COMPUTATIONAL INVESTIGATION OF A LARGE DUAL FUEL MARINE ENGINE

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ABSTRACT

Continuously increasing environmental demands in conjunction with the planned strong penetration of the LNG, render the use of LNG as an attractive alternative marine fuel. In this framework, the traditional ship propulsion plants based on Diesel engines running with HFO, should be revisited and compared to the more efficient and environmentally friendly propulsion systems that use dual fuel engines. The present study deals with the computational investigation of a marine four-stroke DF engine, in both diesel and DF mode operation. The engine model was set up in GT-Power and used to examine the differences of the performance and emissions parameters of the investigated engine operation at steady state conditions. The engine diesel mode was initially set up and the model was calibrated to adequately represent the engine operation. Subsequently, the engine dual fuel model was set up by considering the injection of two different fuels; methane and pilot diesel fuel. The derived results were analysed for revealing the differences of the engine behaviour at each engine mode. In addition, a method was proposed for facilitating the dual fuel mode heat release analysis. By applying this for the analysis of the derived pressure diagrams, the dual fuel combustion characteristics are analysed and discussed.

Keywords: Dual Fuel engine, simulation, performance/emissions prediction, heat release analysis.

NOMENCLATURE

1D	*One-Dimensional*
3D	*Three-Dimensional*
BMEP	*Brake Mean Effective Pressure*
BSFC	*Brake Specific Fuel Consumption*
BSEC	*Brake Specific Energy Consumption*
CO	*Carbon Monoxide*
CO_2	*Carbon Dioxide*
DF	*Dual Fuel*
ECA	*Emission Control Area*
ECU	*Engine Control Unit*
HC	*Hydrocarbons*
HFO	*Heavy Fuel Oil*
IMO	*International Maritime Organization*
LNG	*Liquefied Natural Gas*
MARPOL	*International Convention for the Prevention of Marine Pollution*
MGO	*Marine Gas Oil*
NG	*Natural Gas*
NOx	*Nitrogen Oxides*
PM	*Particular Matter*
RPM	*Revolutions per Minute*
SOx	*Sulphur Oxides*

1. INTRODUCTION

Gaseous emissions from marine diesel engines, i.e. NOx, SOx, PM, HC, CO and CO_2, have been steadily increasing throughout the last years. It is estimated that shipping accounts for 2-3% of global emissions (Bows–Larkin et al., 2014). In order to control these emissions and eliminate air pollution, various regulatory bodies such as IMO (IMO, 2014a) (IMO, 2014b), EMSA (EMSA, 2015) and EPA (EPA, 2015) have adopted a series of regulations. The International Maritime Organisation (IMO) and the national authorities introduced legislation for limiting non-greenhouse gaseous emissions including NOx and SOx, as well as the greenhouse gaseous emissions; mainly CO_2. These amendments of the international legislation introduced the Energy Efficiency Design Index (EEDI) as well as the Ship Energy Efficiency Management Plan (SEEMP) that can be based on the Energy Efficiency Operational Indicator (EEOI). The expected benefits from the implementation of the above include not only the reduction of the environmental impact of gaseous emissions, but also the reduction of the fuel consumption throughout the ship lifetime leading to minimised operating costs that affect the competitiveness of the shipping companies (Theotokatos & Tzelepis, 2013).

Due to these regulatory limitations that have arisen throughout the last years, the engine manufacturers e.g. (MAN, 2009) and (Wärtsilä, 2015a), as well as Classification societies e.g. (GL, 2013) have performed studies focusing on the emissions reduction. In addition, the engine manufacturers accomplished efforts to improve the combustion characteristics, so that to maximise the engine efficiency, thus reducing fuel consumption, as well as to reduce the engines gaseous emissions. Marine engine manufacturers have also developed dual-fuel versions both for large two stoke slow speed engines and small to medium size, four-stroke engine. These engines can run on dual fuel and diesel modes; in the former mode by using natural gas and pilot diesel fuel, in the latter mode by burning diesel fuel (HFO or MGO).

Natural gas (NG) is the greenest fossil fuel that is a proven and available solution. Whilst conventional diesel fuels will remain the main option for the majority of the existing vessels in the near future, the commercial opportunities of NG are attractive for new-built vessels. The sulphur content of natural gas is almost zero (about 0.004% by mass to mass basis), which is well below the 0.1% sulphur required in ECAs from 2015. Therefore, the SOx emissions of the engines operating in DF mode are very low (SOx emissions can be reduced up to 90-95% compared with the diesel mode operation at HFO). In addition, the DF engines can achieve up to 85% NOx emissions reduction (as they operate in the lean burn combustion concept) and CO_2 emissions decrease up to 20-25% (due to the natural gas lower carbon to hydrogen ratio), whereas the particulate matter (PM) emissions are almost eliminated. Furthermore, there is no visible smoke during engine operation at DF mode, whereas the price of LNG is also attractive; about 60% of the HFO price (Wärtsilä, 2012) (Livanos et al., 2014). Four-stroke DF engines operate at low gas pressure (5-7 bar) with a typical brake mean effective pressure (BMEP) at around 25 bar.

