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COMBUSTION IN A DUAL FUEL COMPRESSION IGNITION ENGINE DIESEL/ETHANOL: ANALYSIS OF THE WORK, HEAT RELEASE AND DURATION

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Abstract. *Nowadays, several renewable fuels have been tested as substitutes for diesel oil in compression ignition engines, ethanol being one of the best alternatives. In this regard, it is necessary to evaluate the performance, emissions and combustion parameters of these engines. Thus, the objective of this research was to evaluate the work produced, the accumulated heat release and the duration of the combustion in a dual-fuel compression ignition engine operating with diesel oil direct injected and ethanol by fumigation method. The engine used originally operated with diesel oil only and was adapted for operation with ethanol fumigation. By measuring the pressure inside the engine cylinder, it was possible to perform the post-processing of these data and obtain the values of the parameters that aimed the search. The results showed that the operation of the engine by the fumigation method was satisfactory, allowing a wide range of substitution of diesel oil for ethanol, even above that recommended by the literature. The total duration of combustion was reduced as well as the start of combustion was delayed, which may contribute to better engine performance compared to the original operating condition with diesel oil only.*

Keywords: *ethanol fumigation; compression ignition engine; thermodynamic cycle; heat release.*

1. INTRODUCTION

The research and development in the field of internal combustion engines was intensified from the moment when the environmental issue began to be observed as relevant by the humanity. During the 1970s, the first rules for limiting gaseous emissions in engines began to emerge, and the techniques used to manufacture, operate and control engines were no longer enough to meet the new standards. Therefore, new technologies have been developed. One of the points of this development is the replacement of fossil fuels with renewable fuels. This research topic is growing worldwide.

In this regard, the use of alternative fuels to replace fossils fuels can directly contribute to the improvement of the performance and reduce emissions of internal combustion engines. A viable alternative in some countries is the use of ethanol, which emerged in Brazil as a result of Proálcool program, created by the federal government to encourage the production of this fuel (Tolmasquim et al., 2007). The use of ethanol alternative proved to be so viable that even today most of the vehicles (flex) are manufactured with specific characteristics to operate simultaneously with gasoline and ethanol. In addition, due to its renewable origin, ethanol contributes to the reduction of the use of fossil fuels and consequently of greenhouse gases.

Thus, the objective of this research was to evaluate the work produced, the accumulated heat released and the duration of the combustion in a dual-fuel compression ignition engine operating with diesel oil, by direct injection, and ethanol, by fumigation method.

1.1 Ethanol fumigation in a compression ignition engine

Munsin et al. (2015) cite that ethanol can be used as a fuel not only in spark ignition engines, as it occurs in its current wide application, but also in compression ignition engines, which can achieve higher efficiencies. The possible methods for combining the use of ethanol and diesel in compression ignition engines are diverse. However, the most common, according to the authors (Imran et al., 2013, Morsy, 2015, Mariasiu et al., 2015, Hansdah and Murugan, 2014) are:

1. Ethanol and diesel mixtures, that is, ethanol mixed directly with diesel oil.
2. Double injection with two direct injection systems.
3. Fumigation (injection) of ethanol on admission.

The use of ethanol by the blends (1) or double injection (2) methods is limited due to low density, low viscosity and insufficient cetane number (Chauhan et al., 2011), limiting the ethanol mixed in diesel about 15% (Sandalci et al., 2014). To use ethanol in diesel engines, typically the fumigation method is used in the intake manifold (Wei et al., 2015; Ferreira et al., 2013).

Fumigation is the term used to denote the ethanol injection system at the engine intake manifold, allowing it to admit a pre-mixed charge of air and fuel, as occurs in spark ignition engines (Munsin et al., 2015). Chauhan et al., 2011, and Sandalci et al., 2014, mention that of the methods used for the use of ethanol in diesel engines, fumigation is the most studied, because it allows higher substitution rate of diesel for ethanol in terms of energy (up to 50%) and because it requires fewer changes in the original physical structure of the engine.

