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THERMODYNAMIC MODELLING DEVELOPMENT AND CALIBRATION TO SUPPORT DIESEL ENGINES INDICATING MEASUREMENTS ANALYSIS

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Abstract: *Indicating measurements are made to support the monitoring and maintenance of outstanding engines performance. However, an analysis of indicating measurements data are not a trivial task. Hence, computational simulations are a powerful tool to support engines diagnoses from an interpretation of these data. In this paper, a zero-dimensional thermodynamic modelling is carried out to represent diesel engines processes while intake and exhaust valves are closed, i.e., compression, combustion and expansion. In addition, a graphical adjustment was made from experimental data to represent intake and exhaust strokes. This modelling took into consideration convection and radiation heat transfer and a dual Wiebe's function to model the heat release. The experimental data was collected from two cylinders of a Wärtsilä W20V32 engine, which is installed on Thermal Power Plant of Viana S.A (TEVISA), a thermal power plant located in Viana, ES, Brazil. Engines performance was evaluated from some combustion parameters and it was found that a difference of 1.8 crankangle degrees in combustion start leads to a reduction of 7.75 kW on indicated power. All processes were properly simulated through software EES (Engineering Equation Solver).*

Keywords: *Combustion analysis, Zero-dimensional modelling, Diesel engines, Indicating measurements, EES.*

1. INTRODUCTION

The power plant internal combustion engines are normally stationary and they operate in full load with constant speed. Most of them are usually large and its losses represent a significantly issue when it is related to environmental and costs concern. Then, indicating measurements are made to support the monitoring and maintenance of good engines performance. In addition, engine indicating includes the measurement of instantaneous in-cylinder pressure, the determination of the top dead centre (TDC) and the measurement of the instantaneous crank angle, which nowadays are made by sophisticated engine indicating measurement system. These measurements are fundamental for engine combustion diagnosis and for indicated work calculation (Bueno *et* al., 2009).

However, a well knowledge and analysis on the gathered data from indicating measurements are necessary to achieve the goal, which is the correct diagnosis of engine performance. In this regard, computer simulations are found to be prominent tools for imposing all possible practical conditions and to predict performance of engines.

There are several modellings used to represent the in-cylinder processes in Diesel engines. The processes can be described with either multi-dimensional models that solve numerically the equations for mass, momentum, energy and species conservation in three dimensions to predict the thermodynamic state at different position, or in the other extreme, with zero-dimensional models that assume the same average state throughout the gas. (Yasar *et* al., 2008).

Although there are various simulations models, thermodynamic zero-dimensional modellings are attractive in the light of less computational complexities involved. Thus, the simulation code used in this work employs a thoroughly validated thermodynamic, one-zone, zero-dimensional computation model i.e. the state of the cylinder charge is defined in terms of average spatial properties. Pressure, temperature and composition of cylinder charge are assumed uniform at each time step, so it means that no distinction is made between burned and unburned gas during the combustion phase inside the cylinder.

The purpose of this paper is to analyze the performance of a compression ignition engine using a computational thermodynamic model, which predicts in-cylinder's temperatures and pressures as functions of the crank angle, with the application of a dual Wiebe's function for the heat release pattern. Furthermore, convection and radiation heat losses were also considered during the modelling.

The software EES was used as the main coding platform to model the in-cylinder processes of an engine. EES is a general equation-solving program capable of dealing with thousands of coupled non-linear algebraic and differential equations (F-Chart Software, 2017).

2. METODOLOGY

In this work, all internal strokes of a direct injection Diesel engine have been studied and represented through a simplified modelling. On developed processes during the closed condition of intake and exhaust valves – compression and combustion – the zero-dimensional thermodynamic modelling was chosen to analyze the tested engine due to its less computational complexities and reasonable results. In addition to this zero-dimensional modelling, a graphical adjustment with the gathered data was used to approach curves on intake and exhaust processes.

2.1 Thermodynamic analysis: Intake and exhaust valves closed

Some assumptions are made on the strokes between the closing of intake valve and opening of exhaust valve. The properties of the mixture inside the cylinders are obtained from a thermodynamic approach considering a closed system and an ideal gas behavior.

