HEAT RELEASE RATE OF A COMPRESSION IGNITION ENGINE FUELED WITH DIESEL OIL AND A DIESEL-STRAIGHT SOYBEAN OIL BLEND

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Abstract. *An experimental and theoretical analysis of the Heat Release Rate (HRR) of a diesel engine fueled with diesel oil and a blend of 50 % v/v of diesel oil and straight soybean oil (SVO) is presented. Fuels were tested on a dynamometer bench with a single cylinder, four strokes, naturally aspired, mechanically driven direct injection, diesel engine with nominal power of 14.7 kW at 2200 rpm. Tests were carried out at 1700, 1800 and 2100 rpm with the mechanical fuel pump open to maximum pumped volume. The mixture containing diesel and soybean oil is preheated before injection to 80 °C. The HRR was obtained from a zero-dimensional model of the combustion chamber, using measured in-cylinder pressure data, geometric parameters of diesel engine, mass of fuel injected, chemical composition of fuel and mass of air admitted to the engine. This model assumed the cylinder content as a homogeneous mixture of air and combustion products in thermodynamic equilibrium at each instant. A triple Wiebe function, characterizing the premixed, diffusive and residual combustion phases, was used to curve fit the HRR for each fuel. The results did not show significant differences between the combustion parameters found for each fuel at the tested speeds, revealing the predominant role of atomization thermo-physical properties in the engine combustion control. The model is limited to the engine and test conditions utilized here, but could be used as a first approximation for the analysis of diesel engines fueled with blends of diesel and soybean oils.*

Keywords: *compression ignition engine, straight soybean oil, diesel oil, heat release rate.*

1. INTRODUCTION

In recent decades, economics and environmental impacts associated with the use of fossil fuels have renewed the interest in the use of biofuels, such as vegetable oils and biodiesel, in compression ignition engines, both for transport and for stationary energy generation. In Brazil, several studies have focused the use of vegetable oils, especially of the crude palm oil, for electric power generation in remote regions (Almeida et al., 2002; Duarte et al., 2010; Pereira et al., 2012).

The differences in the physical-chemical properties of vegetable oils when compared to diesel oil, especially higher dynamic viscosity, higher density, lower volatility and smaller Lower Heating Value (LHV), are the main drawbacks addressed by the researchers (Agarwal et al., 2008; Franco and Nguyen, 2011). Blending with diesel oil has been studied as a simple way of reducing the viscosity of vegetable oils. Tests with vegetable oils obtained from rapeseed, palm, sunflower, cottonseed and soybean are found in the literature (Nwafor and Rice, 1996; Machacon et al., 2001; Balafoutis et al., 2011; Chalatlon et al., 2011). These tests showed a reduction in the power and increase in the specific fuel consumption when compared to neat diesel fuel. The blends with over 20% vegetable oil content presented poor combustion. Heating the fuel before injection is another way of reducing the viscosity of vegetable oils. Preheating is effective and allows the engine to operate with 100 % vegetable oil for brief periods without modifying the engine (Nwafor, 2004; Canakci et al., 2009; Agarwal and Dhar, 2010).

Blending and heating approximate the physical properties of vegetable oils or their blends to diesel oil, but there are differences in the chemical properties that can influence significantly the combustion process. In this context, this work presents an experimental analysis on the heat release rate (HRR) of the operation of a diesel engine with diesel oil and a blend of 50 % v/v of diesel oil and straight soybean oil (SVO) preheated before injection to 80 °C in order to achieve better understanding of the development of the combustion with each fuel. Based on zero-dimensional modeling of the combustion chamber, the heat release rate was obtained. A combustion model was fitted from the heat release rate, obtaining a function that describes the burning of the fuel and it could be used as a first approximation for the analysis of diesel engines fueled with blends of diesel and soybean oils. The function used to fit the combustion model was a triple Wiebe function. Results showed similarity in the combustion parameters for both fuels tested, indicating the heating was effective to close the performance of the engine with the blend to that with diesel fuel.