1.2 The combustion in dual fuel compression ignition engine diesel oil/ethanol

Karim (2003) reported that there are three distinct phases in the pilot-ignition in dual-fuel combustion. The first stage involves the combustion of a little pilot fuel and some premixed fuel that has been drawn into the spray. The second step involves the combustion of remaining pilot fuel and premixed fuel in its vicinity. The third stage involves the propagation of the flame by the remainder of the premixed charge. However, for Goldsworthy (2013) fumigation in a compression ignition engine in which most of the energy comes from standard diesel fuel, combustion processes may differ from this standard. Even so, the author cites that the three models described by Karim (2003) can be used as a guide for the understanding of combustion.

Pre-injection of a low reactivity fuel into the intake manifold in a compression ignition engine causes the combustion process to have different characteristics over conventional operation with fuel injected only directly into the combustion chamber. The duration of the phases described previously can have significant modifications, with beginning and end displaced, that is, delayed or advanced. Characteristics of dual-fuel combustion in compression ignition engines are reported in the literature, as discussed in Tab. 1.

Table 1. Combustion phases on dual fuel compression ignition engine ethanol/diesel oil (by fumigation).

Combustion phase	Characteristics
Ignition delay	The reduction of the temperature of the allowed load, due to the vaporization of the ethanol during the times of admission and compression, tends to cause an increase in the ignition delay (Geo et al., 2017).
	The low cetane number of ethanol tends to delay the begin of combustion (Babiker et al., 2016).
Premixed combustion	The presence of a pre-mixed charge gives the occurrence of a flame front, as in the case of spark ignition engines, so the proportion of fuel burned increase in the pre-mixed phase (Tsang et al., 2010).
Diffusive combustion	There is a reduction in the duration of this phase, because a larger portion of the fuel is burned in the pre-mixed phase (Jamuwa et al., 2017).
Late combustion	-

As a result of the use of ethanol fumigation in a compression-ignition engine, Jamuwa et al. (2017) reports that there is a reduction in the combustion duration. In the heat release curve obtained by Nour et al. (2017), the peak of heat release was reduced as a function of the addition of ethanol by fumigation, and they reported also the displacement of this peak and the onset of the heat release later than the original position.