Previous research on dual fuel engines are reported in detail in Karim (2015). Past research efforts dealt with the increase the efficiency of the DF engines, optimisation of the pilot injection and gas substitution rate, extending the operating range of DF mode and avoiding knocking as described in Xu et al. (2014). Simulation tools of various complexities (1D/3D) (Merker et al., 2006) have been used extensively for investigating the DF engine steady state performance and transient response (Xu et al., 2014) as well as for analysing marine engines and ship propulsion systems (Theotokatos & Tzelepis, 2013), (Baldi et al., 2015)

The main objective of the present study is to develop and use a model capable of simulating a marine dual-fuel engine. Based on this, the investigation of the engine steady state performance and exhaust emissions can be carried out at the engine discrete operating modes (diesel/dual fuel). By analysing the derived results, the processes that affect the engine efficiency and gaseous emissions can be revealed enabling the elaboration on possible ways to increase the engine efficiency and reduce emissions. In addition, a new method based on three Vibe functions is developed for calculating the heat release of dual fuel engines, the validity of which has been verified based on the derived pressure diagrams. The obtained results can be used as guidance during the design process of the dual fuel engines.

2. ENGINE MODELLING

2.1 ENGINE SPECIFICATIONS

The Wärtsilä engine W9L50DF was used for the present study, which is a 4-stroke, non-reversible, turbocharged and intercooled DF engine. The engine consists of nine cylinders placed in-line. This type of engine is widely used due to its high power output along with its fuel flexibility, low emission rates, high efficiency and reliability. The engine details are reported in the manufacturer project guide (Wärtsilä, 2014). The main engine characteristics are illustrated in Table 1. The engine operation at constant speed of 514 rpm was investigated in the present study. Engine operation under these conditions can be found in electric propulsion systems, where engine-electric generator sets are used for producing the ship required electric energy.

Table 1: Engine main characteristics.

MCR	8775 kW @ 514 rpm	
BMEP	20 bar	
	Gas mode	Diesel mode
BSEC	7258 kJ/kWh	
BSFC Pilot fuel	1.0 g/kWh	190 g/kWh
Number of valves	2 inlet and 2 exhaust valves per cyl.	
Cylinder configuration	9 in-line	
Turbocharger	1 unit	

2.2 ENGINE MODELLING AND PARAMETERS CALIBRATION

The software used in the present work is GT-Power™ (Gamma Technologies, 2015), which is a widely used 1D simulation program for engine modelling and analysis. The data required for the modelling stage as input was acquired from the product guide (Wärtsilä, 2014) and the engine 3D model (Wärtsilä, 2015b). Initially, the model for one cylinder block was developed (Figure 1a) and validated for the diesel mode operation and subsequently, the model was extended to cover the dual fuel operation (Figure 1b). Then the whole engine model including the turbocharger and air cooler (shown in Figure 1c) was developed.

The steps required to set up the engine model are as follows. Initially, the component blocks are selected, which sufficiently represent the engine layout and the appropriate interconnections are established. Then, the input data of all blocks are set. Preliminary calibration of the model constants is performed for a reference point and simulation runs are carried out. Finally, the fine tuning of the model constants is accomplished, so that the required accuracy is obtained. The following input data are needed to set up the model: the engine geometric data, the intake and exhaust valves profiles, the compressor and turbine performance maps, the constants of engine sub-models (combustion, heat transfer and friction), the engine operating point (load/speed) and the ambient conditions. Initial conditions are required for the temperature, pressure and composition of the working medium contained in the engine cylinders, pipes and receivers.

The Woschni heat transfer model (Woschni,1967) was used to calculate the in-cylinder gas to wall heat transfer coefficient. The heat release rate was simulated according to single Vibe model (Merker et al., 2006) for the engine diesel operation mode, whilst in the case of the DF mode, the multi-Vibe model was utilised by imposing two different Vibe curves representing the pilot diesel fuel and gas fuel (methane) combustion, respectively.

For the simulation of the diesel mode operation, MGO was used as the model fuel type along with the injection timing and injected fuel amount; both were provided as function of the engine load. However, for DF mode, the specific details of the gas injection system were not available and thus, an equivalent gas injector approach connected on the engine cylinders was adopted according to which, the amount of injected gaseous fuel was calculated by considering the known gas specific energy consumption and the fuel lower heating value. More specifically, the gas fuel injector element is connected to the cylinder, injecting the appropriate fuel (methane) amount during the cylinder induction stroke. In this way, the air-fuel equivalence ratio was kept within the required limits.

