ | turbine |
| $TC$ | turbocharger |
| $tot$ | total |
| $u$ | upstream |
| $vol$ | volumetric |
| $w$ | wall |
| $W$ | Cooling water |

engines [9,10], timings for injection and exhaust valve opening/closing are managed by computer-controlled high-pressure hydraulic systems instead of being operated directly by the camshaft; waste heat recovery systems [11–14] are now used for recovering part of the energy rejected by the engines to produce thermal and/or electrical power; with the aim of improving the propulsion engines low loads performance, retrofitting packages for turbocharger units isolation, exhaust gas bypass and turbochargers with variable geometry turbines have been presented [15–18].

Design, experimentation and prototyping are expensive processes in manufacturing industries, and in particular in the case of marine engines. As a solution to this issue, computer modelling of engines and their systems/components has been extensively used as a mean of testing alternative options and possible improvements during the engine design phase by employing a limited amount of resources. Engine models of a varying range of accuracy and computational time can be employed depending on the required application [19,20]. Cycle mean value engine models (MVEM) [21–29] and zero-dimensional models (0-D) [30–36] are extensively used both for the evaluation of engine steady-state performance and transient response, in cases where the requirements for predicting details of the combustion phase are limited. The former are simpler and faster and provide adequate accuracy in the prediction of most engine output variables [25,29]; the latter include more detailed modelling of the engine physical processes and therefore, more realistic representation of the physical processes as well as higher accuracy can be obtained at the expense of additional computational time.

MVEMs are based on the assumption that engine processes can be approximated as a continuous flow through the engine, and hence average engine performance over the whole operating cycle. As a consequence, the in-cycle variation (per crank-angle degree) of internal parameters such as pressure and temperature cannot