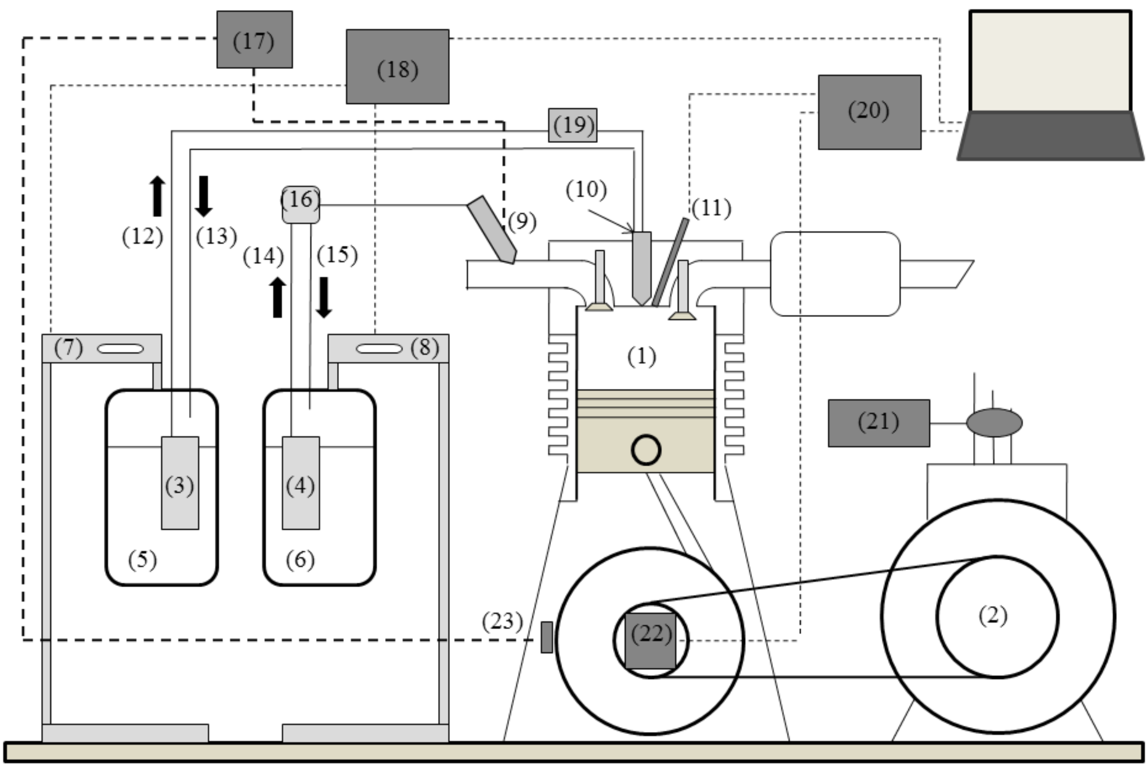
2. MATERIALS AND METHODS

2.1 Experimental setup, instrumentation and experimental procedure

The engine and alternator specifications used in the tests are shown in Tab. 2. Fig. 1 shows the experimental setup prepared for performing the tests.

Table 2. Engine and alternator technical data (Agrale, 2016).

Characteristic	Specification	Unit
Manufacturer and model (Engine)	Agrale - M93ID	-
Operation principle	Diesel 4 tempos	-
Displacement	668	cm³
Power at 2500 rpm	12/8.8	CV/kW
Cylinder number	1	-
Compression ratio (CR)	21:1	[-]
Injection point (initial)	19º	BTDC
Cooling system	Forced air	-
Manufacturer and model (Alternator)	L132 – Kohlbach	-
Power	10/8	kVA/kW
Three-phase voltage	220	Volts



(1) compression ignition engine; (2) alternator; (3) diesel oil pump; (4) ethanol pump; (5) diesel oil tank; (6) ethanol tank; (7) diesel oil load cell; (8) ethanol load cell; (9) ethanol nozzle; (10) diesel oil nozzle; (11) pressure transducer; (12) diesel oil supply line; (13) diesel oil return line; (14) ethanol supply line; (15) ethanol return line; (16) pressure regulator valve; (17) electronic control unit; (18) Arduino micricontroller; (19) injector pump; (20) pressure transducer data acquisition; (21) voltmeter and amperimeter; (22) encoder; (23) hall sensor.

Figure 1. Experimental setup.

During all tests the engine speed was maintained at approximately 2560 rpm, required for the alternator to produce energy at the 60 Hz, as well as the load was kept constant at approximately 7.2 kW (alternator electric charge). For determination of the load power, the electric current consumed using an ammeter and the voltage produced by the alternator were measured using a multimeter, for which the values were maintained at approximately 19.1 A and 218 V during all tests.

The consumption measurements of ethanol and diesel oil in the reservoirs were carried out using two spl-type load cells, with a capacity of up to 5 kg, in which the respective reservoirs were suspended. For data collection of each load cell an Arduino UNO R3 microcontroller was used. Fig. 2 shows the balance assembly.

