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Effects of n-Butanol Compression Ignition of Ethanol by Rapid Compression Machine

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ABSTRACT

Currently, several transport and fuel production sectors are aiming to develop new technologies that can replace traditional fossil fuels by fuels produced from renewable sources, most of them dubbed as Biofuels. In the particular case of n-butanol, an additive widely used in the blend with diesel oil, it has achieved considerably good results; however, the dependence on fossil fuels still remains present. In this investigation, in order to replace diesel oil in CI engines; n-butanol was tested as an ignition improver when blended with ethanol (Biofuel) as energy source. For this purpose, a rapid compression machine (RCM) was used under different test conditions, such as compression ratio from 16:1 to 24:1. The combustion process of Ethanol / n-butanol blend (10% w/w) was compared to a diesel S10 (as baseline fuel) and with mixture of ethanol / polyethylene-glycol 400 (13% w/w) and showed interesting results. These results were discussed and showed the necessity of several modification in the fuel and the compression ignition system in order to burn the mixture of Ethanol / n-butanol.

1. INTRODUCTION

Currently, petroleum, natural gas, and their sub-products represent approximately 57.5% of the global energy consumption [1]. The utilization of these fuels has increased enormously on the global scale, and therefore, suitable alternative fuels are being actively searched. Since the 1970s oil crisis, the global concern over dependency on fossil fuels (e.g. international tensions, shortages, oil embargoes, etc.) has increased, resulting in local energy alternatives and the progressive research for new energy sources [2].

Today, there are over 1.5 billion vehicles around the world. The global transportation sector, besides others such as the agroindustry, mining, and manufacturing, still represents a huge weight factor on the world economy; being the performance of these sectors highly sensitive to energy conversion efficiency, fuel availability, and costs [3]. Concerns over future petroleum supply and environmental degradation have led to substantial research in the production and use of biomass-derived fuels because of their lower global impact on CO_2 emissions, and its availability. As part of actions along the last decades and in order to reduce the greenhouse gases and preserve the environment, several environmental issues were discussed on international platforms, first in the Club of Rome (1968), then in the United Nations Conference on Environment and Development in Rio de Janeiro (1992), the Kyoto Protocol (1997), and finally, the Paris Agreement (2015) [4]. Some of the problems that were discussed on these international platforms and which are also part of several studies are the diesel engine emissions. Studies indicate how it is necessary to reduce the combustion chamber temperature in order to achieve a uniform equivalence ratio distribution, this with the aim to reduce the local rich region, achieving simultaneous NO_X and PM emission reduction [5].

Until a few years ago, the process named “homogeneous charge compression ignition”, dubbed HCCI, was the most thermodynamically ideal combustion method to achieve this goal; however, nowadays there are other alternative techniques; such as, the reactivity-controlled compression ignition (RCCI), a combustion technique recently investigated, where burning a uniform mixture of fuels with low reactivity (low cetane number) injected in the air, on the port (PFI) or into the combustion chamber (DI), is complemented with another highly reactive (high cetane number) fuel into the cylinder for mixture ignition [6].

Because diesel oil is a non-renewable fossil fuel, current strategies find to employ biofuels to decrease dependence, and also to avoid the emissions associated with diesel oil. Besides biodiesel, the ethanol is also considered as a substitute of diesel oil. In this context, Brazil is a privileged country, not only due to its extensive natural resources; but also because of its favorable climatic and geographical conditions, necessary for biomass-based fuel plantations. Based on this, the “National Alcohol Plan”, a nationwide program financed by the Brazilian government, to phase out fossil fuels in automobiles in favor of sugar cane ethanol, has started in 1975 [7]. This program promoted the use of ethanol in spark ignition engines and by far, it is considered to have achieved the greatest success in replacing petroleum-based fuels if compared with other countries worldwide [8].

When CI engines are operating with ethanol there are some limitations with relation the fuel due to its low cetane number, poor properties in lubricity, viscosity, high hygroscopicity, and its lower heating value [9]. Despite these disadvantages and challenges, ethanol is being increasingly used by CI engines, owing to certain techniques that emerged in the recent years. These techniques include Ethanol fumigation, where ethanol is injected to the intake air. Dual-fuel direct injection, where each fuel is injected separately. Ethanol-diesel blends, where fuels are mixed prior to injection. And additive-ethanol, where an additive is added to ethanol for improving its cetane number [2, 6, 9-11].

The two main approaches for using ethanol in diesel cycle engines include: (1) the use of an ignition improver additive in combination with increased engine CR (e.g., in the Diesel Scania engine, the CR was changed from 17:1 to 24:1), and (2) the use of a glow plug ignition system to heat the air into the cylinder during the compression process [12, 13]. Both concepts have showed advantages and disadvantages. From an engine development point of view, the use of an ignition improver additive is simpler because the ignition aid on the engine is expendable; thereby, implying lower development costs. However, the increase in the fuel cost caused by the use of the ignition improver additive as well as the logistics for the ignition improver distribution could be a significant drawback.