Equation (1) represents the differential form of equation of state for ideal gases, where p, V, T and m represent pressure, volume, temperature and mass inside the cylinder, respectively. R is the universal gas constant.

$$pdV + Vdp = mRdT \tag{1}$$

On a closed system, the differential form of First Law of thermodynamic is given in Eq. (2), where W, U, and Q are work, internal energy and net heat transferred to the system due the burned fuel (Q_{fuel}) and losses on cylinder wall (Q_{loss}).

$$\delta Q - \delta W = dU \tag{2}$$

According to work and internal energy definition for ideal gas model, these equations can be written as Eq. (3) and Eq. (4), where C_v is the constant volume specific heat. Thus, it is possible to substitute in Eq. (2) and rewrite the First Law as represented in Eq. (5). Moreover, all components from Eq. (5) are divided for $d\theta$, which θ is the crankangle.

$$\delta W = pdV \tag{3}$$

$$dU = mC_v dT \tag{4}$$

$$\frac{\delta Q_{fuel}}{d\theta} - \frac{\delta Q_{loss}}{d\theta} - p\frac{dV}{d\theta} = \frac{C_v}{R}\left(p\frac{dV}{d\theta} + V\frac{dp}{d\theta}\right) \tag{5}$$

Replacing the Eq. (6) in Eq. (5), where k is the specifics heat ratio, the final differential equation form, Eq. (7), is reached out and its solution provide the pressure as a function of crankangle.

$$\frac{C_v}{R} = \frac{1}{k-1} \tag{6}$$

$$\frac{dp}{d\theta} = \frac{k-1}{V}\left(\frac{dQ_{fuel}}{d\theta} - \frac{dQ_{loss}}{d\theta}\right) - k\frac{p}{V}\frac{dV}{d\theta} \tag{7}$$

In order to determine the behavior of pressure as a function of crankangle is primary indispensable to appoint how volume, heat released by burned fuel and losses through heat transfer are solved in function of crankangle.

Heat losses through convection and radiation on cylinder walls were considered, as shown in Eq. (8), where h is convection's heat transfer coefficient, T_w is cylinder wall's temperature which is assumed constant according to Rakopoulos *et* al. (2004), σ is Stefan-Boltzmann's coefficient, β is emissivity and ω is engine speed in rad/s. Stone (1999) suggests a $\beta = 0.576$ for emissivity of Diesel Engines.

$$\frac{dQ_{loss}}{d\theta} = \frac{h\,A\,(T - T_w) + \beta\,\sigma\,(T^4 - T_w{}^4)}{\omega} \tag{8}$$

Heywood (1988) recommends an equation modelling from Woschni (1967) to determine the convection's heat transfer coefficient (h), as represented in Eq. (9), where D is cylinder's diameter and w is average cylinder gas velocity, given by Eq. (10).

$$h = 3.26\,D^{-0.2}\,p^{0.8}T^{-0.55}w^{0.8} \tag{9}$$

$$w(\theta) = 2.28\,\overline{U} + C_1\,\frac{V_d T_r}{p_r V_r}(p - p_m) \tag{10}$$

In Eq. (10), V_r, T_r and p_r are reference state properties at closing of inlet valve and p_m is the pressure at same position, obtaining p without combustion. The value of C_1 is given for compression process as C_1 = 0, and for combustion and expansion processes as C_1 = 0.00324. The average piston speed $\overline{U}$ is calculated from Eq. (11), where N is the engine speed in rpm and S is the engine stroke. V_d is the displacement volume of the cylinders.

$$\overline{U} = \frac{2\,N\,S}{60} \tag{11}$$

The rate of the heat release from fuel for diesel engines can be modeled using a dual Wiebe's function (Miyamoto *et* al., 1985) as presented in Eq. (12) and Eq. (13), where p and d refer to premixed and diffusion phases of combustion. The parameters $\Delta\theta_p$ and $\Delta\theta_d$ represent the duration of combustion in each phase. Also, $x(\theta)$ is the rate of burned fuel and x_p represents the fraction of total fuel burned in premixed phase. The ignition angle (θ_{ig}) represents the start of combustion and the constants a, m_p, m_d are selected to match experimental data. LHV is the fuel's low heating value.