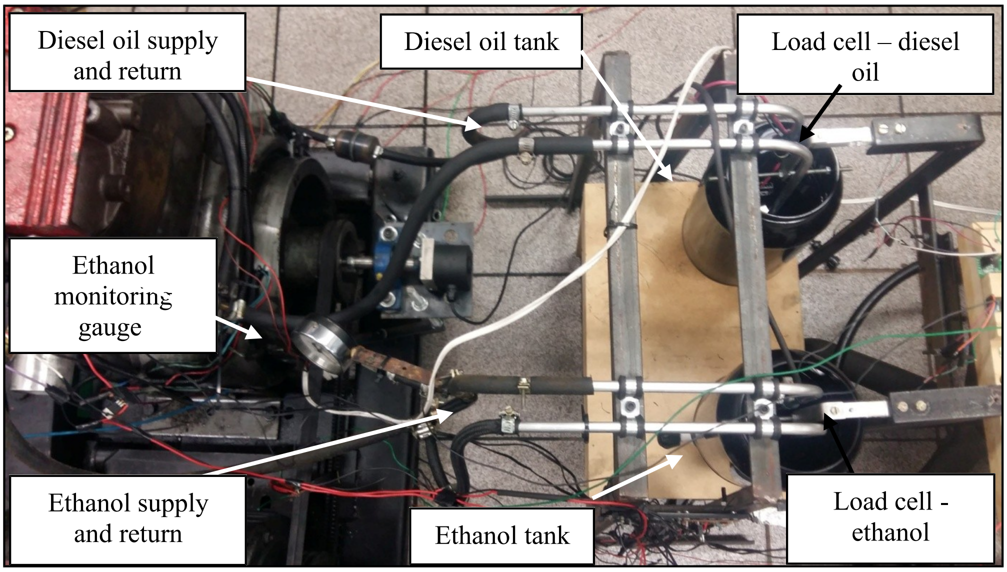


Figure 2. Experimental setup.

The pressure in the ethanol supply line remained constant at 5 bar throughout tests, which was monitored through an analog manometer installed in the ethanol supply line to the injector nozzle. The diesel oil line was kept pressurized with low pressure (not measured) just to maintain constant flow in the return line.

To measure the pressure inside the cylinder a resistive piezo type sensor, manufacturer Kulite, model ECWT-312 was used. To acquire the crankshaft angle an incremental encoder, manufacturer S & E instruments, model E30A1A-360-PP, with resolution of 1° was used. The acquisition of pressure and angle data was performed by a specific dsPIC30f401 microcontroller for signal processing, which operates at a frequency of 100 MHz, the board being connected to the computer via RS232 serial port.

For the execution of the tests it was initiated by the operation without load during 15 minutes, with load for another 30 minutes for heating and later recording of the data for the test. After the test the tests were started with fumigation of ethanol. The test time in each condition was 3 minutes, and at the end of the tests the engine was run on diesel oil for 15 minutes with load and finally for 15 minutes with no load for cooling.

2.2 Combustion parameters analysis

Based on the pressure curve inside the cylinder, it is possible to derive more information about the combustion process and thus evaluate its behavior by analyzing the heat release rate, the burned fuel mass fraction, the determination of the start and the end of the combustion process. The flowchart of Fig. 3 shows the steps of the post processing of combustion data.

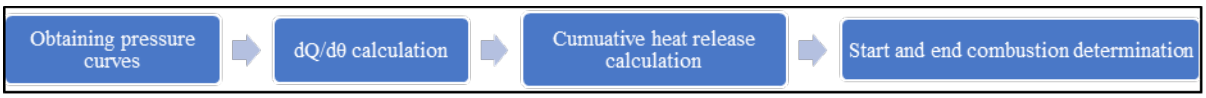


Figure 3. Steps of data processing after combustion.

To obtain the heat release curve, or heat release rate as a function of crankshaft angle (dQ / dθ), we used Eq. (1) (Heywood, 1987).

$$\frac{dQ}{d\theta} = \frac{1}{k-1}\left[kP\frac{dV}{d\theta} + V\frac{dP}{d\theta}\right] \quad (1)$$

In Eq. (1), d is the term representing the differential, Q is the released heat, θ is the crankshaft angle, V is p instantaneous cylinder volume, P is the pressure inside the cylinder, and k is the polytropic constant. As the encoder resolution used was 1°, Eq. (1) was analytically solved with dθ equal to 1°, obtaining the form of Eq. (2).

