Simplified calculation of friction mean effective pressure for fast simulation of fuel consumption

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Abstract

An environmentally friendly and economical ship operation can be accomplished through many different methods. Most of these approaches focus on technological solutions, e.g. internal engine measures in order to make the engine more eco-friendly, by changing engine control parameters for a better connection between propulsion system and ship, by usage of different fuels or fuel supplements or installation of exhaust gas-treatment systems. For many ships, it is neither efficient nor economically viable to replace or improve existing power generation or propulsion systems in order to improve efficiency or reduce emissions. Some of the internal measures used to reduce NO_x-emissions like exhaust gas recirculation even lead to a higher fuel consumption. The vessel itself is still controlled by a crew and they should be kept in the loop to improve efficiency. Therefore optimal operational procedures for handling ships and specifically the outcome of engine manoeuvres is a substantial source for eco-friendly ship operations. The German research project MEmBran (Modelling Emissions and Fuel Consumption during Ship Manoeuvres) addresses especially the basis for optimising ship engine manoeuvres. It focusses on very detailed simulation of the processes of currently existing ship diesel engines, especially in a first step 4-stroke engines in order to implement models in wider comprehensive ship handling simulation software. As part of an existing planning and prediction software that can be used on board, it enables the watch keeping nautical officer and the shipping company to forecast and compare the fuel consumption of the ship for each manoeuvre. In order to reach this goal it is necessary to use fast calculating and stable methods that can be used to forecast the power output of the engine and the fuel consumption. This paper discusses an approach to calculate friction mean effective pressure.

Keywords Transient engine operation · Friction mean effective pressure · Engine power output

List of symbols

A	Engine specific friction coefficient	i	Working cycles per revolution
B	Engine specific friction coefficient	K_p	Proportional coefficient
CO ₂	Carbon dioxide	K_I	Integral coefficient
CR	Common rail	K_D	Derivative coefficient
CFD	Computational fluid dynamics	MDO	Marine diesel oil
EEDI	Energy efficiency design index	NCR	Nominal continuous rating
$e(t)$	Error value/control deviation [rpm]	NO _x	Nitrogen oxides
FMEP	Friction mean effective pressure	n	Revolutions per second
$\dot{H}$	Enthalpy flow [kW]	p	Cylinder pressure [Pa]
		$p_{m,ind}$	Net indicated mean effective pressure

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$p_{m,f}$	Friction mean effective pressure
PID	Proportional Integral Derivative
P_e	Effective power [W]
T	Average cylinder temperature [K]
TC	Turbocharger
u	Specific inner energy [kJ/kg]
$u(t)$	Control variable fuel flow [kg/h]
U	Inner energy
V	Volume
V_H	Swept volume
$\dot{Q}_B$	Heat release reaction enthalpy [kJ/s]
$\dot{Q}_W$	Wall heat release [kW]
φ	Crank angle [°]
φ_{BC}	Crank angle at begin of combustion [°]
$\Delta\varphi_{BD}$	Duration of combustion [°]
λ	Air fuel ratio

1 Introduction

A common problem a mariner is confronted is that ships, especially the one he is working on, is responsible for toxic emissions and the release of greenhouse gases. The motivation to start the project MEmBran was to save fuel and reduce emissions in coastal areas. Crew on board often drives ships differently. Especially on short voyages carried out by coastal ships or ferries that carry out the same route every day there are differences in the fuel consumptions that can't only be explained by different weather influences, technical alterations or surrounding traffic. The influence of the crew in certain areas can lead up to 20% more fuel consumption during manoeuvring (Finger et al. [1]). In order to forecast the fuel consumption in connection with a ships model, easy and fast methods must be chosen to calculate the mean effective pressure and the power output of the engine at different states. Therefore a thermodynamic engine model was set up including control system. Test bed runs had been carried out in order to validate the simulated data. One of the main targets of the research project MEmBran was, that the generic engine models should be easy to adapt to other existing engines and therefore a complex simulation of various ship-engine combinations should be possible. The main challenge is that for this case a lot of data are not available and on board test trials need a lot of measurement and are sometimes not even possible due to the fact, that a lot of vessels are only equipped with operational measurement technologies that are sufficient enough to monitor the engine. Therefore a new and simpler method to calculate the effective power output was developed depending on the mean effective pressure.