$$x(\theta) = 1 - \left(x_p\,\exp\left(-a \left(\frac{\theta - \theta_{ig}}{\Delta\theta_p} \right)^{m_p + 1} \right) + (1 - x_p)\,\exp\left(-a \left(\frac{\theta - \theta_{ig}}{\Delta\theta_d} \right)^{m_d + 1} \right) \right) \tag{12}$$

$$\frac{dQ_{fuel}}{d\theta} = m\,LHV\,\frac{dx(\theta)}{d\theta} \tag{13}$$

The instantaneous cylinder volume and area are given in Eq. (14) and Eq. (15), respectively, where L is the connecting rod length and r_c is compression ratio.

$$V(\theta) = \frac{\pi\,D^2}{8}\left[2L + S - S\cos(\theta) - \sqrt{4L^2 - S^2\sin^2(\theta)} + \frac{2S}{r_c - 1} \right] \tag{14}$$

$$A(\theta) = \frac{\pi\,D}{2}\left[D + 2L + S - S\cos(\theta) - \sqrt{4L^2 - S^2\sin^2(\theta)} + \frac{2S}{r_c - 1} \right] \tag{15}$$

Once the pressure is calculated, the temperature of the gases in the cylinder can be calculated using Eq. (16).

$$\frac{1}{T}\frac{dT}{d\theta} = \frac{1}{p}\frac{dp}{d\theta} + \frac{1}{V}\frac{dV}{d\theta} \tag{16}$$

2.2 Intake and exhaust strokes: intake or exhaust valves are opened

The modelling related to strokes, while the intake and exhaust valves are opened, is normally carried out on more complex approaches, which also take account fluid dynamic equations, such as multidimensional modellings. The thermodynamics' approaches consider only the interval in which valves are closed. Because of that, some of its results present bigger errors due to the fact of not considering the effects from intake and exhaust strokes.

Thus, a graphical adjustment from experimental data was applied to represent in a simple form intake and exhaust strokes. The modelling of intake stroke was made as an expansion process with constant pressure equals the intake air's pressure of the engine. For the ending of cycle, or the interval where the exhaust valve is open, a third-degree polynomial function equation was a good approach to calibrate the experimental data, as shown in Fig. 1.

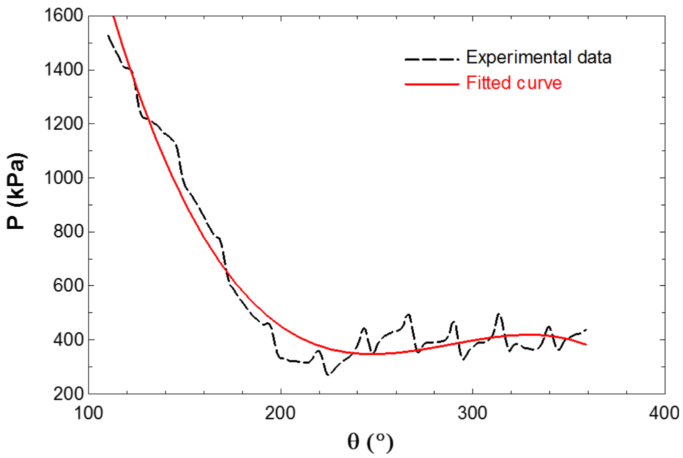


Figure 1. Adjustment of the exhaust pressure curve with a 3rd degree polynomial function.

2.3 Performance parameters

The main performance parameters, calculated after the calibrations of experimental curves and modelled curves, are the indicated power per cylinder in kW, the specific fuel consumption in g/kWh, the brake mean effective pressure in kPa and the thermal indicated efficiency. These parameters are calculated by Eq. (17), Eq. (18), Eq. (19) and Eq. (20), respectively, where $\dot{m}_f$ is fuel mass flow per cylinder in kg/s.

$$P_i = \frac{N}{120} W \tag{17}$$

$$\eta = \frac{W}{Q_{fuel}} \tag{18}$$

$$bsfc = \frac{3600000\, \dot{m}_f}{P_i} \tag{19}$$

$$BMEP = \frac{W}{V_d} \tag{20}$$

3. ENGINE DATA AND EXPERIMENTAL CONDITION

The presented methodology was applied to a heavy-duty diesel engine Wärtsilä W20V32, turbocharged and intercooled. Although this engine has twenty cylinders, the analysis will be presented for only two cylinders that presented the best and worst performance according the experimental data.