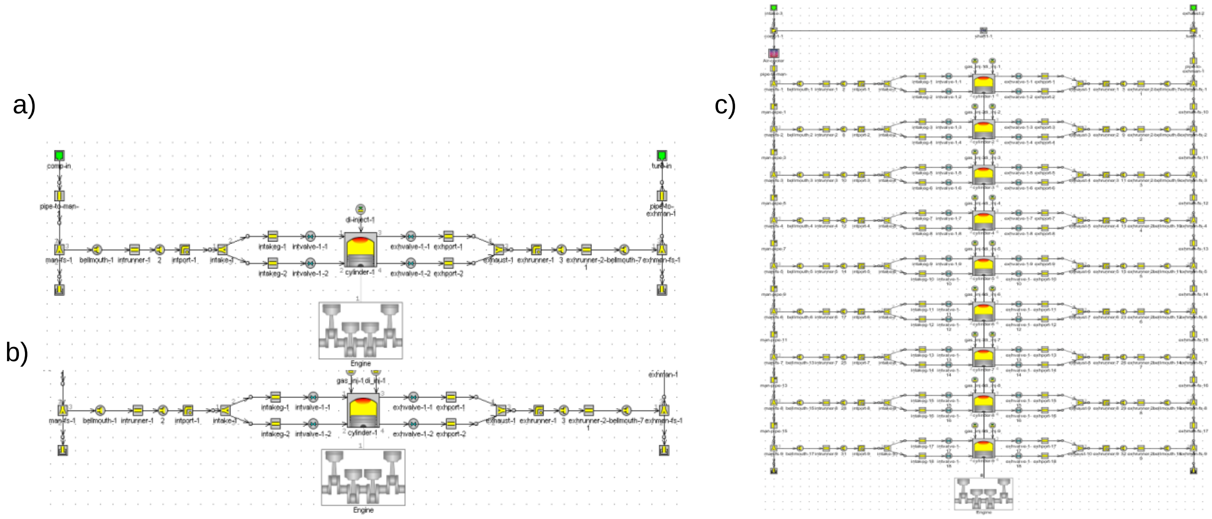

Figure 1: Model layouts; a) 1-cylinder diesel mode, b) 1-cylinder DF mode c) overall engine model air-cooler and turbocharger.

Finally, the complete engine model with the intercooler and the turbocharger was built for the diesel and DF modes. For this model, a waste gate valve was also used, which opens above 75% load. The open waste gate controls the pressure level of the engine receivers and as a result the engine cylinders air flow rate, which allows the engine operation at the DF mode within the required air-fuel equivalence ratio window (values between 2.0 and 2.2 are accepted). The developed GT-Power™ models layouts are presented in Figure 1.

2.3 DUAL FUEL ENGINE HEAT RELEASE CALCULATION MODEL

2.3 (a) Calculation Methodology

During the engine design stage, the boundary conditions for cycle simulations are provided by using in-cylinder models, in which the combustion model represented by the heat release profile is the basic input. In turn, the heat release can be derived when the in-cylinder model gets accurate enough for the engine performance prediction or when pressure measurements are available. Ignoring the combustion effects between diesel and natural gas, the combustion process in dual fuel engine includes three parts: the diesel premixed combustion, the diesel diffusive combustion and the natural gas premixed combustion. Thus, the total heat release rate of the engine operating in dual fuel mode can be calculated by adding three different single Vibe curves, two for the diesel pilot fuel combustion and one for the natural gas combustion. By assuming that the total heat release is obtained by adding the individual heat release rates of pilot fuel and natural gas , the following equation is derived, which is able enough to depict the total heat release by considering eight parameters (SOC_D, EOC_D, SOC_N, EOC_N, b_1, m_1, m_2 and m_3):

$$Q_{comb}=\left[b_1\cdot\left(1-e^{-a\cdot\tau_D^{m_1+1}}\right)+b_2\cdot\left(1-e^{-a\cdot\tau_D^{m_2+1}}\right)\right]\cdot m_{f,D}\cdot u_{comb,eff,D}+b_3\cdot\left(1-e^{-a\cdot\tau_N^{m_3+1}}\right)\cdot m_{f,N}\cdot u_{comb,N} \quad (1)$$

In eq. (1), b_1 and b_2 denote the weighting factors for the pilot fuel premixed and diffusive combustion parts, the sum of which should be equal to 1; b_3 is the weighting factor for the natural gas premixed combustion; m_1, m_2 *and* m_3 are the shape factors of the considered Vibe curves; $m_{f,D}$ and $m_{f,N}$ denote the injected fuel amount per cycle and cylinder for the pilot fuel and natural gas, respectively; $u_{comb,eff,D}$ and $u_{comb,N}$ represent the effective heat of combustion for the pilot fuel and natural gas, respectively. The normalised time (τ) is calculated by using the following equation as function of start of combustion for each fuel (SOC_D , SOC_N), end of combustion for each fuel (EOC_D, EOC_N) and crank angle (CA):

$$\tau_D=\frac{\text{CA-}SOC_D}{EOC_D-SOC_D} \quad \text{and} \quad \tau_N=\frac{\text{CA-}SOC_N}{EOC_N-SOC_N} \quad (2)$$