2 Engine simulation

2.1 Testbed engine

As a testbed for measurements serves the MAN 6L23/30 engine at the University of Applied Sciences Wismar is used. The testbed is a medium-speed, four-stroke, non-reversible and turbocharged propulsion engine, that is equipped with operating measurement technique and is normally used to train future technical officers.

The main engine characteristics are illustrated in Table 1 (MAN [2]). The fuel injection is carried out via a conventional pump-nozzle system. The speed is maintained by a PID controller, while the load is controlled by a water brake. The speed can be set between 520 and 900 rpm. Marine Diesel oil (MDO) was used to carry out test bed trials.

2.2 Engine model

In order to develop a fast calculating but easy extendable engine model all calculations have been made at first in Matlab™ and then be converted into a C++ based Windows application. No commercial code was used for the investigation. The model consist of a zero-dimensional approach based on a simple energy balance to calculate the pressure and temperature inside he cylinders (1).

$$\frac{dU}{dt} = -p_z \frac{dV}{dt} + \dot{Q} + \dot{Q}_W + \dot{H}_{BB} + \dot{H}_{in} - \dot{H}_{out} \quad (1)$$

The inner energy u can be determined by a function of temperature and air–fuel ratio by Justi [3] (2). To maintain the real-time capability it is inevitable to increase the step size of the calculation and taking inaccuracies into account. The pressure volume work $p dV$ expresses the work carried out by the gas during intake, compression, expansion and exhaust phase. $\dot{Q}_B$ is the heat released by fuel combustion. The time, duration and form of heat release can be described using Vibes approach [3–5] (3). $\dot{Q}_W$ is heat flow through the liner

Table 1 Engine main characteristics

Parameter	Unit	
No. of cylinders	–	6
Bore	mm	225
Stroke	mm	300
NCR speed	r/min	900
Nominal continuous rating (NCR) power	kW	960
BMEP at NCR	Bar	17.9
Maximum cylinder pressure at nominal continuous rating	Bar	135

wall. The heat transfer coefficient can be calculated using the approaches of Woschni [6]. $\dot{H}_{in}$ and $\dot{H}_{out}$ stands for the Enthalpy flow through inlet and outlet valve. $\dot{H}_{BB}$ marks the blow by through piston rings and leakages.

$$u(T, \lambda) = 0,1445 \left[1356, + \left(489,6 + \frac{46,4}{\lambda^{0,93}} \right) * (T - T_{Bez}) 10^{-2} + \left(7,768 + \frac{3,36}{\lambda^{0,8}} \right) (T - T_{Bez})^2 10^{-4} - \left(0,0975 + \frac{0,0485}{\lambda^{0,75}} \right) (T - T_{Bez})^3 10^{-6} \right] \quad (2)$$

The Enthalpie flow bases on the results of (2) at inlet or outlet and is depending on the crank angle φ , the geometrical opening and closing of the inlet and exhaust valve and the gas states in the exhaust gas and charge air receiver.

$$\frac{dQ_B}{d\varphi} = \frac{Q_B}{\Delta\varphi_{BD}} * 6.908 * (m + 1) * y * e^{-6,902y^{m+y}} \quad (3)$$

$$y = \frac{\varphi - \varphi_{BC}}{\Delta\varphi_{BD}} \quad (4)$$

The shape factor m is varied using the function of Woschni & Anisits [7] and is depending on pressure, and temperature when the inlet valve closes as well as the current engine speed. The total amount of released heat Q_B is depending on the filling at fuel pump. The amount of fuel that is injected depends on the speed setpoint of the nozzle and current engine speed and is calculated using a PID-controller in order to be used as a dynamic model.