The experimental data was gathered from a combustion analyzer model Windrock DA6320, which can measure the pressure as a function of crank angle. It collects the pressure data as a function of crankangle for each cylinder at a time. For each cylinder are collected thirty cycles and the obtained result through experiments represents an average between the thirty measured values. The geometric and operating data for the engine are given in Table 1.

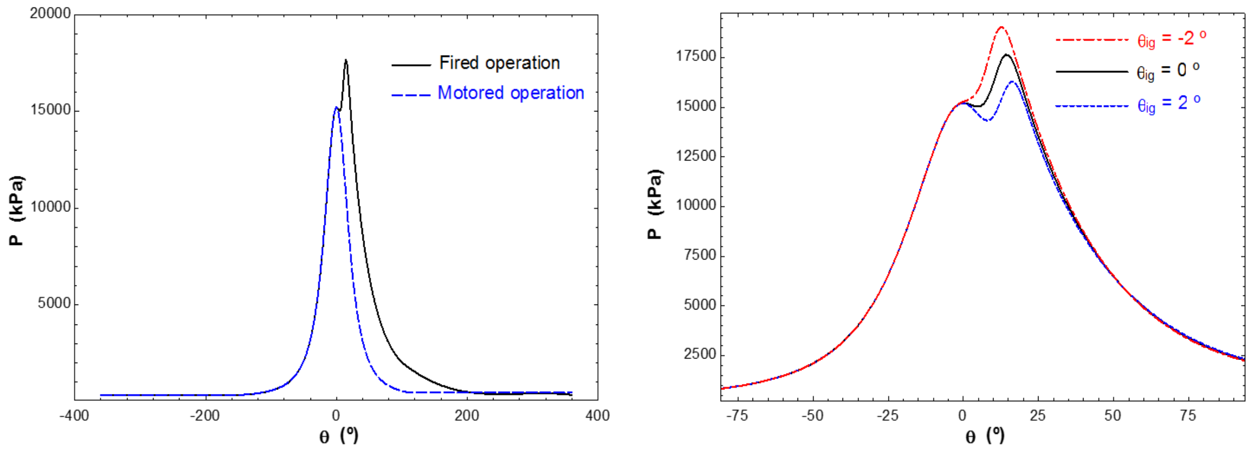
Table 1. Geometric and operating data for the engine.

Geometric and operating parameters		Value
Fuel		OCB1
Low heating value	(kJ/kg)	40785
Cylinder diameter	(m)	0.320
Piston stroke	(m)	0.400
Connecting rod length	(m)	0.960
Compression ratio		16
Engine speed	(rev/s)	12
Number of cylinders		20
Inlet pressure	(kPa)	341
Air mass flow [1]	(kg/s)	16.2
Fuel mass flow [1]	(kg/s)	0.5
Inlet temperature	(K)	328
Cylinder wall temperature	(K)	473
Inlet valve closing	(º)	-150
Exhaust valve opening	(º)	110

[1] mass flow measured per engine

4. RESULTS AND DISCUSSION

It is commonly known that the commercial combustion analyzers are used as a main tool to engine diagnosis through the interpretation of *p-θ* curve provided by these equipments. Furthermore, the format of the obtained curves are directly affected by combustion process. However, the simple analysis of the curve does not provide precise information about the several parameters which influence the combustion progress. Hence, using properly the modellings allow to know better the influence of main parameters related to combustion on engine performances and consequently it can become a powerfull support tool to a correct diagnosis. Figure 2(a) shows the influence of combustion presence and its start angle (Fig. 2(b)) on in-cylinder pressure.