2. EXPERIMENTATION

2.1 Experimental set-up

The experimental setup is composed for a single cylinder, four strokes, direct injection diesel engine (Yanmar, model YT22) and 14.7 kW of power coupled to an electromagnetic dynamometer (Schenk, model W70). The dynamometric bench is equipped with a fuel supply system, a measurement system of the in-cylinder pressure, and a control and data acquisition system. An electric heater comprised of an aluminum tube (12.7 mm diameter), electric resistance of 119 Ω and ceramic insulation was manufactured to heat the blend before its entry to the engine. Thermocouples were installed at the engine cooling water outlet, the exhaust manifold, the intake manifold, the heater outlet and the injection pump inlet. An electromagnetic valve 3/2 was installed to switch the fuel employed. This enables the passage of the diesel oil from the original tank or of the fuel under test. The fuel consumption was measured using an electronic balance (Marte, model AD5000) with serial communication. The instant reading of the data allowed the calculation of the fuel flow. The engine torque was measured with an extensometer type load cell installed on the dynamometer arm. The speed was measured with an incremental encoder 360 pulses/second (Autonics, model E40S) coupled to the engine shaft. The air admitted mass was measured with the hot-film air-mass meter (Bosch, model HFM 2). The tests were controlled through an electronic control system developed with software LabVIEW 2011.

2.2 Pressure acquisition

A piezoelectric pressure sensor (Kistler, model 6041A) was used to measure the in-cylinder pressure. The sensor was installed in the cylinder head, according to technical specifications and its signal was synchronized with the incremental encoder to determinate the in-cylinder pressure for each crank angle. The measuring range of the sensor is $0 - 250$ bar, which is adequate for internal combustion engines. Pressure sensor was connected to a charge amplifier (Kistler, model 5018A) and its signal was registered through data acquisition system (National Instrument, model SCB-68).

In-cylinder pressure data were used for the analysis of the heat release rate from zero-dimensional thermodynamic model. The experimental pressure data require to be processed in order to eliminate spurious data and to obtain representative pressure data about engine combustion. The in-cylinder pressure data processing used consists of four steps: spurious data elimination, averaging many cycles, filtering and fitting of average pressure data. The data processing was developed in software Matlab.

2.3 Fuels

The fuels tested were diesel oil and a blend of 50 % v/v of diesel oil and straight soybean oil. The diesel oil was commercial Brazilian diesel oil, which has a volumetric addition of 8 % of biodiesel in accordance with national regulations. The soybean oil used was raw, that is, it had undergone no refining process. Density, dynamic viscosity, LHV and surface tension are the fuel properties that most significantly affect the performance of diesel engine with each fuel tested. These properties were measured for diesel oil and its blend, and the results are presented in Tab. 1. One can be observed the effect of heating on blend in order to close the physical properties to that diesel oil at room temperature. In relation to LHV, diesel oil presents 7 % higher LHV than the blend. The fuels were labeled as: 100D - diesel fuel and 50S/50D - blend 50 % v/v of soybean and diesel oils.

Table 1. Physical-chemical properties of the diesel oil and blend tested.

Property	100D	50S/50D	
	25 °C	25 °C	80 °C
Density (kg/m³)	829	874	836
Dynamic viscosity (mPa.s)	3.64	12.69	3.92
Surface tension (mN/m)	27.77	29.20	25.83
Lower Heating Value (kJ/kg)	42435	39566[a]	
Molecular formula	$C_{12}H_{26}$[b]	$C_{19.89}H_{39.12}O_{1.043}$[c]	

[a] Value calculated from experimental data of LHV for diesel and soybean oils.

[b] Considering the diesel oil as dodecane.

[c] It was calculated from experimental data of the elemental composition of SVO and of the composition of diesel oil as dodecane.