$$u(t) = K_p e(t) + K_I \int e(t) dt + K_{DP} \frac{de(t)}{dt} \quad (5)$$

The engine model contains a model to calculate the charge air pressure and the exhaust gas pressure. As a result the pressure inside the cylinder and the net indicated mean effective pressure using the swept volume V_H can be determined:

$$p_{m,ind} = \oint \frac{p_z \cdot dV}{V_H} \quad (6)$$

To determine the effective power P_e output of the engine a calculation of the mean effective pressure $p_{m,e}$ is necessary. The difference between net indicated mean effective pressure and mean effective pressure is the friction mean effective pressure $p_{m,f}$. In connection with the swept Volume with the swept volume, the revolution per second n and the working cycles per revolution i the effective Power can be calculated.

$$P_e = (p_{m,ind} - p_{m,f}) \cdot V_H \cdot i \cdot n \quad (7)$$

If the calculation of mean effective pressure is faulty, the calculated torque could be wrong, so the engine

speed would drop or increase. Consequently the amount of injected fuel would be adjusted by the PID-controller. An exact amount of fuel is necessary to calculate emissions like nitrogen oxides using Zeldovichs [8] approach

combined with a two zone model (Heider [9]). The counter torque for acceleration can be provided by a static propeller curve or a dynamic ship model.

2.3 Calculation of friction mean effective pressure

The friction mean effective pressure $p_{m,f}$ can be calculated using different approaches like the widely confirmed and accepted model Chen-Flynn [10]/Ciulli [11] friction correlation model. These models base normally on a variety of sets like engine speed, bore and maximum cylinder pressure and require a lot of measurements due to their dependencies on a lot variables. For the calculation to be used in fast time simulation a few assumptions has to be made. The first assumption is that the frictional losses are depending on the operational point of the engine. The next assumption is that the friction inside the cylinder is depending on the pressure applied from the combustion. These two assumptions build the main dynamic part about the friction and that there should be a focus on in a dynamic prediction. Another assumption is that the engine oil temperature has no influence in the normal ship operations in contrary to a road application. The engine oil is preheated due to the continuous cleaning process and therefore the calculation can be made without a change in oil temperature.

For the project MEmBran it was possible to project an easier model consisting of just one additional measurement point additionally to one test bed result.

$$p_{m,f} = A * e^{\frac{B \cdot p_{m,ind}}{p_{m,ind,n}}} \quad (8)$$

In Eq. (8) the coefficients A and B are motor specific factors. $p_{m,ind,n}$ is the net indicated mean effective pressure at nominal continuous rating. The net indicated pressure takes the whole 720° —cycle for a four-stroke engine into consideration. $p_{m,ind}$ is the calculated the net indicated mean effective pressure at the current operation point from the engine simulation of chapter 2.2. The indicated mean effective pressure is theremain dependence from the engine model. Therefore the calculation of the effective power and torque as mentioned in (7)

can be carried out and build a dynamic model. For the 6L23/30 engine the factor A was determined as 1.3318e5 and B=0.839. The results of this approach are discussed in chapter 2.4. The data are consisting of two measurements. For the determination of the two factors only two validation points of the calculations of Table 2 has been needed. Due to the construction of the formula A and B are dependent on each other to find a correct solution at nominal power output. This measurement is normally given during test bed trials or can be easily measured on board on ship during operation. In order to maintain the overall goal of data accessibility during ship operations the measurements can depend on the ships route and track.

2.4 Model validation

To validate the model 26 steady operating points of different propeller curves and a generator curve had been simulated. The simulated engine performance parameters were compared with the respective data experimentally obtained from testbed trials. The data of used operation points are provided in Table 2.