The eight Vibe parameters that meet the required accuracy are obtained through quantitative analysis of the in-cylinder pressure results derived by using the detailed model described in section 2.3. Eq. (1) provides the total heat release rate. The heat release rate of the pilot fuel and the natural gas could be calculated by considering the following equations:

$$Q_{comb,D}=\left[b_1\cdot\left(1-e^{-a\cdot\tau_D^{m_1+1}}\right)+b_2\cdot\left(1-e^{-a\cdot\tau_D^{m_2+1}}\right)\right]\cdot m_{f,D}\cdot u_{comb,eff,D} \quad (3)$$

$$Q_{comb,N} = b_3 \cdot \left(1 - e^{-a \cdot \tau_N^{m_3+1}}\right) \cdot m_{f,N} \cdot u_{comb,N} \tag{4}$$

The relative residual distribution between the derived pressure data and the heat release calculation results is selected to quantitatively analyse the accuracy of the presented HR model. The used criterion is described as follows: the relative residual near peak pressure varies in the range of -10%-10% (for drastic combustion pressure fluctuation), whilst in other crank angle regions, the range turns to be in the range of -5%-5%; the crank angle error for the peak pressure point should be less than ±3°. Crank angle near peak pressure is defined as (CA_1-Δ, CA_1+Δ), in which Δ represents the crank angle difference between peak pressure point crank angle (α_1) and start of combustion (*SOC*).

2.3 (b) Model in MATLAB/SIMULINK Environment

The heat release calculation model was built in MATLAB/SIMULINK computational environment. Figure 2 illustrates the upper layer of the model structure. The primary input of the model is the pressure diagram (pressure versus crank angle); the required cylinder volume is calculated by using the provided crank angle input and the cylinder geometric data. The model output includes the in-cylinder temperature and Combustion Reaction Rate (CRR), which is one of the heat release calculation parameters. Figure 3 presents the combustion rate calculation sub-model, in which the diesel and natural gas calculations are considered separately.