Besides these PEG derivate additives, another substance such as the butanol and its isomers may also be considered as ignition improver of ethanol, as mentioned in the patent EP2204433A1 [14]. Some properties of butanol isomers are showed in Table 1. However, butanol produced from biomass (Biobutanol), usually assumes a straight-chain molecular structure (n-butanol). Biobutanol is obtained by fermenting sugar derived from plant and algae biomass using microorganisms (usually *Clostridium*). This process is called acetone-butanol-ethanol (ABE) fermentation and it involves biphasic fermentation stages: Acidogenesis (acid production stage) and solventogenesis (solvent production stage) [15].

Table 1. Comparison of butanol isomers

Properties	1-butanol	2-butanol	Tert-butanol	Iso-butanol
Density (kg/mm^3)	811.5	806.0	789.0	802.0
Octane number	96.0	108.0	107.0	105.1
Boiling temperature °C)	117.4	99.5	82.4	108
Enthalpy of vaporization (kJ/kg)	582	551	527	566
Auto-ignition temperature (°C)	343	406.1	477.8	415.6
Lower heating value (MJ/kg)	33.2	33.9	32.6	32.96
Flammability limits vol.%	1.4 - 11.2	1.7 – 9.8	2.4 - 8	1.2 – 10.9
Viscosity (mPa s)	2.54	3.09	-	4.31

As objective of this work and in order to compare the combustion performance of the fuels studied, the Ignition Delay (ID) or delay time was considered. The methods to determine ID can be broadly classified into two categories: Direct method (DM) and indirect method (IM) [16].

The IM used to determine the ID in this work, is based on the pressure-time history of a combustion cycle. With a pressure sensor installed in the cylinder head, it is possible to capture the variation of pressure with respect to time accurately. The first and second derivatives of the pressure curve with respect to time (dp/dt and d^2p/dt^2) also allow the evaluation of the ID with good accuracy. The results show the mean ($\bar{X}$) of 50 tests, for each operation regime, and the mean deviation (MD) for the procedure to determine the ID calculated by the IM. In this work, the second derivate of the pressure curve (d^2p/dt^2) was used because it was often not possible to determine the ID based on the first derivate of the pressure curve (dp/dt).

2. MATERIALS AND METHODS

2.1 Fuels

Along the development of the tests, the rapid compression machine, used for this purpose, was fueled with three different fuels: The Diesel oil S10, the mixture of ethanol and PEG 400 (ethanol/PEG400 blend), and the ethanol/n-butanol blend.

The commercial diesel oil S10 or diesel B0 is a type of fuel without biodiesel in its composition; the concentration of sulfur is less than 10 ppm [17]. Moreover, hydrous ethanol, a blend composed of 92.6 – 93.8% (w/w) of anhydrous ethanol and less than 7.4% (w/w) of water was used to prepare the mixtures [18], and this was also purchased from a gas station in Rio de Janeiro. The blends were created by adding PEG 400 and n-butanol in the hydrous ethanol. PEG is an alcohol with a polyester chain that has several applications, from industrial manufacturing to medicine. PEG is soluble in water, methanol, ethanol, benzene, and dichloromethane; however, it is insoluble in ether ethylic and hexane.

Based on the information provided in Table 1 and the fact that when the butanol is produced from biomass; it usually assumes a straight-chain molecular structure (n-butanol). In consequence, n-butanol was selected for use as the ignition improver additive of ethanol in these tests. In order to compare the different properties of some fuels, Table 2 summarizes the physical and chemical [19, 20], and according to these properties, it's possible to indicate that n-butanol or biobutanol have a high potential to overcome the drawbacks introduced by the low-carbon alcohols. Therefore, this fuel can be considered a renewable fuel or an ignition improver additive for ethanol, as showed in this study.

Table 2. Properties of some alcohols and conventional fossil fuels.

Properties	Diesel	Ethanol	n-butanol
Density (g/mL)	0.82-0.86	0.79	0.81
Cetane number	40-55	8	25
Octane number	20-30	108	96.0
Boiling temperature (°C)	180-370	78.4	117.4
Enthalpy of vaporization (kJ/kg) at 25°C	270	904	582
Auto-ignition temperature (°C)	210	434	343
Lower heating value (MJ/kg)	44.5	26.8	33.2
Flammability limits vol.%	1.5-7.6	4.3-19	1.4 - 11.2
Viscosity (mm²/s) at 40°C	1.9-4.1	1.08	2.63

2.2 Experimental setup

A rapid compression machine was installed at the Vehicular

Engineering Laboratory at PUC-Rio (Figure 1). This machine could quickly and easily operate the Otto and diesel cycles.