Each Measurement point was compared to fuel consumption and calculated mean effective pressure. Basing on the model of chapter 2.2 and the simulated output of the net indicated mean effective pressure and the calculated friction mean effective pressure as proposed. Figure 1 shows the result of the calculation of mean effective

Table 2 Measurement validation points

Measurement No	1	2	3	4	5	6	7	8	9	10
Speed (rpm)	900	900	900	900	900	900	900	900	900	861
Brake power (kW)	961	864	770	670	578	481	385	288	192	876
Measurement No	11	12	13	14	15	16	17	18	19	20
Speed (rpm)	835	799	755	719	660	599	550	521	552	603
Brake power (kW)	777	679	585	472	385	288	192	155	188	256
Measurement No	21	22	23	24	25	26				
Speed (rpm)	655	702	765	804	851	893				
Brake power (kW)	322	402	489	591	728	839

Fig. 1 Comparison of measured friction mean effective pressure using new method and Chen-Flynn suggested

Fig. 2 Comparison of simulated fuel consumption and measured fuel consumption

pressure in various measurement points. Additionally the calculation using the widely approved method proposed by Chen-Flynn/Ciulli is displayed in red circles.

Furthermore the engine fuel consumption is displayed in Fig. 2. The simulated output results from the dynamic model PID-controller shows that the presented approach is overall more accurate for this engine. 12 of 26 measurement points have a deviation of less than 1%. A further 11 measurements are in a 2% range. Therefore this method seems to be fit for purpose a fast time calculation as proposed by Finger et al. [12] and can be used for a prediction of fuel consumption in specific manoeuvres.

In order to find proof the validation a cross-checking has been carried out. In this case 25 of 26 measurements of Table 1 has been chosen as the second data source for calibration of the friction model as proposed in chapter 2.3 and the simulation for each measurement point has been executed. The first data source, the nominal continuous rating point, was kept as the one steady point. Figure 3 shows the fraction of all simulated friction mean effective pressure and the measured friction mean effective pressure measured during test bed trials. Independent of all the sources a maximal deviation of 0.7% has been detected.

3 Results & Discussion

The presented results are good results for the investigated MAN 6L23/30 and could not be tested on other engines due to a lack of test beds. Therefore more data are needed to validate the proposed approach on other engines. The test had been carried out with an engine that is conventional driven via fuel pump. The calculation

Fig. 3 Fraction of Measured Value based on cross-referenced test-bed data

had been based on a few assumptions like a constant oil temperature which should be considered as normal during ship operations. Nevertheless should different oil temperatures show different with the same coefficients as proposed in chapter 2.3. Future research should focus on a discussion about the impact on engine oil temperatures. It should be possible to combine these results with existing data of engine oil temperature and friction losses. The next focus in future research should be the applicability depending on different oil qualities and quantities. The main assumption that the friction is caused by the movement and the pressure inside the cylinder and that it is independent from the oil and therefore future test and additions could also focus on the reduction impacts on the factors A and B. The proposed contribution should not be considered as a substitution to commonly accepted CFD models for the construction and enhancement process. It is a simplified approach to calculate the engine power output based on limited measurement data in connection with common and emission and engine models. Simulations and measurements and the wide range of validity as can be seen in Fig. 3 show a strong applicability for the test bed engine. The approach had been already tested with a 6M20 Common Rail (CR) engine with preliminary and not validated data showing differences between a generator curve and in propeller mode. In this case there is only one set of variables valid either for generator or propeller mode. Therefore this approach is so far only validated for the presented conventional test bed engine and is approximately in need of two sets of data for CR-Systems depending on the operation mode. There are no results from 2-stroke engines or dual-fuel engines.

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Declarations

Conflict of interest The author(s) declare that they have no competing interests

Ethical approval Hereby, I Georg Finger consciously assure that for the manuscript “*Simplified calculation of friction mean effective pressure for fast simulation of fuel consumption*” the following is fulfilled: (1) This material is the authors’ own original work, which has not been previously published elsewhere. (2) The paper is not currently being considered for publication elsewhere. (3) The paper reflects the authors’ own research and analysis in a truthful and complete manner. (4) The paper properly credits the meaningful contributions of co-authors and co-researchers. (5) The results are appropriately placed in the context of prior and existing research. (6) All sources used are properly disclosed (correct citation). Literally copying of text must be indicated as such by using quotation marks and giving proper reference. (7) All authors have been personally and actively involved in substantial work leading to the paper, and will take public responsibility for its content.

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