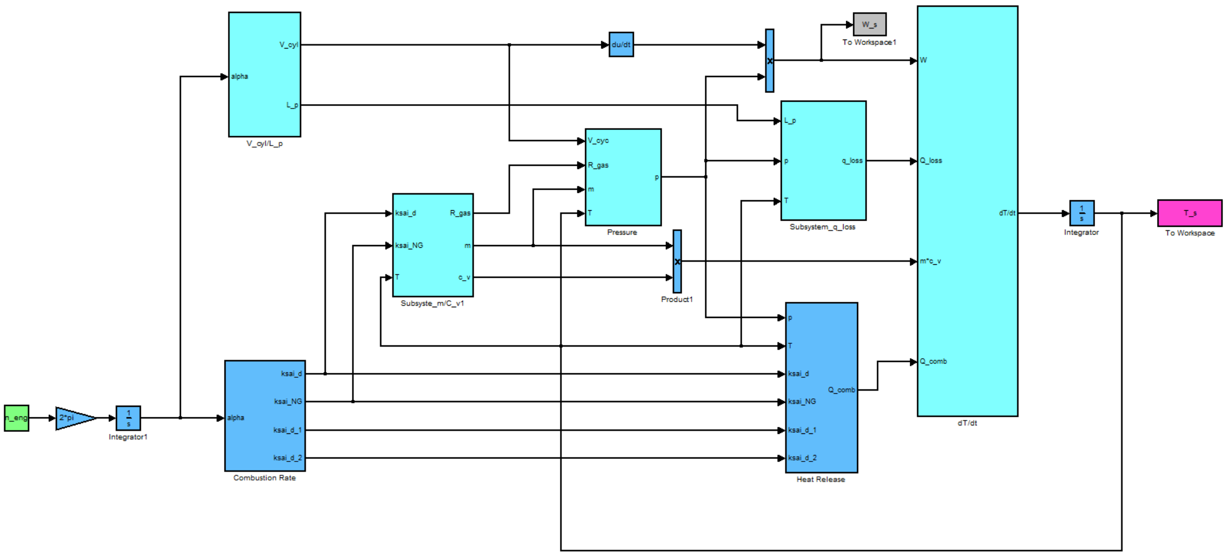

Figure 2: Heat release calculation model in MATLAB/SIMULINK environment

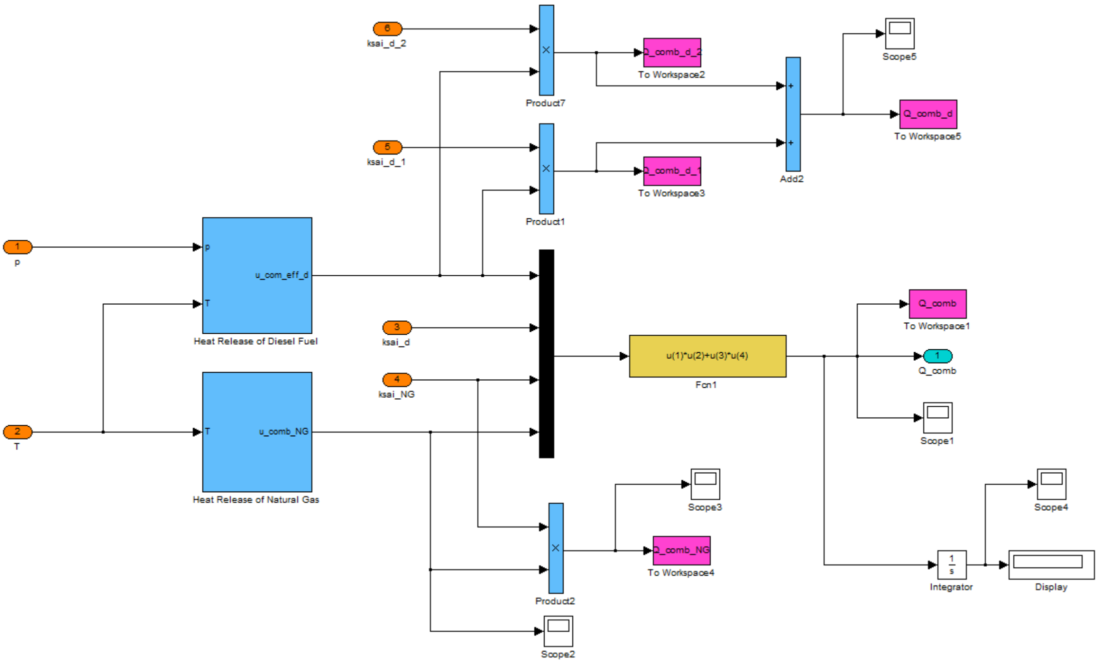

Figure 3: Combustion rate calculation sub-model

3. RESULTS AND DISCUSSION

The investigated marine DF engine steady state operation at both diesel and dual fuel modes was examined by performing simulation runs in a load range from 25% to 100% and constant engine speed at 514 rpm. A set of the derived results including cylinder maximum (peak) pressure, indicated and brake mean effective pressures, brake specific energy consumption, maximum in cylinder temperature of burnt gas zone, air-fuel equivalence ratio, brake efficiency, specific NOx emissions and specific CO_2 emissions is presented in Figure 4. By comparing the simulation results with the engine parameters provided in the engine project guide (Wärtsilä, 2014), it was derived that the obtained accuracy was quite adequate (within the range of ±3%). Therefore it can be concluded that the simulation can be used to represent the engine steady state behaviour. By considering the derived heat release rate diagrams (the HRR diagram for 100% load is shown in Figure 8), it can be inferred that the diesel mode combustion starts closer to the cylinder top dead centre (TDC), whereas in the case of DF mode the pilot injection and combustion starts earlier to avoid knocking problems. The DF combustion tends to exhibit a reduced diesel premixed combustion part due to the reduced injected diesel fuel quantity. The duel fuel operation also produces a longer ignition delay due to presence of natural gas in the combustion chamber as it is also reported in Liu & Karim (1997). The peak heat release rate of dual-fuel combustion is slightly higher and main combustion duration is shorter than the ones of diesel mode. However, lower maximum pressure level is observed in the case of DF mode, which is attributed to the engine turbocharger operation and the opening of waste gate at the high load region. In terms of the air-fuel ratio (λ), it is observed that the engine operates with almost constant λ values around 2.2 in DF mode, whereas the obtained values for λ are much higher in the diesel mode (in the range of 4.20 to 3.8), which means that more air passes though the engine cylinders in the latter case. Thus, it is expected that the turbocharger compressor operating point is different in each mode, which denotes that the turbocharger matching needs special attention for a DF engine compared to the diesel engine respective process, as in the former case the requirements for two discrete modes need to be satisfied.

In terms of the engine power output and mean effective pressures behaviour, it can be observed that similar values are obtained in each mode; the indicated mean effective pressure of the diesel mode seems to be only slightly greater, however the brake mean effective pressures in both modes are exactly the same as the difference is compensated by the slightly higher friction mean effective pressure (due to the greater maximum pressure of the diesel mode). In terms, of the engine efficiency at the two operating modes, it can be observed that the DF mode is more efficient at loads above 70% obtaining values up to 47.3% at 100% load. Operating in diesel mode, the engine obtains its higher efficiency of 44.6% at 75% load, whereas the engine efficiency only slightly changes in the load region 70% to 100% (it varies between 44% and 44.6%). For the DF mode, the efficiency decreases at a more steep way as the load decreases reaching a value of at 32.7% at 25 % load, whereas 37.4% efficiency is obtained at the same load for the diesel mode. This is attributed to the specific characteristics of diesel and DF operating modes. Similar conclusions can be derived by analyzing the brake specific energy consumption, which is the reciprocal of engine brake efficiency. The energy provided by the pilot diesel fuel accounts for 0.3% to 2.3% of the totally supplied fuel energy (the values increase with decreasing load).