The RCM simulates a single compression and partial expansion process, which allows more detailed studies of injection, mixing, vaporization, ignition. Sometimes flame development, and combustion duration aren't considered due partial expansion process. This includes optical diagnostics, piston displacement, and combustion chamber pressure data collection [21].

The RCM was equipped with a high-pressure common-rail diesel injection system (Figure 2) and the injector used on the tests was a Bosch 0 445 110 231. The air was introduced into the combustion chamber before compression, and the fuel was injected near of TDC. The injection time and pressure of the fuel mixtures were readjusted to provide the same amount of chemical energy injected than a diesel combustion process.

Figure 1. RCM installed on VEL at PUC - Rio

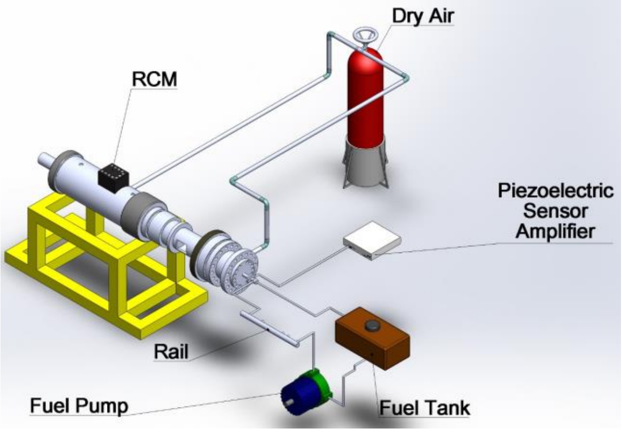

Figure 2. Adaptations and setup in the RCM for compression ignition mode tests

The lower heating value (LHV) for Diesel oil S10 is approximately 44 MJ/kg [22]. Therefore, for each combustion process with ethanol blends, an approximately 80% greater mass of mixture fuel had to have injected compared with the amount of diesel oil S10 injected because the LHV of ethanol is approximately 26.8 MJ/kg and of their mixtures were near of this value [23]. When it's necessary to inject more amount of fuel and the level pressure of the pump and the time of injection are on the limit, it's import to consider the multiple injections, similar to engines, where there are multiple injections, pre-injection, main injection, pos-injection. The temperatures on the wall and piston bowl were set to approximately 90°C. The operation conditions for each test are listed in Tables 3, 4, and 5.

Table 3. Input operations conditions for the test with diesel oil S10 in the RCM

Characteristic	Value		
Compression ratio (-)	16:1	20:1	24:1
Engine speed simulated (rpm)	1500	1900	2400
Maximum piston displacement (mm)	125	140	170
Driving pressure (bar)	19.0	25.5	34.0
Throttle distance (mm)		10	
Air intake pressure (bar)	1.65	1.71	1.74
LHV (MJ/kg)		44.0	
Start of Injection, before TDC (mm)	-1.0, 0.0, 1.0 and 2.0		
Fuel injection conditions	1400 bar/ 1.09 ms	1400 bar/ 1.29 ms	1400 bar/ 1.64 ms

Table 4. Input operations conditions for the test with ethanol/PEG 400 (13%) (w/w) blend in the RCM

Characteristic	Value	
Compression ratio (-)	20:1	24:1
Engine speed simulated (rpm)	1900	2400
Maximum piston displacement (mm)	140	170
Driving pressure (bar)	26	33.4
Throttle distance (mm)	10	
Air intake pressure (bar)	1.71	1.74
LHV (MJ/kg)	25.6	
Start of Injection, before TDC (mm)	1.0, 2.0, 3.0, 4.0, 5.0, 6.0	-1.0, 0.0, 1.0, 2.0, 3.0
Fuel injection conditions	1400 bar / 2.37 ms	1400 bar / 2.96 ms

Table 5. Input operations conditions for the test with ethanol/n-butanol (10%) (w/w) blend in the RCM

Characteristic	Value	
Compression ratio (-)	20:1	24:1
Engine speed simulated (rpm)	1900	2400
Maximum piston displacement (mm)	140	170
Driving pressure (bar)	26	33.4
Throttle distance (mm)	10	
Air intake pressure (bar)	1.71	1.74
LHV (MJ/kg)	26.44	
Start of Injection, before TDC (mm)	3.0, 4.0, 5.0, 6.0	2.0, 3.0, 4.0, 5.0
Fuel injection conditions	1400 bar / 2.29 ms	1400 bar / 2.86 ms

3. RESULTS

All tests were conducted in the rapid compression machine while respecting the quantity of the energy injected into the system for each CR. In addition, it is important to mention that 50 tests were conducted for each condition. This section is divided into three sub-chapters for easier understanding.

- Tests with diesel oil S10.
- Tests with ethanol / PEG 400 (13% w/w) blend.
- Tests with ethanol / n-butanol (10% w/w) blend.

