NUMERICAL SIMULATION OF THE PERFORMANCE OF A MARINE ENGINE USING DIESEL AND BLENDS OF MARINE DIESEL WITH ETHANOL

Filipe V. Waldheim, filipevw@gmail.com
Marcelo J. Colaço, colaco@mecanica.coppe.ufrj.br
Albino J. K. Leiroz, leiroz@mecanica.coppe.ufrj.br
Federal University of Rio de Janeiro - COPPE/UFRJ
Dept. of Mechanical Engineering
Cx. Postal 68503, Rio de Janeiro, RJ, 21941-972

Abstract. *In this work we performed the numerical simulation of the working cycle of a MAN Inovator 4C engine, located at the Laboratory of Thermal Engines, Federal University of Rio de Janeiro. This engine is part of a new facility designed to test heavy fuel oils fuels and their lubricants developed by Petrobras and is the second of this kind in the world. The AVL Boost software was used for this task, resulting in cylinder pressure, power and emission curves, as well as other fundamental parameters of engine performance. The results obtained in the simulation were compared with experimental data to validate the computational model with good agreement among them.*

Keywords*: Marine Engines, Bunker, Thermodynamic simulation*

1. INTRODUCTION

Recent advances have been made toward the development of cleaner diesel engines, such as the use of common rail systems, fuel injection control strategies, exhaust gas recirculation (EGR), exhaust gas after-treatment, enriched intake air, etc (Fergunson, 1986; Heywood, 1988; Rakopoulos et al, 2007). It is worth mentioning that the U.S. Government has made a commitment to increase bio-energy threefold within approximately 10 years, which has added an impetus to the search for viable bio-fuels (Rakopoulos et al, 2007; Hansen et al, 2005). In addition, the European Union projected that biofuels would replace 5.75% of the market for gasoline and diesel by the end of 2010 (Rakopoulos et al, 2007; Hansen et al, 2005; Koerbitz, 1999). Similarly, the Brazilian Government has increased the percentage of biodiesel utilization in the diesel fuel by 5% in 2010.

Increasingly restrictive emissions legislation has promoted the search for optimizing the current engines, especially when they are to be used with different blends of biofuels. Some studies have considered the use of thermal barrier coatings, particularly in the combustion chamber, to simulate adiabatic engines. This could improve the exhaust emissions characteristics and the performance of the engines (Buyukkaya, 2008; Kojima and Nishiwaki, 1994; Prasad and Samria, 1990). More recent advances focus on the use of functionally graded materials to reduce the discontinuity in the thermo-physical properties among the different materials (Buyukkaya, 2008). Some studies have focused on the numerical optimization of the compression ratio and the start-of-combustion timing in spark-ignition engines to obtain the lowest fuel consumption accompanied with high power and low emissions (Ibrahim and Bari, 2007). The optimization of the equivalence ratio, charge pressure, charge temperature, and the start and duration of combustion has also been optimized in a natural-gas spark-ignition engine for nitrogen oxides reduction (Kesgin, 2004).

Although there are several studies focused on the minimization of the emissions in Diesel engines for road applications, marine Diesel engines still contribute largely on the emissions due to the very particular characteristics of the fuel used in such applications. Since marine engines are usually used for transportation of goods for very large distances, and consequently high temperature gradients occur along their journey, use of biofuels can be prohibited due to the fact that these fuels have their application limited to tropical temperatures. Also, the diesel fuel used in marine applications is usually heavier than the one used in road vehicles. Thus, some studies concerning the reductions of emissions in marine Diesel engines is needed.

The key factor for the optimization of current engines is the numerical simulation of the heat conduction processes in the piston and the modeling of the combustion in the cylinder. In this paper we performed a First Law analysis of the combustion process in a marine Diesel both for performance and emission assessment. Results are validated against experimental data and some analysis are conducted for blends of marine Diesel and ethanol.