2.4 Experimental procedure

Before starting the test, the valves of the diesel engine are opened and the charge amplifier is reset in reference to the atmospheric pressure. The tests were performed at the maximum flow rate of the injection pump. In each test the engine operation was started with diesel fuel until the heating period was completed, that is, when the cooling water temperature reached 60 °C. The diesel oil was then replaced by the fuel under test. After stabilizing the engine, the brake process was started with the dynamometer. The load was applied in order to obtain 1700, 1800 and 2100 rpm.

For each load the engine operated until steady state, which was verified by the stabilizing of the data. Measurements of torque, speed, power output, break specific fuel consumption and air flow mass were recorded in the steady state. The measurements were registered at intervals of 5 seconds, making fifteen readings under each load condition. Additionally, in-cylinder pressure readings were recorded in the steady state during 20 seconds with a frequency of 10000 data/s.

3. HEAT RATE RELEASE

Analysis of the heat release rate is developed over compression and expansion strokes of the engine operating cycle, when valves are closed. Combustion chamber was modeled according to zero-dimensional modeling of Krieger and Borman (1966). The model assumes that combustion chamber contains a homogenous mixture of air and combustion products at each instant, i.e. no gradients of temperature, pressure and chemical concentration species in the volume. Gas mixture is considered as perfect gases. The combustion is assumed as a uniform heat source. Non-equilibrium compositions, fuel vaporization and mixing of air and fuel are neglected.

Control volume is shown in Fig. 1. When the intake and exhaust valves are closed, the only mass flow across the system boundary is the fuel injected, assuming there are no leaks through gaps in the cylinder. The energy conservation equation and the mass conservation equation for the control volume are presented in the Eq. (1) and Eq. (2).

$$\frac{d(mu)}{dt} = \frac{dQ}{dt} - p\frac{dV}{dt} + h_f\frac{dm_f}{dt} \tag{1}$$

$$\frac{dm}{dt} = \frac{dm_f}{dt} \tag{2}$$

where m is the mass in the combustion chamber, u is the internal energy of the gas mixture, dQ/dt is the rate of heat transfer from the gas across the system boundary, pdV/dt is the rate of work transfer done by the system, h_f is the fuel enthalpy, and dm_f/dt is the fuel burning rate. Though the mass in the cylinder only varies because of the fuel injection, as the single combustion model assumes that the fuel burns instantaneously as it enters the combustion chamber, and the thermodynamics states of the working fluid are affected by the release of the chemical energy of the fuel, the term dm_f/dt can be considered as the mass burning rate of the fuel.
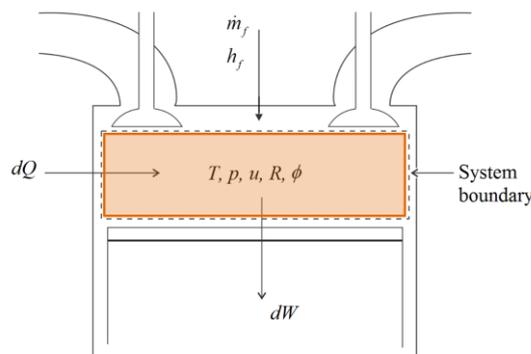

Figure 1. Control volume defined by the combustion chamber.

The equivalence ratio ϕ at each instant is given by

$$\frac{d\phi}{dt} = \frac{1}{m_a(FA)_s}\frac{dm_f}{dt} \tag{3}$$

where m_a is the air mass in the cylinder and $(FA)_s$ is the stoichiometric fuel-air ratio. The equation of state can be written in differential form as

$$\frac{1}{V}\frac{dV}{dt} + \frac{1}{p}\frac{dp}{dt} = \frac{1}{T}\frac{dT}{dt} + \frac{1}{R_g}\frac{dR_g}{dt} + \frac{1}{m}\frac{dm}{dt} \tag{4}$$