Table 6. Test results with diesel oil S10, in the RCM

	CR = 16:1		**CR = 20:1**		**CR = 24:1**	
	SOI (mm)	**Result**	**SOI (mm)**	**Result**	**SOI (mm)**	**Result**
Comb CP_{max} (bar)	-1.00 ~-10°	Not 36.66	-1.00 ~-12°	Yes 46.88	-1.00 ~-11°	Yes 60.47
Comb CP_{max} (bar)	0.00 ~1°	Yes 58.98	0.00 ~1°	Yes 67.56	0.00 ~1°	Yes 86.30
Comb CP_{max} (bar)	1.00 ~10°	Yes 67.15	1.00 ~9°	Yes 88.41	1.00 ~10°	Yes 112.23
Comb CP_{max} (bar)	2.00 ~13°	Yes 75.76	2.00 ~12°	Yes 94.96	2.00 ~13°	Yes 125.95

3.1 Tests with diesel oil S10

The results of these tests are listed in Tables 6 and 9, in the Appendix. Through Table 6 it's possible to observe that there was no combustion when the diesel oil was injected after the TDC; further, the CR was 16:1. In the engine, it is normal to observe that the main injection occurs after the TDC under different operation conditions. This fact and others described throughout this work imply that it would not be correct to compare the results obtained using the RCM with the results obtained using the engine. Further, in the internal combustion engines, it is possible to observe other injections (pre- and post-injections) beside the main injection; however, in this work, only one injection was used. This and other injection techniques are used to decrease emissions and improve efficiency [11].

Table 6 indicates the influence of the start of injection (SOI) and CR, in the combustion process of diesel oil S10. The highest probabilities of the ignition of diesel oil S10 was when it is injected with greater advance of SOI because it enables greater evaporation of the fuel and better mixing with the compressed air inside the cylinder. Further, it was observed that there is a stroke interval of SOI in which fuel can be injected for combustion, out of this interval it's possible that the fuel not burn, or the combustion be incomplete. It was observed that this SOI interval may be higher when the CR is higher; however, the choice of SOI depends on some combustion parameters; for example, pressure peak, maximum pressure rate, detonation, etc. Therefore, along the tests, it was possible to observe that there was no fuel ignition with a CR of 16:1 when the SOI was in the TDC. This problem was overcome when tests were performed with a higher CR. In addition, it is important to note the increase in the pressure peak when the SOI is greater.

Figure 3 shows the pressure behavior in the tests for each SOI; further, it is important to note the start of ignition and the moment when the pressure into the cylinder is the maximum because these parameters may define the range of the SOI where there is fuel combustion and the engine components are preserved. Normally, the start of ignition occurs in advance when the advance of the SOI is greater; however, the ID may be higher.

In addition, through Figure 3 it's possible to observe that the start of ignition for the SOIs in advance before TDC, and this phenome may be prejudicial because it can originate negative work (efficiency loss) and in the worst scenario, it can knock or detonate (to damage the engine components). An increase of 1 mm in relation to the SOI in the TDC results in the maximum pressure being increased to 13.85%. Further, when this increase is 2 mm, the maximum pressure is incremented to 28.45%. The pressure peaks for the test with the SOI of 0.0, 1.0, and 2.0 mm before TDC are 1.07 and 0.21 ms after TDC and 0.15 ms before TDC, respectively. The analysis of the results shown in Figure 3 and Table 9 indicates that the maximum pressure ratios, after fuel injection for tests with SOIs of 0.0, 1.0, and 2.0 mm before TDC, were 80.17, 98.77, and 128.88 bar/ms, respectively. These maximum pressure ratios occurred 0.83 and 0.02 ms after the TDC and 0.36 ms before the TDC. According to these results, there may have been an increase in noise when the SOI was anticipated.

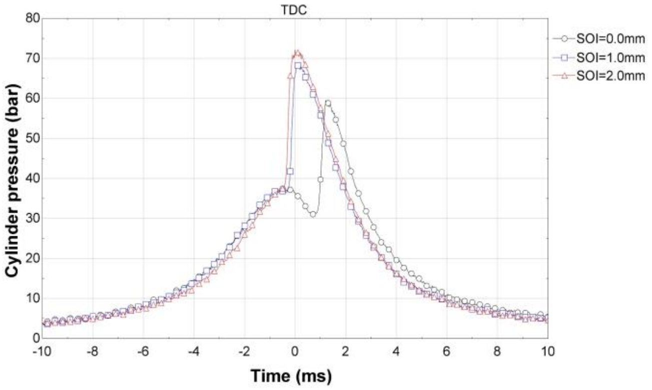

Figure 3. Cylinder pressure for the tests with diesel oil S10, 1500 rpm and CR = 16:1, in the RCM

The curve of the apparent heat release can be used to determine the start of ignition and ID; however, in this work, the method that use the second derivate of the pressure curve (d^2p/dt^2) was selected to determine this parameter.