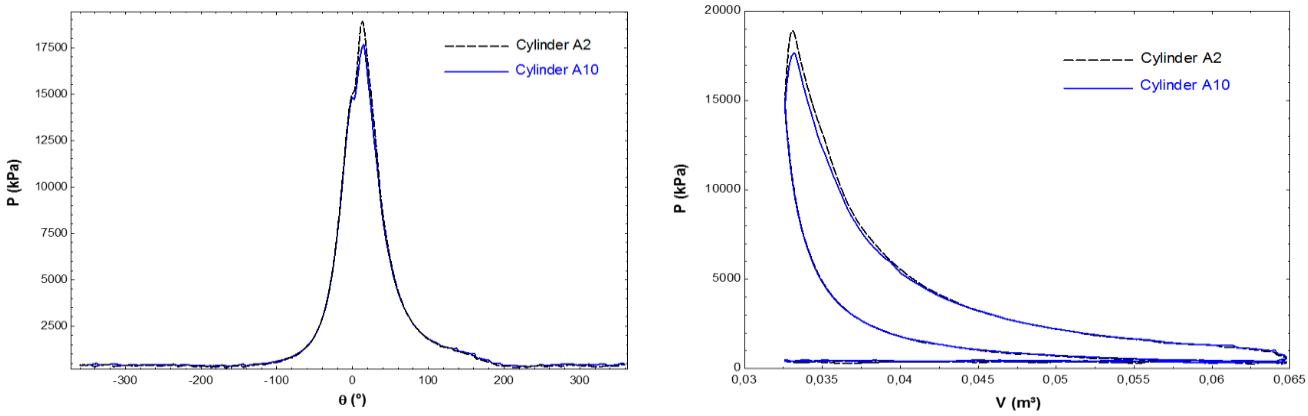


(a) (b)

Figure 2. Influence of combustion presence (a) and its start angle (b) on in-cylinder pressure.

The fuel injection timing can control the start of combustion and, as shown in Fig. 2(a), an early combustion takes higher pressures levels inside the cylinder. It is interesting to know that despite higher pressures increases the indicated power, it also increases emissions levels of NO_x (Souza Jr., 2009).

As mentioned before, although there are twenty cylinders per engine, it had been taken only experimental data from two cylinders to calibrate the modelling. In an effort to observe better the influence of combustion parameters, it was selected the best and the worst performance, cylinders A2 and A10, respectively, and they will be analyzed separately. Figure 3 shows the difference on experimental curves of both cylinders from the tested engine.



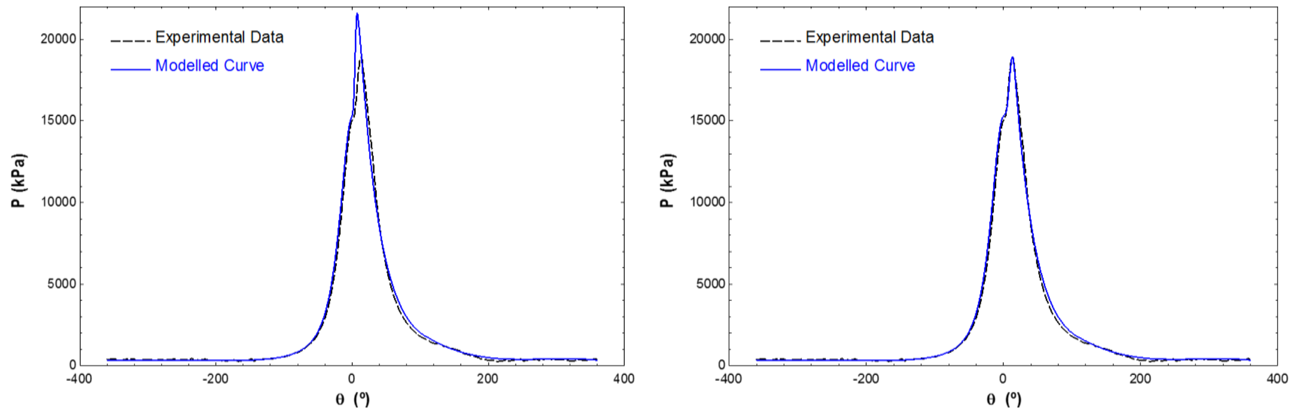
(a) (b)

Figure 3. Experimental curves of cylinders A2 (a) and A10 (b).

The calibration of thermodynamic modelling with the experimental curves must be made only according to a set of variables related to combustion, Eq. (12). Since, others input data are operating parameters and/or geometric characteristic of engine, then they will be set as fixed values.