$$\frac{dQ}{\theta_i - \theta_{i-1}} = \frac{1}{k-1}\left[k\left(\frac{P_i + P_{i-1}}{2}\right)\left(\frac{V_i - V_{i-1}}{\theta_i - \theta_{i-1}}\right) + \left(\frac{V_i + V_{i-1}}{2}\right)\left(\frac{P_i - P_{i-1}}{\theta_i - \theta_{i-1}}\right)\right] \tag{2}$$

Subscripts i-1 refer to the previous angle and subscript i to the current angle. Performing the sum of the heat release rate throughout the combustion process gives the total heat released. The term of pressure variation as a function of angle (dp/dθ) is called pressure rise rate, and according to Jamrozik et al. (2018), the operating characteristics of compression ignition engines are substantially affected by this parameter.

The accumulated heat release was calculated by summing the heat released at each angle range. In addition, dividing the sum of the partial values by the total heat released gives a heat release curve, which varies between 0 and 1, and can compare it with the heat curve contained in the fuel, aiming to characterize the total losses that occur due to inefficiency of combustion.

Dhole et al. (2016), mention that the combustion duration can be calculated from the burned fuel mass fraction curve and the accumulated heat release curve, according to the information presented in Fig. 4.

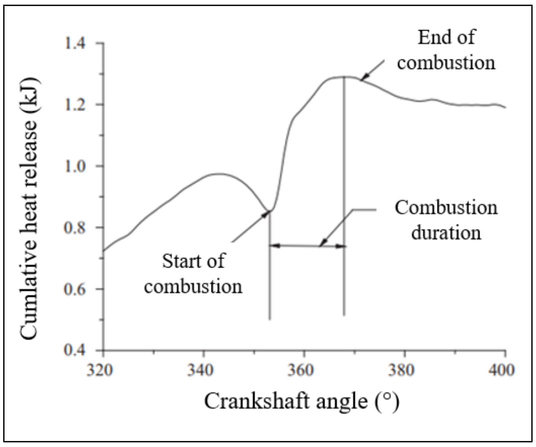


Figure 4. Combustion duration diagram. (Dhole et al., 2016).

The beginning of combustion is considered as the accumulated minimum heat released angle (CHR) of the curve in the region after diesel fuel injection, and the end of combustion is the maximum angle CHR. This method was the one used in this work and based on the angles defined as beginning and end of combustion, it was possible to raise the fuel burned mass fraction curve.

2.3 Ethanol mass fraction (EMF)

To calculate the replacement of diesel oil by ethanol, ethanol mass fraction (EMF), we used Eq. (3).

$$EMF = \frac{\dot{m}_e}{\dot{m}_e + \dot{m}_d} \tag{3}$$

In Eq. (3), the ethanol mass flow rate is represented by $\dot{m}_e$ and the diesel oil mass flow rate is represented by $\dot{m}_d$. In the case of a fraction, the values may vary between 0 and 1, being 0 when no ethanol is used, only diesel oil, and 1 when only ethanol is used.

2.4 Work of cycles

Work at each phase of the cycle and net work were calculated according to Eq. (4).

$$W = \int p dV \quad (4)$$

As the encoder resolution was 1°, Eq. (4) was analytically solved with dθ equal to 1°, obtaining the form of Eq. (5).

$$W_{liq} = \sum_{\theta=0^\circ}^{\theta=720^\circ} \frac{P + P_{i-1}}{2} .(V_\theta - V_{\theta-1}) \quad (5)$$

3. RESULTS AND DISCUSSIONS

From the data collected in the tests and the analyses performed according to the methodology, results were obtained regarding the operation cycle and the combustion, which will be presented below. Tab. 3 shows the ethanol mass fraction and fuel flow rates in each test performed.

Table 3. Fuel consumption and ethanol mass fractions.

