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Computational investigation of a large containership propulsion engine operation at slow steaming conditions

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HIGHLIGHTS

- Extension of the compressor map towards the low speed region using a novel way.
- Incorporation of the new compressor model and a blower submodel to the MVEM tool.
- In-depth study of large two-stoke marine diesel engine operation at low load region.
- Comparison of blower activation and turbocharger cut-out alternatives.
- Blower can be activated in engine load below 25% if one turbocharger is cut-out.

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ABSTRACT

In this article, the operation of a large containership main engine was investigated with emphasis at slow steaming conditions. A cycle mean value approach implemented in the MATLAB/Simulink environment was adopted to simulate the two-stroke marine diesel engine due to the fact that it combines simplicity with adequate prediction accuracy. For accurately representing the compressor performance when the engine operates at low loads, the extension of the compressor map at the low rotational speed region was carried out based on a non-dimensional parameters method incorporating a novel way of calculating the compressor isentropic efficiency. The compressor map extension method results were validated using a corrected similarity laws approach. The engine steady state operation for various loads was simulated and the predicted engine performance parameters were validated using shop trial measurements. Furthermore, the engine transient operation in the load region below 50% was studied and the simulation results including the compressor operating points trajectory are presented and discussed. Based on the obtained results, the influence of the activation/deactivation of the installed electric driven blowers and the turbocharger cut-out on the engine operation was analysed.

1. Introduction

The shipping industry faces multiple challenges in the last few years taking into account the unprecedented rise of fuel prices [1–3] and the consequential impact on ship operating cost which, in turn, affects cargo transport cost [4–6] and company competitiveness and viability. In addition, the increasing demand for shipping transport capacity from last financial crisis [7] is expected to further increase marine fuels demand and as a result the emitted CO_2 emissions. Furthermore, the continuously increasing concern for greening shipping industry activities, the imposed new international and national regulations for limiting greenhouse and non-greenhouse emissions [8–10], as well as the significant reduction of chartered ship rates [11] have introduced additional burdens. Therefore, slow steaming has been rendered as a standard operating procedure in shipping industry especially for the containerships which were sailing at high speeds in the range of 20–24 kn [12]. A significant number of containership operators have even adopted the super slow steaming operating strategy [13] in order to save a considerable fuel amount and limit the emitted gaseous emissions.

In order to achieve the more efficient and environmentally cleaner operation of their products, the engine manufacturers have developed and applied a number of measures including the introduction of the electronically controlled versions of marine diesel engines [14,15], in which the camshaft that exists in traditional

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Nomenclature

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	diameter (m), deviation
h	specific enthalpy (J/kg)
H_L	fuel lower heating value (J/kg)
I	polar moment of inertia (kg m^2)
k	coefficients
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
M_{inl}	compressor impeller tip Mach number
N	rotational speed (r/min)
p	pressure (Pa)
pr	pressure ratio
Q	torque (N m)
$\dot{Q}$	heat transfer rate (W)
r_C	compressor impeller radius (m)
R	gas constant (J/kg K)
t	time (s)
T	temperature (K)
u	specific internal energy (J/kg)
U_C	compressor blade tip velocity (m/s)
$\dot{V}$	volumetric flow rate (m^3/s)
z_{cyl}	number of engine cylinders

Greek symbols

γ	ratio of specific heats
$\Delta\phi_{cy}$	engine cycle duration (deg)
Γ_C	non-dimensional torque coefficient
ζ	proportion of the chemical energy of the fuel contained in the exhaust gas
η	efficiency
ρ	density (kg/m^3)
ϕ	crank angle (deg)
φ	non-dimensional flow coefficient
Ψ	non-dimensional isentropic head coefficient

Subscripts

a	air
amb	ambient
AC	air cooler
AF	air filter
BL	blower
$comb$	combustion
cor	corrected
cyl	cylinder
C	compressor
d	downstream
ep	exhaust pipe
eq	equivalent
ev	exhaust valve
ew	entrained water
E	engine
ER	exhaust receiver
f	fuel
i	isentropic
in	inlet
out	outlet
P	propeller
ref	reference
sp	scavenging ports
SC	scavenging receiver
Sh	shafting system
tip	impeller tip
T	turbine
TC	turbocharger
U	upstream

Abbreviations

ISO	International Organization for Standardization
MCR	maximum continuous rating
MVEM	mean value engine modelling

engine versions for adjusting the injection timing and exhaust valve opening has been replaced by computer-controlled high-pressure hydraulic systems, and waste heat recovery systems [16–18]. In addition, retrofitting packages for turbocharger units isolation, exhaust gas bypass and turbochargers with variable geometry turbines have been presented [19,20] for maximizing the engine efficiency throughout the engine operating envelope, especially when the engine runs at slow steaming conditions. In addition, cylinder cut-out has also been proposed and ways for isolating a number of cylinders at slow steaming operation have been investigated. In consequence, more attention is needed to be paid to the understanding of engine behaviour at low engine load region.

Since the size and weight of two-stroke marine diesel engines as well as their procurement, running and experimental testing costs are enormous [21], various engine modelling techniques have been extensively used for investigating the engine steady-state performance and transient response, for testing various engine designs and for developing the engine control system. In the previously published studies [22–33], various models have been developed and applied for simulating marine engines under steady-state and transient conditions. The most commonly used ones are the zero-dimensional models [22–27] and the cycle mean value engine models (MVEM) [28–33]. The former can represent the engine working processes more accurately but they are more complex, and require a greater amount of input data and execution time. The latter are simpler and need a less amount of input data, while predicting the engine behaviour with sufficient accuracy. The basic assumption adopted in the cycle mean value models is that the air and fuel flows entering and exiting the engine cylinders are continuous. So the engine cycle averaged temporal evolution of the engine operating parameters can be calculated, whereas their in-cycle variation (per degree of crank angle) cannot be represented [33,34].

For accurately representing the engine behaviour at slow steaming conditions, the compressor characteristics at the region of very low rotational speed are required. Since the compressor performance maps provided by turbocharger manufacturers usually do not cover the region below 40% of the maximum turbocharger rotational speed [35], the extension of the compressor map must be performed. In previous studies [35–41], various ways of modelling the turbocharger compressor as well as extending the compressor performance map were described. Jensen et al. [36] used a compressor model based on the dimensionless parameters method, which is considered to be one of the most stable ways to predict compressor performance with adequate accuracy. Guillaume and Vincent [37] proposed an approach based on the turbomachinery physics for modelling the turbocharger compressor and