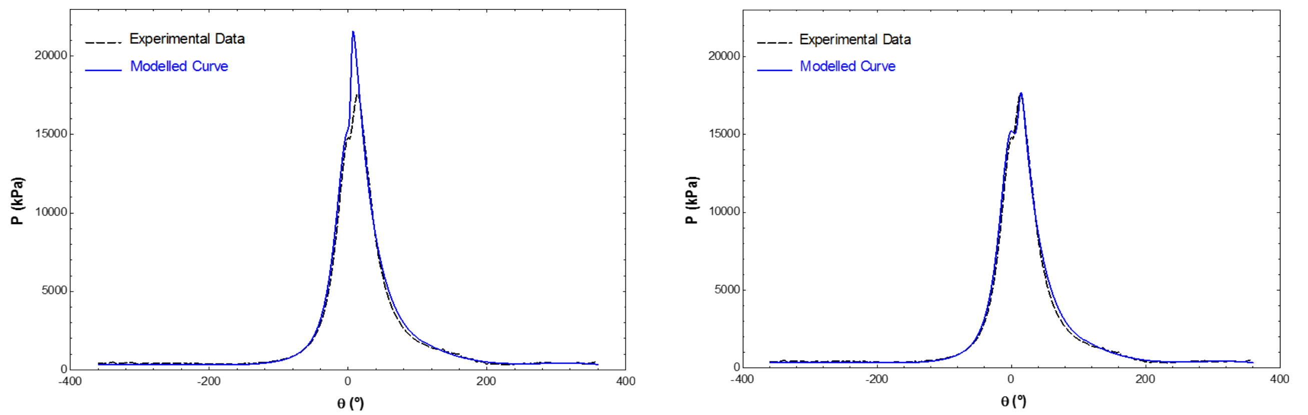
A work from Miyamoto *et* al. (1985) made an analyzes on experimental diagrams related to in-cylinder pressure of compression ignition engines and they determined, for these type of engines, that the parameters m_p and m_d remain constant and equal to 3 and 1, respectively, even if the operating conditions are changed. Besides that, they suggest values of 4.605 for a, 0.25 for x_p and approximately 10º and 90º for $\Delta\theta_p$ e $\Delta\theta_d$, respectively. These values were used as initial estimation to the curves calibration.

The Fig. 4 and Fig. 5 show the pressure curves as a function of crankangle to cylinders A2 and A10, respectively, before and after the calibration. The obtained final values for each variable, after the calibration, are shown in Table 2.



(a) (b)

Figure 4. Cylinder A2 before (a) and after (b) calibration.



(a) (b)

Figure 5. Cylinder A10 before (a) and after (b) calibration.

Table 2. Calibrating results in the modelling for the combustion variables.

Combustion variables		Cylinder A2	Cylinder A10
m_p		3	3
m_d		1	1
a		4.605	4.605
θ_{ig}	(º)	-1.8	0
$\Delta\theta_p$	(º)	18	18
$\Delta\theta_d$	(º)	85	85
x_p		0.25	0.25

As it can be seen, just the variables that are dependents on the type of engine and from combustion injection characteristics were changed (θ_{ig}, $\Delta\theta_p$ e $\Delta\theta_d$) from initial estimation guess. In addition, only the start of combustion (θ_{ig}) differed significantly the obtained curves, and hence, was the only parameter that varied between both tested cylinders.

Cylinder A2, which presented best performance, showed an early combustion at 1.8º in relation to cylinder A10, which its combustion began exactly while the piston was located on top dead centre (TDC). In the modelling, this difference between the ignition point of each cylinder generated a difference in indicated power around 7.75 kW. If some assumptions are made, such as no mechanical failure inside the cylinders that can cause this difference, an earlier fuel injection of cylinder A10 could correct this divergence. The indicated power, that had been calculated and measured for both cylinders, and its relative error are presented on Table 3.

Table 3. Calculated and measured indicated power.

		Cylinder A2		Cylinder A10	
		Experimental	Modelled	Experimental	Modelled
P_i	(kW)	516.41	504.85	488.80	497.1
Relative Error	(%)	- 2.23		1.69	

Some performance parameters could not be measured experimentally for each cylinder, separately. In fact, to obtain these parameters are normally used an average value of all cylinders from one engine. However, using the modelling it was possible to calculate these parameters, which are presented on Table 4. As expected, cylinder A2 also presented a better efficiency, *BMEP*, and less specific fuel consumption than cylinder A10.