Since the properties of the gases in the cylinder are function of T, p e ϕ, the internal energy u and the gas constant R_g can be expressed as,

$$\frac{du}{dt} = \frac{\partial u}{\partial T}\frac{dT}{dt} + \frac{\partial u}{\partial p}\frac{dp}{dt} + \frac{\partial u}{\partial \phi}\frac{d\phi}{dt} \tag{5}$$

$$\frac{dR_g}{dt} = \frac{\partial R_g}{\partial T}\frac{dT}{dt} + \frac{\partial R_g}{\partial p}\frac{dp}{dt} + \frac{\partial R_g}{\partial \phi}\frac{d\phi}{dt} \tag{6}$$

Equations for temperature variation in function of time and fuel burning rate were obtained from Eq. (1) to Eq. (6). These equations are shown in Eq. (7) and Eq. (8).

$$\frac{dT}{dt} = \frac{\left[\frac{1}{V}\frac{dV}{dt} + \left(\frac{1}{p} - \frac{1}{R_g}\frac{\partial R_g}{\partial p}\right)\frac{dp}{dt} - \left(\frac{1}{R_g m_a (FA)_s}\frac{\partial R_g}{\partial \phi} + \frac{1}{m}\right)\frac{dm_f}{dt}\right]}{\frac{1}{T} + \frac{1}{R_g}\frac{\partial R_g}{\partial T}} \tag{7}$$

$$\frac{dm_f}{dt} = \frac{\frac{dQ}{dt} - \left(\frac{m}{V}B + p\right)\frac{dV}{dt} - \left[mB\left(\frac{1}{p} - \frac{1}{R_g}\frac{\partial R_g}{\partial p}\right) + m\frac{\partial u}{\partial p}\right]\frac{dp}{dt}}{u - h_f + \frac{m}{m_a (FA)_s}\frac{\partial u}{\partial \phi} - B - \frac{mB}{R_g m_a (FA)_s}\frac{\partial R_g}{\partial \phi}} \tag{8}$$

where $B = \frac{\partial u}{\partial T}\left(\frac{1}{T} + \frac{1}{R_g}\frac{\partial R_g}{\partial T}\right)^{-1}$.

The volume variation was calculated from geometrical relations of the engine. The values of pressure and its derivate were obtained from experimental data with the piezoelectric pressure sensor. The heat transfer rate was calculated through Eq. (9).

$$\frac{dQ}{dt} = h_c A(T - T_w) + \varepsilon \sigma A(T^4 - T_w^4) \tag{9}$$

where A is the area, h_c is the heat transfer coefficient, σ is the Stefan Boltzmann constant, $\sigma = 5.67 \times 10^{-8}$ W / (m²K⁴), T is the mean gas temperature, T_w is the wall temperature and ε can be considered a global emissivity because the temperature used is the mean gas temperature. The heat transfer coefficient can be calculated from correlations found in the literature. In this work was used the Hohenberg correlation shown in the Eq. (10).

$$h_c = 130 V^{-0.06} p^{0.8} T^{-0.4} \left(\overline{S}_p + 1.4\right)^{0.8} \tag{10}$$

where V is the cylinder volume and $\overline{S}_p$ is the mean piston speed. Equations (7) and (8) can be solved numerically to obtain the mean gas temperature and the fuel mass burned. The heat release rate is calculated as,

$$\frac{dQ_f}{dt} = PCI \frac{dm_f}{dt} \tag{11}$$

The thermodynamic properties and their partial derivatives related to the temperature, pressure and equivalence ratio were determined by using the routines proposed by Olikara and Borman (1975). The fuel enthalpy at injection conditions was calculated from data found in the literature for dodecane and SVO.