2. PHYSICAL PROBLEM

The physical problem considered in this paper deals with the simulation of the combustion process in a marine Diesel engine located at the Laboratory of Thermal Engines of UFRJ. Such engine is connected to a power generator with a constant speed of 1200 rpm, and has a total power of 500 kW. The bore and stroke of each one of its 5 cylinders have 160 mm and 240 mm, respectively, and the compression ratio is equal to 15.2. The engine has three different oil carters that can be used to analyze different lubricants simultaneously. Also, different fuels can be used (such as Diesel fuel or Bunker oil) and thus an auxiliary system is connected to the engine, in order to eliminate impurities, and adjust the viscosity and temperature of the fuel before it reaches the engine. Figure 1 shows the engine as well as the auxiliary systems.

(a)

(b) (c)

Figure 1. Diesel Engine (a) and auxiliary equipments (b,c)

In order to numerically simulate the combustion process of this engine, a First Law based model was used for the processes occurring inside the combustion chamber. The gas inside the cylinder was considered a perfect gas, whose physical properties were supposed to vary with the temperature according to the well known JANAF equations (U.S. National Bureau of Standards, 1971). Also, to predict the emissions of such engine, when operating with different fuel compositions, the following two zone model was used, where the First Law was applied separately to the burned and unburned gases (Heywood, 1988):

$$\frac{dm_b u_b}{d\theta} = -P\frac{dV_b}{d\theta} + \frac{dQ_F}{d\theta} - \sum \frac{dQ_{Wb}}{d\theta} + h_u \frac{dm_b}{d\theta} - h_{BB,b}\frac{dm_{BB,b}}{d\theta} \tag{1}$$

$$\frac{dm_u u_u}{d\theta} = -P\frac{dV_u}{d\theta} - \sum \frac{dQ_{Wu}}{d\theta} - h_u \frac{dm_b}{d\theta} - h_{BB,u}\frac{dm_{BB,u}}{d\theta} \tag{2}$$

where the subscripts u and b denote the unburned and burned gases, respectively. In this equations, u is the specific internal energy, h is the enthalpy, m is the mass, P is the pressure, V is the volume, θ is the crank angle, Q_F is the heat from the fuel and Q_W is the wall heat loss. The last terms of these equations represent the enthalpy flow due to the blow-by.

From the analysis of Eqs. (1) and (2), one can see that a model for the heat release rate of the fuel is needed. One of the expressions, encountered in the literature, uses a double Wiebe's function model (Miyamoto et al, 1985; Murayama et al, 1982), given by

$$\frac{dQ_F}{d\theta} = 6.9\frac{Q_p}{\theta_p}(M_p+1)\left(\frac{\theta-\theta_{ig}}{\theta_p}\right)\exp\left[-6.9\left(\frac{\theta-\theta_{ig}}{\theta_p}\right)^{M_p+1}\right] + 6.9\frac{Q_d}{\theta_d}(M_d+1)\left(\frac{\theta-\theta_{ig}}{\theta_d}\right)\exp\left[-6.9\left(\frac{\theta-\theta_{ig}}{\theta_d}\right)^{M_d+1}\right] \tag{3}$$

where M_p is the shape factor for the premixed flame, M_d is the shape factor for the diffusive flame, Q_p is the energy released during the combustion with premixed flame, Q_d is the energy released during the combustion with diffusive flame, θ_p is the duration of the combustion with premixed flame, θ_d is the duration of the combustion with diffusive flame and θ_{ig} is the ignition angle. Such parameters have to be obtained by trial and error (as done in this work) or by some optimization method (Colaço et al, 2010).

The wall heat loss can be modeled locally by using a heat transfer coefficient h_c:

$$Q_W = h_c A\left(T - T_{gas}\right) \tag{4}$$

where T is the local temperature at the wall, T_{gas} is the temperature of the gas and A is the surface area. Several correlations for such heat transfer coefficient at the gas-walls surfaces are available in the literature (Heywood, 1988; Ramos, 1989; Borman and Nishiwaki, 1987). In this paper, we used the Woschni's model, given as:

$$h_c(\mathrm{W/m^2K}) = 3.26B(\mathrm{m})^{-0.2}P(\mathrm{kPa})^{0.8}T(\mathrm{K})^{-0.55}w(\mathrm{m/s})^{0.8} \tag{5}$$

where B is the bore of the cylinder and w is the average cylinder gas velocity, which, for a four-stroke, water-cooled, four-valve direct-injection CI engine, is expressed as:

$$w = \left[C_1\bar{S}_p + C_2\frac{V_d T_r}{P_r V_r}\left(P - P_m\right)\right] \tag{6}$$

where V_d is the displaced volume, P_r, V_r and T_r are taken at some reference state, P_m is the motored cylinder pressure at the same crank angle as P and the constants C_1 and C_2 are given as:

$$\begin{array}{ll} \text{For the gas exchange period:} & C_1 = 6.18; C_2 = 0 \\ \text{For the compression period:} & C_1 = 2.28; C_2 = 0 \\ \text{For the combustion and expansion period:} & C_1 = 2.28; C_2 = 3.24\text{x}10^{-3} \end{array} \tag{7}$$

After the models of the heat release rate, given by Eq. (3), and the wall heat losses, given by Eqs. (3)-(7), have been defined, the pressure versus crank angle curve for a closed can be obtained through the solution of Eqs. (1) and (2), by using some integration technique, such as a fourth order Runge-Kutta scheme. The above model was used as it is implemented in the AVL BOOST software, together with other parameters of the engine, as listed in Table 1.

Table 1. Engine parameters

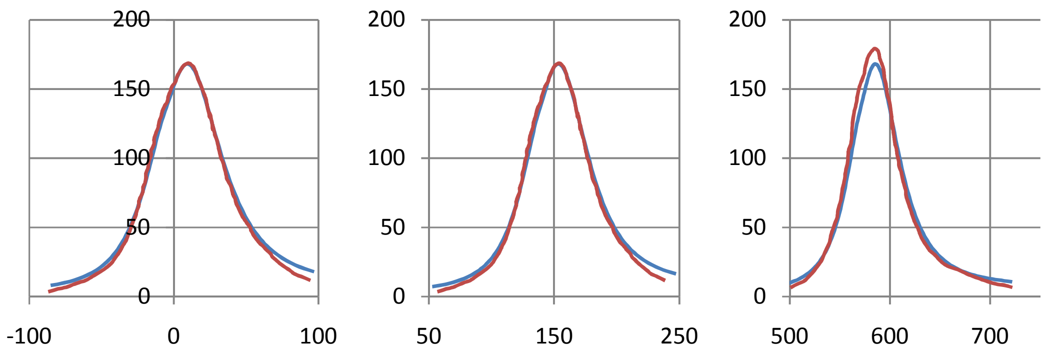
Parameter	Value
Engine manufacturer	M.A.N.
Engine model	L16-24 Stationary (Innovator 4C)
Number of cylinders	5
Valves per cylinder	4
Bore	160 mm
Stroke	240 mm
Connecting rod length	480 mm
Compression ration	15.2
Speed	1200 rpm
Firing order	1-2-4-5-3
Rated power	500 kW
Diameter of the admission valves	57 mm
Clearance of the admission valves	0.4 mm
Diameter of the exhaust valves	54 mm
Clearance of the exhaust valves	0.5 mm
Turbo charging principle	Constant pressure with intercooling
Turbo charging compression ratio	3.6

The dimensions of the inlet and outlet ducts were measured carefully and the lift valve curves were obtained by the manufacturer of the engine.

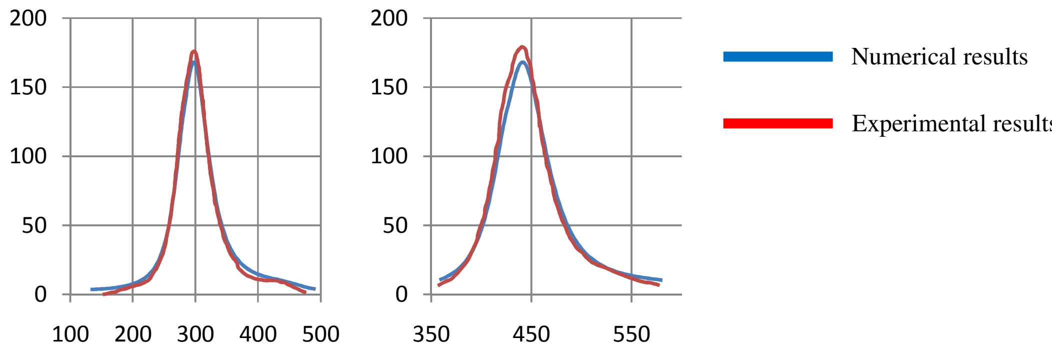
A model for prediction of emissions were also used, based on the Zeldovich mechanism and implemented in the BOOST software (AVL, 2009).