Considering the calculated NOx and CO_2 emissions, the following remarks can be noted. Specific NOx emissions are lower for the case of DF mode operation. It should be noted that to calculate NOx emissions, a two-zone model was used in GT-Power, which was calibrated only for 100% load operation at diesel mode. Then the model was used to predict the NOx emissions at the other investigated loads and operating modes. The NOx emissions for the diesel mode comply with Tier II limits, whereas the Tier III limit requirements are satisfied for the DF mode. In addition, higher values of the specific NOx emissions are obtained as the load reduces, which is also reported in Heywood (1998). The NOx differences between the engine operating modes can be explained by considering the in-cylinder burnt zone temperature. As it can be observed from Figure 4, the maximum temperature of the burnt zone for the diesel mode is higher than the respective values obtained for the DF mode operation. In average, a reduction of 85% in NOx emissions is obtained when changing operating mode from diesel to DF. The CO_2 emissions of the DF mode are also reduced (25% in average) due to the lower carbon to hydrogen ratio of the natural gas compared to the one of diesel fuel. Larger reduction is obtained at engine loads above 70% where the efficiency difference between the DF mode and diesel mode is greater. In summary, it can be concluded that the engine environmental impact is much lower when the engine operates at DF mode.

The heat release analysis results for the case of 100% load engine operation in DF mode are presented in Figures 5-10. By adjusting the used multi-Vibe function parameters, the in-cylinder pressure and relative residual distribution were obtained by using the HRR model described in section 2.3.2, as shown in Figure 5 (denoted as Simulation) and 6, respectively. The peak pressure value derived from GT-Power (denoted as Measured in Fig. 5) was found to be 135.6 bar and occurs at a crank angle point of 195.8°, whilst the peak pressure value calculated by using the HRR model was 125.6 bar located at 1197.7° CA. Crank angle range near peak pressure is calculated to be between 166°and 225.6° by analysing the pressure diagram and estimating the start of combustion (*SOC)*. It can be observed from Figure 6 that the relative residual varies in the range of -8.5% to -1% near peak pressure crank angle point and in the range of -3% to 1.3% in other regions, which meets the defined criterion explained in previous section. Therefore, it can be concluded that the in-cylinder process model can predict the dual fuel engine operating performance with acceptable accuracy.