Test number	EMF	$\dot{m}_d$, (g/s)	$\dot{m}_e$, (g/s)
1	0.000	0.687	0.000
2	0.155	0.652	0.119
3	0.282	0.566	0.223
4	0.400	0.493	0.329
5	0.501	0.429	0.446
6	0.578	0.386	0.529
7	0.634	0.361	0.627

It is noted that the replacement range explored was widely and with progressive increase, even above the recommended by the literature (up to 50%), reaching the ethanol mass fraction equal to 0.634. For each condition tested the integration of the work produced was performed, and then the net work was calculated, whose results are shown in Tab. 4.

Table 4. Work produced during cycles.

Test number	EMF	$W_{compression}$ (J)	$W_{expansion}$ (J)	W_{intake} (J)	$W_{exhaust}$ (J)	W_{net} (J)
1	0.000	-328.6	1030.0	-4.0	-21.1	675.7
2	0.155	-323.9	1044.9	-4.8	-30.0	686.2
3	0.282	-320.7	1012.7	-4.8	-22.3	664.8
4	0.400	-314.1	988.5	-5.4	-17.0	651.9
5	0.501	-311.8	987.2	-4.5	-25.1	645.7
6	0.578	-314.1	972.2	-4.7	-22.6	630.7
7	0.634	-309.0	970.2	-4.2	-25.7	631.2

On average the intake work represents 0.46% while the exhaust work represents 2.33% of the expansion work. On the other hand, compression work represents, on average, about 31.7% of expansion work. In relation to net work, there is initially an increase of 1.5% in test 2 compared to the control test and later a progressive decrease as the ethanol fraction increases, reaching a maximum reduction of 7.5% in test 7.

Despite the higher pressure peak and greater expansion work during the test in the region near the PMS, as shown in Fig. 5 and Tab. 4, during a longer expansion period the pressure remains higher in test 2, which may be related to the combustion of ethanol blended with air outside the diesel spray region. This combustion would occur later than the burning of the fuel from the spray, providing significant pressure rise when the piston was already in the advanced course of expansion time.

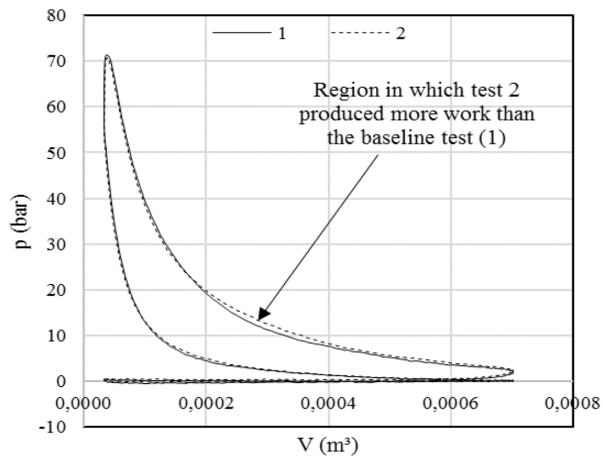
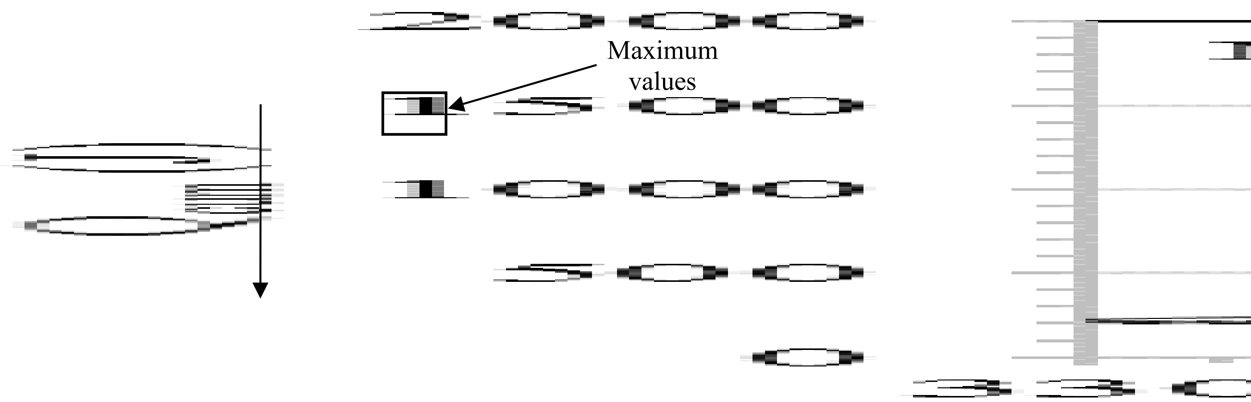