3. RESULTS

In order to validate the model presented in the previous section, a comparison was made between numerical and experimental results for the engine burning Diesel fuel. Initially, the pressure curves, obtained by a sensor located inside each one of the combustion chambers were obtained and the empirical parameters of the model were adjusted to match such pressure curves. Figure 2 shows such comparisons for the engine running at its full load, where one can see a very good agreement between the numerical and experimental results. Some deviations can be observed and are due to the fact that the model does not consider variations among the cylinders, whereas the experimental data are subjected to such variations as well as to fluctuations in the fuel and air supply.



(a) (b) (c)



(d) (e)

Figure 2. Comparison between numerical and experimental values for the pressure for (a) cylinder 1; (b) cylinder 2; (c) cylinder 3; (d) cylinder 4; (e) cylinder 5 with the engine running at its full load. The abscissas represent the crank angle and ordinates show the pressure measured in bar.

The brake mean effective pressure (bmep) and the maximum pressure per cycle were also compared against experimental data and these results are presented in Table 2. Again, the model predicts equal values for all cylinders, whereas the experimental values have some deviations. However, the average relative error is equal to 3.6% for the maximum pressure and 3.3% for the bmep, indicating a reasonable good result.

Table 2. Comparison for the brake mean effective pressure and the maximum pressure per cycle with the engine running at its full load

	P_{max} (bar) Cylinder 1	P_{max} (bar) Cylinder 2	P_{max} (bar) Cylinder 3	P_{max} (bar) Cylinder 4	P_{max} (bar) Cylinder 5	BMEP (bar)
Experimental data	168.5	168.5	180.0	176.0	180.0	20.70
Numerical data	168.2					21.39

Finally, Table 3 shows a comparison between the measured and predicted NOx emissions for the L16-24 stationary engine running at full load with Diesel fuel. One can check the excellent result obtained, with a deviation equal to 1.06% between the numerical and simulated results.

Table 3. Comparison for NOx emissions with the engine running at its full load

	NOx (g/kW.h)
Experimental data	9.4
Numerical data	9.5

Once the validity of the model was verified against experimental data, it was checked the possibility of using a blend of Diesel with ethanol, in order to take advantage of the large capacity of production of this renewable fuel in Brazil. Two different blends were tested: (i) a blend with 90% of marine Diesel fuel and 10% of ethanol; and a blend with 95% of marine Diesel fuel and 5% of ethanol. Table 4 shows the results for the maximum pressure in the combustion chamber and the emission of NOx.

Table 4. Results for blends of marine Diesel and ethanol

	P_{max} (bar)	NOx (g/kW.h)
Marine Diesel	170	9.4
95% of marine Diesel + 5% of ethanol	172.8	21.9
90% of marine Diesel + 10% of ethanol	172.2	21.3

From the analysis of Table 4, one can verify that the presence of ethanol initially increases the maximum value of the pressure. Although this needs to be further investigated, one possible cause is the presence of oxygen in the molecular structure of the ethanol that can contribute to a better combustion. When the percentage of ethanol increases, P_{max} starts do decrease since the combustion chamber was saturated by oxygen and then the lower heating value of ethanol starts to decrease the amount of heat available. One side effect of adding ethanol has to do with the larger emissions of Nitrogen oxides. Since the production of NOx is related to high temperatures and also to the presence of Oxygen, the addition of ethanol contributes to an increase in these emissions.

4. CONCLUSIONS

In this paper we performed a First Law analysis of the combustion process in a marine Diesel both for performance and emission assessment. Results were validated against experimental data with very good agreement between them, Some analysis were conducted for blends of marine Diesel and ethanol showing that some improvements on the maximum pressure of the cycle can be obtained by adding small amounts of ethanol. However, the concentration of Nitrogen oxides increase due to the presence of Oxygen and also due to the higher temperature in the combustion chamber. Further investigations are needed in order to optimize the blend both for high power and low emissions.

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