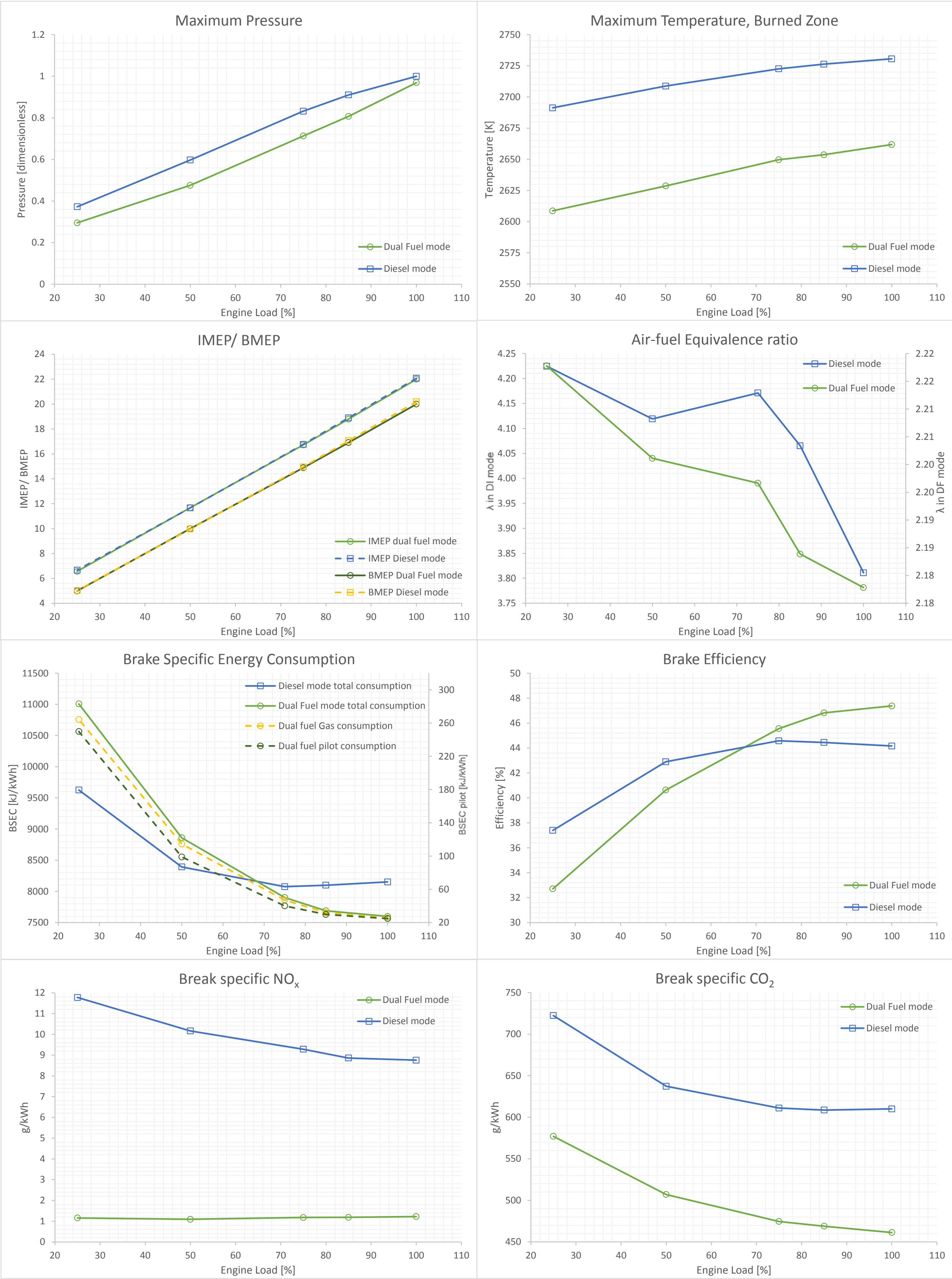

Figure 4: Results for diesel mode and dual fuel mode operation.