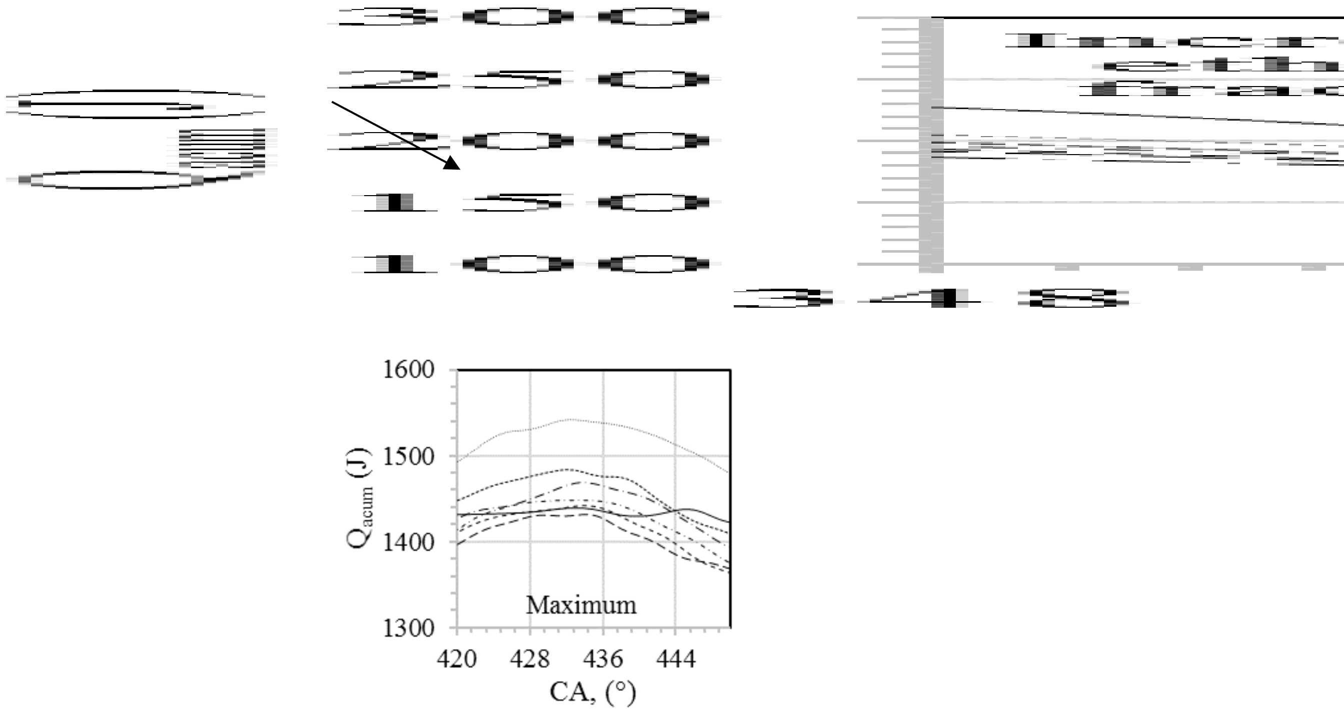
Figure 5. Pressure x Volume diagram of tests 1 and 2.