The curves shown in Figure 4 indicate that it was possible to determine the ignition delay (ID) for these tests by using the indirect method (IM). For the tests with the SOI of 0.0, 1.0, and 2.0 mm before TDC, the ID was 0.59, 0.51, and 0.48 ms, respectively. According to these results, it would be possible to affirm that ID decreases when the SOI is greater; however, during this study, this affirmation was not achieved in all the tests, because there were some tests where the ID increased when the SOI was increased. As described at the beginning of this chapter, 50 tests were conducted for each condition. Therefore, the value of the ID showed in each analysis was chosen by the mean value. Table 10, in the Appendix, shows the mean value of ID ($\overline{X}_{ID}$) and its MD ($D\overline{X}_{ID}$), for better understanding.

Figure 4 shows that it is possible to observe pressure behavior in cylinders and how the SOI influence the start of combustion and pressure peak when the diesel oil is injected under conditions that were listed in Table 3 (CR 20:1). Initially, it was possible to observe that combustion occurred when diesel oil was injected after TDC; this may be caused by a higher CR, which let the pair P-T, pressure and temperature of compressed air to be above the minimal optimal condition at the moment when the fuel is injected, thereby burning the fuel.

Similarly, for the tests with CR of 16:1 and 20:1, when the SOI was higher, the start of ignition occurs in advance. Further, the combustion pressure peak is lower than the compression pressure peak for the SOI after TDC. This effect may be produced because the fuel injected absorbs an energy fraction of the compressed air for vaporization. Therefore, when the ignition starts, the pressure at this point is lower than the compression pressure peak, thereby causing several combustion pressure peaks that are less than the compression pressure peaks, as shown in Figure 4.

Figure 4 shows that the pressure peak decrease to 30.64% when the SOI was 1 mm after TDC in comparison with the pressure peak with the SOI in the TDC. When the SOI is 1 mm before TDC, the pressure peak increased by 30.85 % in comparison with the pressure peak for the test with the SOI in the TDC. In addition, the pressure peaks for the SOI −1.0, 0.0, 1.0, and 2.0 mm before the TDC occurred 3.16, 1.07, 0.64, and 0.17 ms before TDC, respectively. Moreover, the maximum pressure ratios for the same SOI mentioned previously were 21.39, 25.02, 36.80, and 59.07 bar/ms; these occurred 2.92, 0.66, and 0.26 ms after the TDC and 0.23 ms before the TDC, respectively. When these values were compared with the values obtained on the tests with the CR of 16:1, the peak pressure was observed to be lower when there is an increase in the CR. This fact can be attributed to two factors: speed of piston (engine speed simulated) and the conditions (P-T) at the start of ignition. The ID was less with an increase in CR, and therefore, for the same SOI, the conditions of air compressed (P-T) may be different when the start of ignition occurs, thereby causing scenarios when the maximum pressure peak is different.

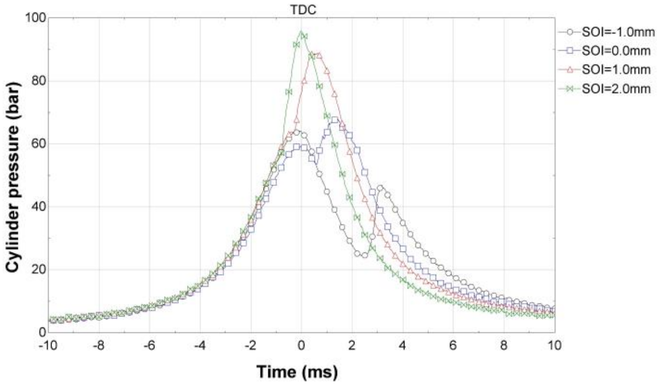

Figure 4. Cylinder pressure for the tests with diesel oil S10, 1900 rpm and CR = 20:1, in the RCM

Figure 5 shows the pressure behavior for the tests with the CR of 24:1 and different SOI. This figure indicates that the pressure peak and maximum pressure rise rate (PRR) can be determined for each test. Similar to the tests with the CR of 20:1, the combustion pressure peak for some tests is lower than the maximum compression pressure. Therefore, in addition to the energy absorbed by fuel to evaporate before the combustion, there is a time interval of vaporization that can be influenced by the displacement velocity of the piston and the air compressed conditions during fuel injection.

Further, the values of the pressure peak, as shown in Figure 5, are higher than the pressure peaks for the tests with the CR of 20:1. This phenomenon is a consequence of the increase in CR. This increase in the pressure peak needs to be monitored because a higher increase in the CR or in the fuel injection advance can cause values of the pressure peaks above the maximum limited to be permitted in the combustion chamber, thereby increasing the noise and the probability of the detonation; in the worst scenario, it can damage the engine components.