3.1 Combustion model

The fuel burning rate was used to fit a combustion model. A common model found in the literature is the Wiebe function. This model is an empirical correlation based on the concept of chain reactions that expresses the mass fraction burned X_b in function of characteristic parameters of each combustion phase. In this work, triple Wiebe function was used, representing the premixed, mixing-controlled (diffusive) and residual combustion phases. The fuel burning rate is related to the Wiebe function as,

$$\frac{dm_f}{dt} = m_f \frac{dX_b}{dt} \tag{12}$$

$$\frac{dX_b}{dt} = \left[\begin{array}{l} \dfrac{a X_p (m_p+1)}{\Delta\theta_p}\left(\dfrac{\theta-\theta_{ig}}{\Delta\theta_p}\right)^{m_p}\left\{\exp\left[-a\left(\dfrac{\theta-\theta_{ig}}{\Delta\theta_p}\right)^{m_p+1}\right]\right\} + \dfrac{a X_d (m_d+1)}{\Delta\theta_d}\left(\dfrac{\theta-\theta_{ig}}{\Delta\theta_d}\right)^{m_d}\left\{\exp\left[-a\left(\dfrac{\theta-\theta_{ig}}{\Delta\theta_d}\right)^{m_d+1}\right]\right\} \\[4mm] + \dfrac{a X_r (m_r+1)}{\Delta\theta_r}\left(\dfrac{\theta-\theta_{ig}}{\Delta\theta_r}\right)^{m_d}\left\{\exp\left[-a\left(\dfrac{\theta-\theta_{ig}}{\Delta\theta_r}\right)^{m_r+1}\right]\right\} \end{array} \right] w\frac{180}{\pi} \tag{13}$$

where m_f is the fuel injected per cycle, dX_b/dt is the rate of mass fraction burned, a is a constant of a Wiebe function, X_p, X_d and X_r represent the mass fraction burned in the pre-mixed, diffusive and residual combustion phases, respectively. Similarly, each combustion phase has a factor m and a duration period $\Delta\theta_p$, $\Delta\theta_d$ and $\Delta\theta_r$. θ_{ig} is the angle of the start of the ignition and w is the angular velocity.

4. RESULTS AND DISCUSSION

Combustion parameters for both fuels tested are presented in Tab. 2. Figure 2 shows the results of pressure in the combustion chamber for diesel oil and its blend at 1700 and 2100 rpm. One can be observed that diesel oil presented higher pressure than blend for all speeds tested, which is related to the LHV. As observed in Tab. 2, the peak pressure for diesel oil was approximately 4 % higher at 1700 rpm and 5.5 % higher at other speeds than that blend. Considering the performance of mechanical injection system of the engine, the fuel mass injected per cycle increases with the decreasing of the speed, for that reason, the values for peak pressure were higher for 1700 rpm than for 2100 rpm for both fuels tested.

Table 2. Combustion parameters of diesel oil and its blend at three speeds tested.

Fuel and parameters	Start of injection (°)	Ignition delay (°)	Peak pressure (kPa)	Angle peak pressure (°)	Duration of pre-mixed combustion phase, $\Delta\theta_p$	Mass fraction burned, X_p	Mass fraction burned, $X_d + X_r$ [a]
100D							
1700 rpm	346	7	7333	369	9	0.10	0.69
1800 rpm	346	8	7111	369	9	0.12	0.68
2100 rpm	346	9	6736	368.5	9	0.11	0.67
50S/50D							
1700 rpm	346	8	7062	369	9	0.12	0.68
1800 rpm	346	9	6739	368.5	9	0.13	0.66
2100 rpm	347	8	6390	368	10	0.13	0.66