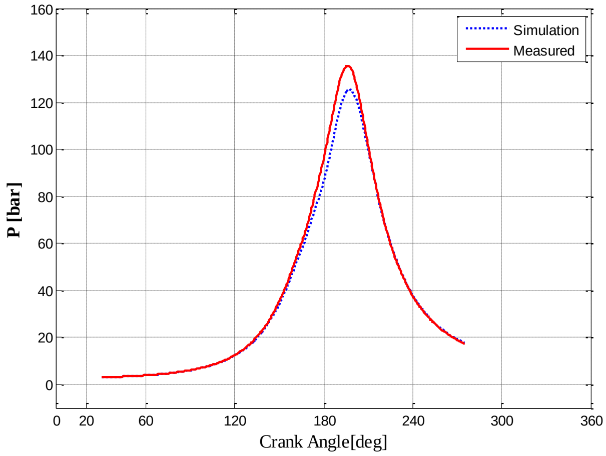

Figure 5: In-cylinder pressure

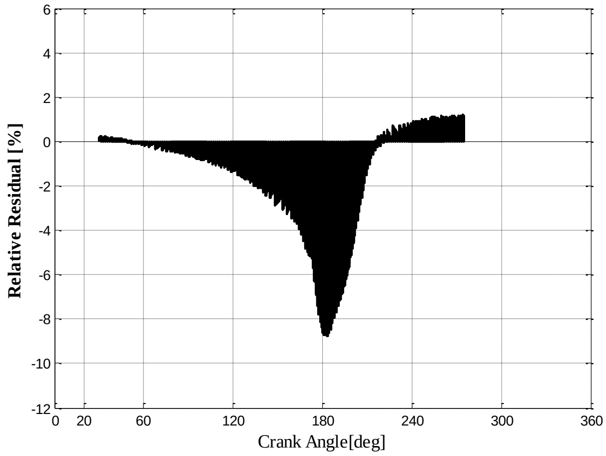

Figure 6: Relative residual distribution

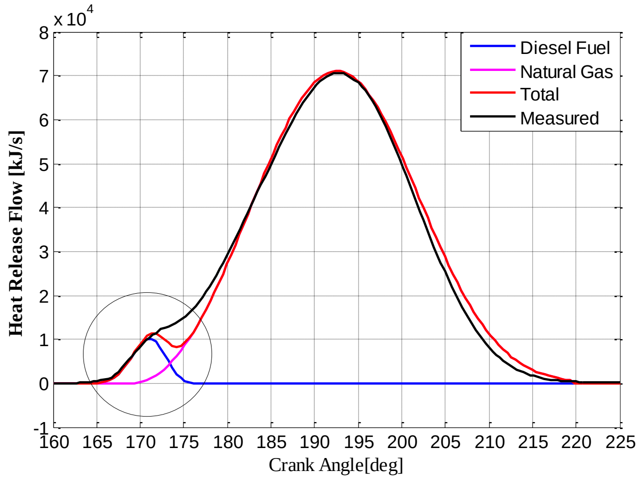

Figure 7: Heat release rate

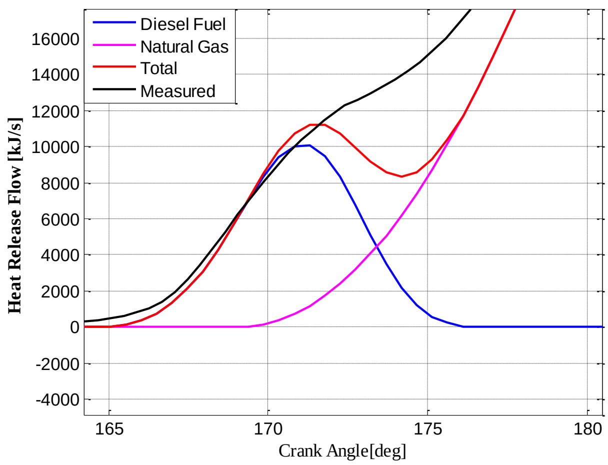

Figure 8: Zoom in of Figure 7

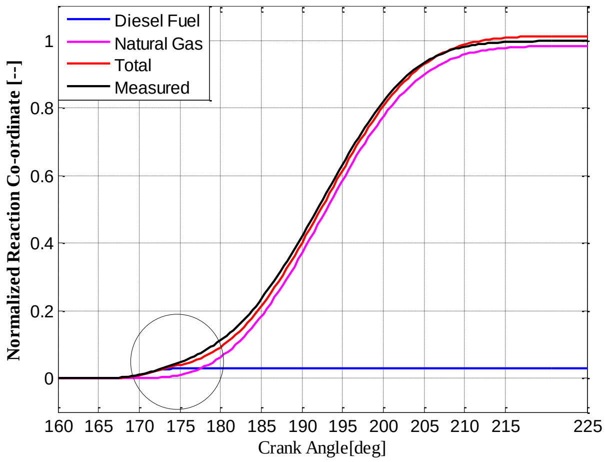

Figure 9: Normalized reaction coordinate

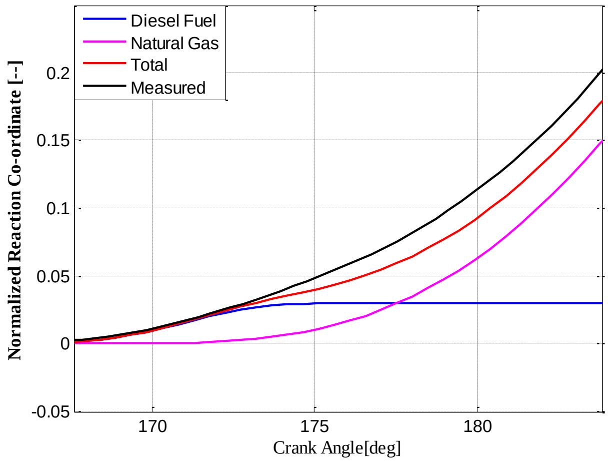

Figure 10: Zoom in of Figure 9

Table 2: Vibe Parameters.

SOC_D	SOC_N	EOC_D	EOC_N	b_1	m_1	m_2	m_3
164	169	176	219.5	0.1	0.43	3.2	2.05

The corresponding Vibe parameters that meet the set accuracy are listed in Table 2. The heat release rates are shown in Figures 8 and 8, whereas the normalized reaction coordinates are illustrated in Figures 9 and 10. According to the calculated value of b_1, the premixed combustion of pilot diesel fuel represents the 10% of the total pilot fuel combustion, with the rest 90% representing the pilot fuel diffusive combustion. The combustion duration of the pilot fuel last 12^o crank angle, whereas the natural gas combustion duration was estimated 50.5^o crank angle. It can be derived from Figures 9 and 10 that the pilot fuel energy approximates to 3% of the total energy provided by the combustion of both fuels. The results presented in Figures 7 and 9 denote that the total heat release rates, determined by eight Vibe parameters exhibit adequate agreement with the simulation detailed model data and therefore it is concluded that the proposed HRR analysis model can be used with fidelity in analysing pressure diagrams from DF engines.

4. CONCLUSIONS

In the present study, a marine four-stroke dual fuel engine was investigated by using GT-Power software in both diesel and DF mode operation. In addition, a heat release rate calculation model was developed for the cylinder pressure analysis of the dual fuel engine mode.

The main findings of the conducted research are summarised as follows:

- The developed model can predict with adequate accuracy the engine performance and emissions parameters both for the diesel and DF operation. For the DF case, the equivalent injector approach was used according to which the natural gas is injected during the induction stroke within the engine cylinder.
- The engine in the DF mode operates with almost constant air/fuel equivalence ratio around 2.2, whereas much higher values of the air-fuel equivalence ratio are used for the diesel mode corresponding to greater air flow. A waste gate is used to control the air flow and therefore the air/fuel ratio at the high load region (above 75%). Special attention must be paid during the turbocharger matching process as there are different requirements in each operating mode.
- In the DF mode, the engine operates at lower receivers and in-cylinder pressure level. However, the mean effective pressure and power output is kept at the diesel mode levels.
- The DF mode is more efficient than diesel mode for loads above 70%; however less efficient operation was observed at the lower load region.
- The CO_2 emissions in the DF mode reduced 25% in average due to the natural gas low carbon to hydrogen ratio. Larger reduction is taken at the high load region where the efficiency is greater than that in diesel mode.
- The NOx emissions were reduced by 85% in average in the DF mode compared with the diesel mode. The diesel mode complies with the Tier II limits, whereas Tier II limits are met in DF operation.
- The developed heat release analysis model can be used for effectively calculating the DF combustion parameters. It is expected that such a tool can facilitate the DF engines pressure diagrams elaboration and the set-up of the engine simulation tools, which are extensively used during the engine design process.

In conclusion, it can be stated that the utilisation of natural gas, which can be stored and handled in liquefied phase (LNG) can provide an attractive and environmentally friendly alternative that should be considered and adopted in the future ship designs.

ACKNOWLEDGMENTS

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