The values of the respective pressure peaks are listed in Table 6; these peaks occurred 2.73, 0.92, 0.31, and 0.32 ms before the TDC for tests with SOI at −1.0, 0.0, 1.0, and 2.0 mm before the TDC, respectively. The maximum pressure ratio was calculated for each test based on the analysis shown in Figure 6. The values of this parameter for combustion processes with SOI −1.0, 0.0, 1.0, and 2.0 mm before TDC, were 52.90, 13.31, 42.03, and 73.04 bar/ms, respectively. These values were affected by the speed of the piston.

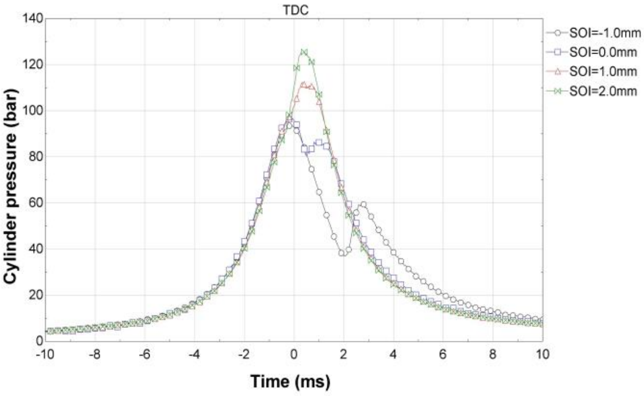

Figure 5. Cylinder pressure for the tests with diesel oil S10, 2400 rpm and CR = 24:1, in the RCM

3.2 Tests with ethanol / PEG 400 (13% w/w) blend

After the tests with diesel oil S10, different tests were conducted using the ethanol / PEG 400 blend, where the percent of PEG 400 in the mixture was 13% w/w. The results of these tests are shown in Table 7. It is important to mention that before conducting the tests described in this work, tests with different percentages of PEG 400 (7, 10 13, 15, and 20% w/w) in the blend have been reported. Unfortunately, the blends with 7 and 10% of PEG 400 did not burn in the tests with the CR of 20:1, because the amount of additive in the blend wasn’t enough. These blends could probably burn in the tests with the CR of 24:1 or higher; however, this was not tested.

It is important to remember that the ID of the ethanol/additive blend decreases when the quantity of the additive (ignition improver) in the mixture is higher; however, its price may increase. Therefore, the ethanol / PEG 400 (13% w/w) blend was selected for the tests with CRs of 20:1 and 24:1. Some preliminary tests showed that this blend does not burn when it is tested under the CR of 16:1; therefore, this CR was not used during the tests with the ethanol blends. The conditions used for the tests are listed in Table 5. In all the tests, the conditions for the injection pressure and equivalence ratio were kept constant. Only the CR and injection duration were changed.

The results of these tests are listed in Table 7, 9A, and 10A; the last two tables are provided in the Appendix. As indicated in Table 7, there was no combustion in the tests with the CR of 20:1 and SOI 0.0 and 1.0 mm before TDC. This phenomenon occurs when ethanol blends are tested under the same conditions as that for diesel oil, and it is attributed to the low cetane number of ethanol and its high enthalpy of vaporization. Therefore, to ensure that the ethanol burns in the CI system, the CR should be increased and the SOI should be anticipated.

Further, Table 7 indicates that the pressure peak increases when the SOI is farther from TDC. During the tests with the CR of 20:1, tests with SOI 7.0, 8.0, and 9.0 were attempted. Unfortunately, the results were not satisfactory because there was no repeatability in the combustion processes under these conditions. Several times, there was no combustion and if there was combustion, the value of the pressure peak was higher than 180 bar. Therefore, to preserve the integrity of the rapid compression machine, these tests were not conducted.

Table 7. Test results with ethanol / PEG 400 (13% w/w) blend, in the RCM

	CR = 20:1		**CR = 24:1**	
	SOI (mm)	**Result**	**SOI (mm)**	**Result**
Combustion	0.00 ~0°	Not	0.00 ~0°	Yes
CP_max (bar)		63.81		87.44
Combustion	1.00 ~9°	Not	1.00 ~9°	Yes
CP_max (bar)		63.10		112.92
Combustion	2.00 ~14°	Yes	2.00 ~13°	Yes
CP_max (bar)		118.94		132.19
Combustion	3.00 ~18°	Yes	3.00 ~17°	Yes
CP_max (bar)		123.23		138.59
Combustion	4.00 ~21°	Yes	-	-
CP_max (bar)		136.17		-
Combustion	5.00 ~23°	Yes	-	-
CP_max (bar)		144.86		-
Combustion	6.00 ~25°	Yes	-	-
CP_max (bar)		148.90		-

As indicated in Table 7, for the tests with SOI 2.0 mm before TDC, the value of the pressure peak increased to 11.14% when the CR changed from 20:1 to 24:1. A similar case is observed for the SOI 3.0 mm before TDC, where the value of pressure peak increased to 12.46% because of the same change. The value of the pressure peak for the test results summarized in Table 8 increased as the SOI goes farther from the TDC.