[a] The term $X_d + X_r$ represents the mass fraction burned at the opening of the exhaust valves, involving mixing-controlled and residual combustion phases.
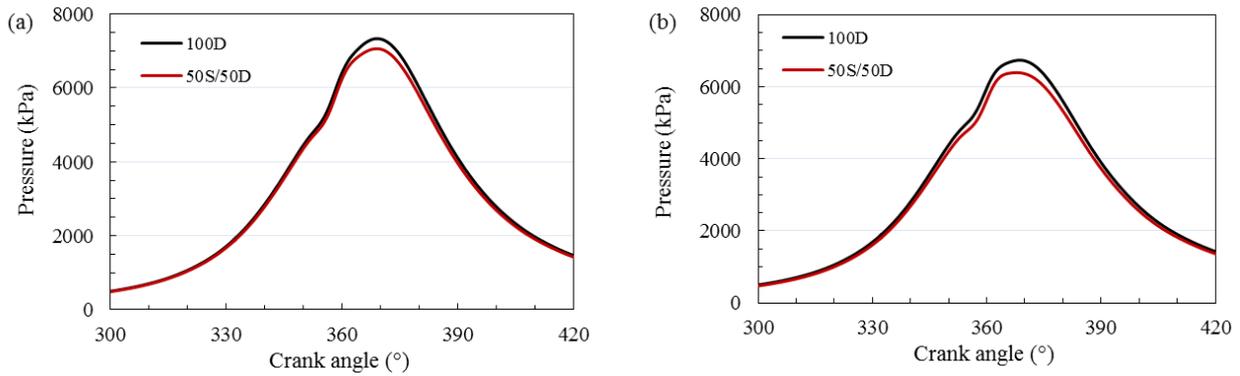
Figure 2. Pressure as a function of crank angle for diesel oil and blend at (a) 1700 rpm, and (b) 2100 rpm.

The ignition delay period is defined as the time between the start of fuel injection into the cylinder and the ignition of the fuel. The crank angle of the start of injection and of the start of ignition was calculated by differentiating of the experimental data of in-cylinder pressure with respect to time. These angles can be identified as the maximum and minimum points in the first derivative of pressure before the top center dead or second derivative of cylinder pressure becomes zero.

The heat release rate for both fuels at 1700, 1800 and 2100 rpm are shown in Fig. 3. It was found a similar ignition delay for both fuels at all conditions tested. The mass fraction burned and the duration of each combustion phase are presented in Tab. 2. The mass fraction burned in the pre-mixed combustion phase was high for the blend. Conversely, mass fraction burned in the mixing-controlled combustion phase was high for diesel oil. Similar behavior in the heat release rate can be explained by the effect of heating on the blend, which closed the physical properties of the blend to that diesel oil.
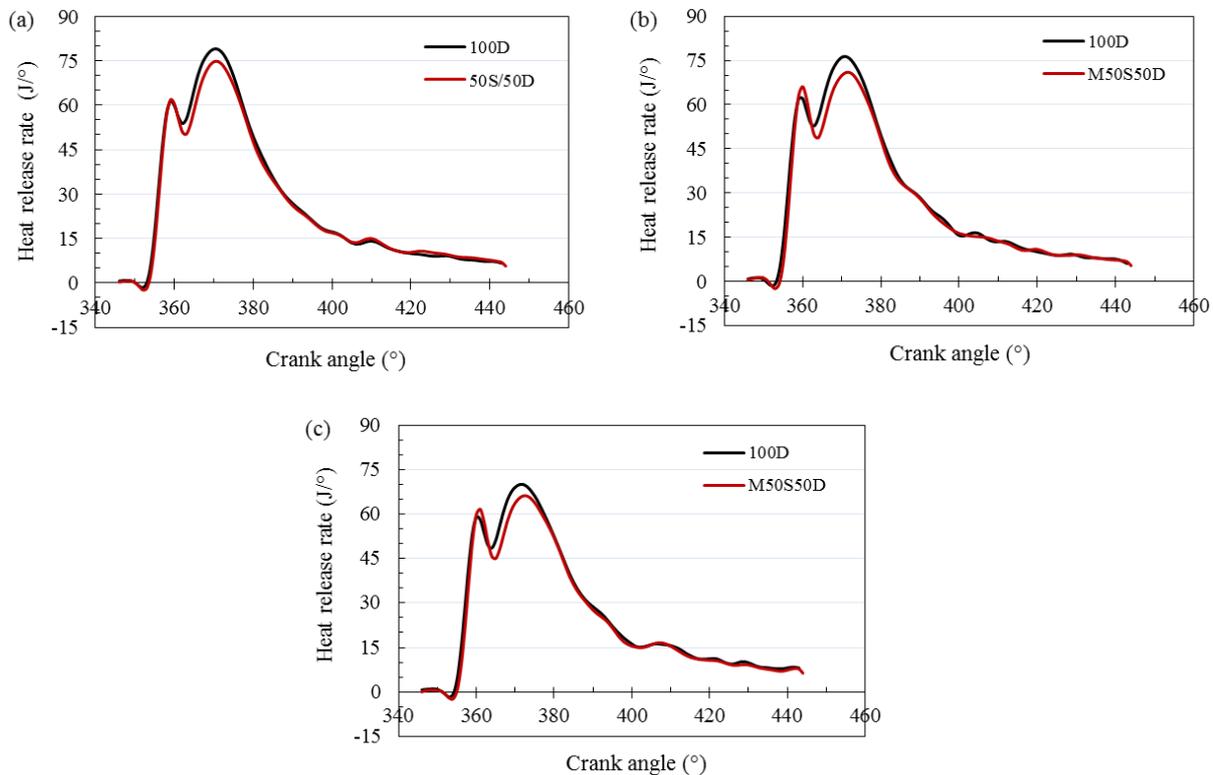