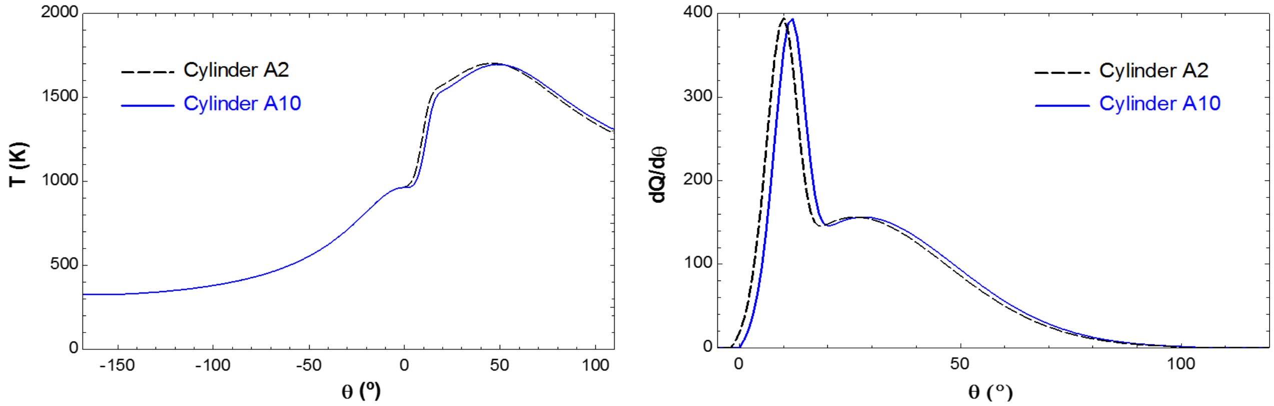
Table 4. Performance parameters.

Performance parameters		Cylinder A2	Cylinder A10
η	(%)	50.01	49.24
bsfc	(g/kWh)	178.3	181.1
BMEP	(kPa)	2610	2572

The start of combustion also affects directly the reached temperatures on combustion chamber. Higher temperatures increase the convection and radiation heat losses, and are responsible too for the emissions of NO_x, and hence, it must be avoided. The reached temperatures on both cylinders can be seen on Fig. 6-a and because of the ignition point was well approached for both cylinders, there were not a significant variation on presented temperatures.

An interesting fact that can be observed on this analysis is that when $\theta > 50$º after TDC, the gases temperature inside the cylinder A10, which presented a late combustion, reach out a greater value than the gases temperature in cylinder A2. The explanation for this behavior is because the combustion in cylinder A2 is occurring early, and its heat release rate is closing to ending, and, for cylinder A10, it was not released all of available heat yet.

Figure 6-b shows the heat release rate for both cylinders due to combustion. It is possible to note the difference in the beginning of heat release i.e. the start of combustion for each cylinder. Moreover, the two characteristics phase (premixed and diffusion) from combustion process can be observed and also it is confirmed that for $\theta > 50$º the heat release rate in cylinder A10 is bigger than A2.



(a) (b)

Figure 6. In-cylinder temperature (a) and rate of heat release (b) for both cylinders.

5. CONCLUSIONS

The thermodynamic zero-dimensional modelling that was used to simulate the compression and combustion strokes with the developed approach for intake and exhaust strokes were able to reproduce precisely the obtained experimental results. Therefore, the modelling had been shown as a powerful tool to support engines diagnoses.

The curve from modelling was validated with the experimental data of the tested engine presenting a maximum difference in indicated power of 2.23%. The methodology through adjustment on curves for intake and exhaust strokes was needful to achieve a comparison between modelling and experimental results. Then, in future works this approach will be carried out on fluid dynamics equations, such as multidimensional modellings.

Viana's S.A. thermal power plant aims to increase its power production through residual heat recovering on exhausting gases from Reciprocating Internal Combustion Engines. However, an installed heat exchanger will increase the exhaust gas backpressure that can affect directly on engine's performance. Thus, another future goal from this work will be to better understand and predict the backpressure's influence in engine's performance.

Furthermore, it is important to note that, despite this methodology was applied only for one specific engine, the modelling can be used to simulate any type of engine throughout operating parameters and geometric information.

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