Figure 6 shows the pressure behavior for the tests of ethanol / PEG 400 (13% w/w), CR of 20:1, and SOI 2.0, 4.0, and 6.0 before TDC. The pressure curves behavior after ignition (Figure 6) is different compared to the pressure curves behavior of the tests with diesel oil S10. The curves for the tests with ethanol / PEG 400 (13% w/w) after the ignition show a higher inclination. The maximum pressure ratios after the ignition for the tests with SOI 2.0, 4.0, and 6.0 mm before TDC were 161.41, 217.44, and 324.86 bar/ms, respectively. Further, the values were obtained at 0.83, 0.57, and 0.29 ms after TDC.

This fact and the higher-pressure peak for the tests with ethanol blends are characteristics of ethanol burn that can cause higher noise levels and damage in the engine components. Normally, ethanol or ethanol blends are injected with an anticipated SOI to ensure that the vaporized fuel finds better conditions of pressure and temperature, thereby causing the ethanol burn, which in several cases may burn fast. Moreover, it is important to mention that in several cases, there may be fuel that is still being vaporized and injected at the ignition instant. Because of the low LHV of ethanol, it is necessary to inject a higher quantity of fuel into the CI process. This challenge can be overcome using different techniques, e.g., increasing the fuel injection pressure, injection duration, or both simultaneously. The techniques described previously do not often solve this problem, and therefore, nowadays, it is necessary to perform more than one injection (pre- and post-injection). The application of multiple injection technology in CI engines allows better control of engine combustion, and it is also effective in reducing not only NO_X and PM but also diesel combustion noise.

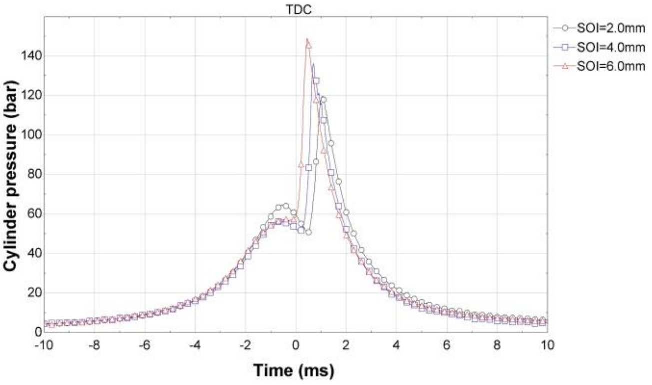

Figure 6. Cylinder pressure for the tests with ethanol / PEG 400 (13% w/w) blend, 1900 rpm and CR = 20:1, in the RCM

The pressure behavior of the tests with ethanol / PEG 400 (13% w/w) and the CR of 24:1 are shown in Figure 8. The figure indicates that there is combustion with the blend injected at the TDC. This result helps understand the effectiveness of the increase in the CR; when the blend is tested with a higher CR, the fuel may be injected close to the TDC, which makes it possible for the combustion processes to not have pressure peaks before TDC and negative works, which may to origin losses in efficiency and engine damages.

Further, similar to the tests with diesel oil S10 and the CR of 24:1, the pressure peak after the ignition was lower than the compression pressure when the SOI is at the TDC or after it. This phenomenon may be attributed to the vaporization time of the fuel and the piston speed (engine speed simulated). As ethanol and ethanol blends have a higher enthalpy of vaporization, it is necessary to reduce the time required for vaporization besides the increase in the CR. This limitation can be resolved by increasing the injection pressure and the number of injections (pre- and post-injection besides the main injection) and reducing the fuel injector hole to increase fuel atomization.

As shown in Figure 7, the pressure peaks for the tests with SOI 0.0, 1.0, 2.0, and 3.0 mm before TDC, occurred 1.05, 0.34, 0.35, and 0.09 ms before TDC, respectively. It is important to highlight that the pressure peak for the tests with the SOI at the TDC was 87.44 bar, which is inferior to the maximum compression pressure of 97.84 bar. Similar to the tests with the CR of 20:1, the inclination of the pressure curve is higher than the diesel pressure curve. For these tests with SOI 0.0, 1.0, 2.0, and 3.0 before TDC, the maximum pressure ratios after the combustion were 23.63, 49.28, 96.05, and 114.08 bar/ms, respectively. These values occurred 0.84, 0.15, and 0.10 ms after TDC, and 0.15 ms before TDC, respectively.