Figure 3. Heat release rate as a function of crank angle for diesel oil and blend at (a) 1700 rpm, (b) 1800 rpm, and (c) 2100 rpm.

The opening of the exhaust valves occurs at 444° of crank angle. Therefore, the curves presented in Fig. 3 correspond to close phase of the cycle. One can be observed that the fuel did not burn completely until the opening of the exhaust valves, prolonging the burning period during the exhaust stroke. Considering the mass fractions burned presented in Tab. 2, the fuel mass burned at the beginning the exhaust stroke was only 79 %. This behavior indicated an inadequate atomization of diesel engine tested. Consequently, the combustion model used to fit the heat release rate was

a triple Wiebe function with the objective to identify the residual combustion phase in the model. Table 3 shows the coefficients of the nonlinear regression model used and the respective coefficient of determination R^2, which presents values higher than 0.9, indicating the accuracy of the fit. The fitting was developed by the Levenberg-Marquardt method using software Matlab.

Table 3. Coefficients of the triple Wiebe function for the diesel oil and its blend at three speeds tested.

Fuel and parameters	a	X_p	X_d	X_r	$\Delta\theta_p$	$\Delta\theta_d$	$\Delta\theta_r$	m_p	m_d	m_r	R^2
100D											
1700 rpm	2.5	0.10	0.38	0.34	9.0	29.0	79.9	0.9	1.8	1.2	0.95
1800 rpm	2.2	0.12	0.35	0.35	9.0	26.9	73.8	0.8	1.9	1.1	0.97
2100 rpm	2.3	0.11	0.31	0.39	9.0	27.0	76.2	0.8	1.9	1.0	0.96
Mean value	**2.3**	**0.11**	**0.35**	**0.36**	**9.0**	**27.6**	**76.6**	**0.8**	**1.9**	**1.1**	
50S/50D											
1700 rpm	2.4	0.12	0.33	0.38	9.0	27.0	78.5	0.8	1.9	1.0	0.96
1800 rpm	2.4	0.13	0.29	0.39	9.0	26.0	75.0	0.6	2.0	1.0	0.95
2100 rpm	2.6	0.13	0.31	0.37	10.0	29.0	73.1	0.8	2.0	1.2	0.94
Mean value	**2.5**	**0.13**	**0.31**	**0.38**	**9.3**	**27.3**	**75.6**	**0.7**	**2.0**	**1.1**	