3.2 Heat release and combustion duration

From the release rate, calculated using Eq. (2), it was possible to sum the heat parcels released at each interval in the angle and thus obtain a cumulative heat release curve for each condition, as shows Fig. 6.



(a)



(b) (c)

Figure 6. Accumulated released heat: (a) all curves (b) minimum region detail; (c) maximum region detail.

It is noted that in the region near the angle of 360°, where the beginning of combustion is expected, the accumulated heat already presents values close to 200 J, that is, the curve does not represent the accumulated heat released during the combustion because it considers the heat added to the working fluid as a function of the compression process. Thus, this curve is used only to determine the beginning and end of combustion according to the method mentioned above.

Applying the same criteria as Dhole et al. (2016), presented in the methodology, the curves of Fig. 6 were used to determine the start and end positions of the combustion, identifying the respective minimum points after the injection of diesel oil, and the maximum points of each of the curves. The values obtained are presented in Tab. 5.

Table 5. Start and end combustion angle.

Test number	EMF	CA_{start} (°)	CA_{end} (°)	Combustion period (°)	$Q_{released}$ (J/ciclo)
1	0.000	356	433	77	1245.75
2	0.155	357	432	75	1360.41
3	0.282	358	432	74	1308.87
4	0.400	359	434	75	1272.02
5	0.501	360	432	72	1272.17
6	0.578	361	434	73	1263.68
7	0.634	362	434	72	1298.33

There is an apparent delay in the start of combustion, starting at 356° in the baseline test and increasing to 362° in the test for the largest ethanol mass fraction. Regarding the end of combustion, there is apparent oscillatory variation of the values obtained, it may be related to the difficulty of accurately assessing the combustion closure, or also to combustion instabilities. The total combustion period decreased with increasing ethanol mass fraction, which can be attributed to the longer ignition delay. There is also the cooling effect caused by ethanol, due to its high latent heat of vaporization, which contributes to the delay in ignition. Telli et al. (2018) state that the latent heat of vaporization of ethanol is 0.92 MJ/kg while that of diesel oil is between 0.23 and 0.60 MJ/kg.

Regarding the heat released during combustion shown in Tab. 5, it already considers only the angular range shown in the same table. Note that the highest value was obtained in test 2 with lower percentage of fumigation, coinciding with the test of higher net work and larger expansion work.

4. CONCLUSIONS

In order to reach the objective proposed in this research, to verify the behavior of the work production in the cycle and the heat release in a compression ignition engine operating in dual-fuel mode with diesel oil and ethanol fumigation. It was measured the pressure inside the cylinder in a small engine and subsequently analyzed the data according to methods proposed in the literature. In this way, after analyses, some conclusions could be listed.

It was possible to operate the engine in an apparently stable manner, without oscillations in the angular velocity. The use of ethanol fumigation wide range of substitution ratios demonstrate the possibility of this technique being feasible for substitution of diesel oil with other fuels of renewable sources, even partially. Other aspects should be accounted for a more accurate conclusion, for example performance and emissions assessment.

The compression work reduced as the ethanol portion was increased, which could be the result of the reduction of the temperature of the admitted load, as well as the expansion work reduced. Therefore, the net work also progressively reduced with the increase of the ethanol fraction, except for the condition of lower substitution.

It was observed that the combustion duration (total period) shows a tendency to fall when using ethanol, which is a favorable point, since the reduction of the combustion time implies a shorter time for the occurrence of heat losses by the walls of the cylinder. This aspect is one of the indicated in the literature as favorable for the use of ethanol in compression ignition engines by the fumigation method.

Finally, the continuity of the research should focus on aspects related to the duration of each phase of the combustion, such as details of ignition delay, duration of pre-mix combustion phases and diffusive combustion, in order to further detail the process and obtain conclusions.

5. ACKNOWLEDGEMENTS

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7. RESPONSIBILITY NOTICE

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