Table 3 also presents the mean value calculated for each parameter. Considering that the standard derivation of mean value calculated is less than 10 %, the mean values were considered as the parameters of the combustion model fitted for each fuel, for the range of speed tested. The validity of the combustion model was tested, modeling the complete cycle of the engine fueled with 100D oil and the blend 50S/50D. The simulation of the cycle replicated the heat release rate and the pressure values in the cylinder. The results were compared to experimental data as shown in Fig. 4 for heat release rate at 1800 rpm. The coefficient of determination R^2 of the simulated curves of heat release rate in relation to the calculated curves from experimental data of the cylinder pressure was 0.97, which indicates that the combustion model fitted presents an adequate accuracy between the experimental and simulated data. As a result, the combustion model can be used to predict the engine performance when the injection parameters are changed such as start of injection or injection duration. Additionally, based on the combustion model, a parametric analysis can be developed to study the influence of the combustion parameters on engine performance tested.
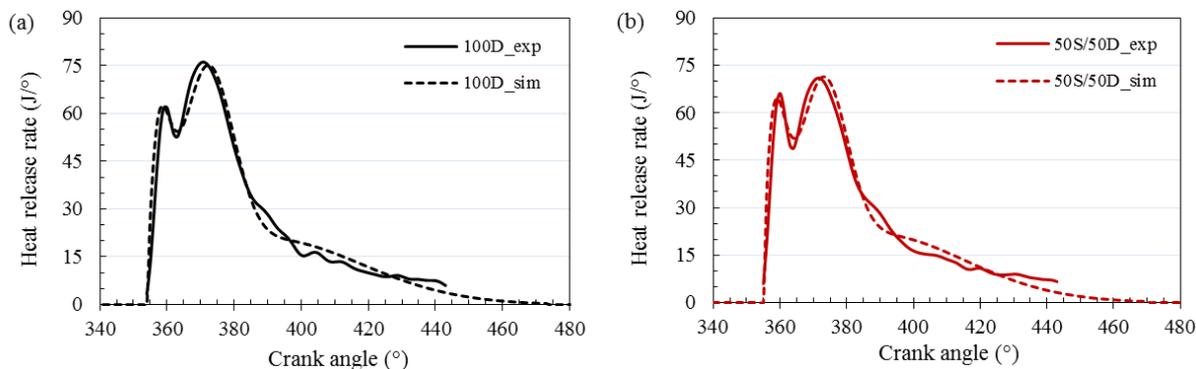

Figure 4. Comparison of experimental heat release rate and simulated results as a function of crank angle at 1800 rpm for (a) diesel oil 100D, and (b) blend 50S/50D.

5. CONCLUSIONS

The heat release rate of a single cylinder, naturally aspired, mechanically pumped and controlled, direct injection, compression ignition engine, operating with diesel oil and a blend 50 % v/v of soybean and diesel oils, was assessed at 1700, 1800 e 2100 rpm. The blend was preheated before injection to 80 °C in order to approximate the physical properties of the blend to that diesel oil.

A piezoelectric pressure sensor was used to measure the in-cylinder pressure. Combustion chamber was modeled according to zero-dimensional modeling, using the experimental data of the pressure in the combustion chamber to calculate the heat release rate for each fuel tested. Experimental results showed similar combustion parameters for both fuels at three speeds. In relation to peak pressure, the diesel oil presented higher pressure than blend for all speeds, which is related to the LHV.

It was found that the engine tested does not atomize correctly, such that the fuel mass burned was only 79 % at the beginning the exhaust stroke. This behavior indicated that the combustion could be modeled as a triple Wiebe function,

where a residual combustion phase is identified. The results of the heat release rate were fitted to a triple Wiebe function in order to obtain the combustion model for each fuel. Simulations of the complete cycle showed accuracy between the experimental and simulated data. The results did not present significant differences between the combustion parameters found for each fuel at the tested speeds, revealing the predominant role of atomization thermo-physical properties in the engine combustion control. Although these results have been obtained for a specific engine and operation conditions, the combustion model obtained can be used as a first approximation for the analysis of diesel engines fueled with blends of diesel and soybean oils, including the analysis of the influence of the injection and combustion parameters.

6. ACKNOWLEDGEMENTS

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