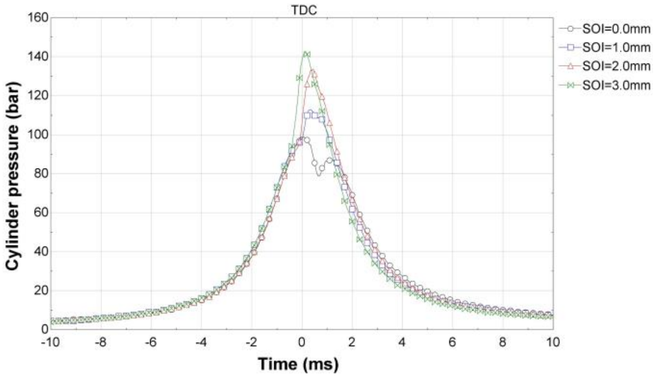

Figure 7. Cylinder pressure for the tests with ethanol / PEG 400 (13% w/w) blend, 2400 rpm and CR = 24:1, in the RCM

The results of the tests with the ethanol / PEG 400 blend are compared with those of the diesel oil S10 under similar CR; the results are listed below.

- Pressure peak: It is higher when the ethanol/PEG 400 blend is tested than when the diesel oil is tested under the same conditions. This result is a characteristic of the combustion with ethanol and its blends. For the same conditions (CR = 20:1 and SOI = 2.0 mm before TDC), the peak pressure in the test with the ethanol/PEG 400 blend was higher than the peak pressure in the test with diesel oil S10. A similar scenario may be observed when the tests under the CR of 24:1 are compared.
- Pressure peak rise: Similar to the pressure peak, this fact is a characteristic of the combustion with ethanol or its blend. High values were observed for the test with the ethanol/PEG 400 blend under the CR of 20:1 and SOI higher than 2.0 mm before TDC. Therefore, tests with this blend under the CR of 24:1 were conducted until the SOI of 3.0 mm before the TDC to preserve the equipment. Further, the PRR value decreased as the CR increased; this fact may be caused by the speed piston.
- ID: Similar to the tests with the diesel oil S10, this parameter decreased when the CR increased. Further, the difference between values of ID (ID for Diesel S10 and ethanol/PEG 400 blend) for the tests with the CR of 24:1 is lower than those for the tests with the CR of 20:1. Thus, a higher CR for the tests with ethanol/PEG 400 blend did not result in similar IDs for both fuels.

3.3 Tests with ethanol / n-butanol (10% w/w) blend

Preliminary tests with mixtures of ethanol and n-butanol (7, 10, and 15% w/w) were conducted. These tests were conducted to determine the appropriate proportion of n-butanol in the ethanol blend, these results can be founded in the SAE Technical Paper 2020-36-0013 [11]. Initially, these blends were tested with the CR of 20.64:1, without considering fuel burn in all tests. Later, these blends were tested under the CR of 26.81:1, considering the fuel burn for the tests with ethanol / n-butanol (10 and 15% w/w) blends.

These preliminary tests indicated similar behaviors of the combustion process for the two mixtures, and the results did not show improvements in terms of the combustion parameters (ID, PRR, etc.). Thus, based on the preliminary result and with an aim of using a minor quantity of the additive in the blend, (lower final fuel price), the ethanol / n-butanol (10% w/w) blend was selected for the tests. One observation related to the preliminary tests with the ethanol/n-butanol blends was the possibility that these blends could burn in a CI process under an intermediate CR between 20.64:1 and 26.81:1. Therefore, to compare the different parameters obtained in these tests, the mixture was tested under CR of 20:1 and 24:1

Table 8. Test results with ethanol / n-butanol (10% w/w) blend, in the RCM

	CR = 20:1		**CR = 24:1**	
	SOI (mm)	**Result**	**SOI (mm)**	**Result**
Combustion CP_max (bar)	2.00 ~14°	Not 61.04	2.00 ~15°	Not 94.07
Combustion CP_max (bar)	3.00 ~18°	Not 60.21	3.00 ~17°	Yes 126.40
Combustion CP_max (bar)	4.00 ~20°	Not 62.10	4.00 ~19°	Yes 140.39
Combustion CP_max (bar)	5.00 ~22°	Not 61.52	5.00 ~23°	Yes 151.00

Table 9. Combustion parameters for diesel oil S10, ethanol / PEG 400 (13% w/w) blend and ethanol / n-butanol (10% w/w) blend, in the RCM

Fuels	**SOI (mm)**	**CR = 16:1**			**CR = 20:1**			**CR = 24:1**		
		$\bar{X}_{ID}$ (ms)	**P_{max} (bar)**	**PPR_{max} (bar/ms)**	**$\bar{X}_{ID}$ (ms)**	**P_{max} (bar)**	**PPR_{max} (bar/ms)**	**$\bar{X}_{ID}$ (ms)**	**P_{max} (bar)**	**PPR_{max} (bar/ms)**
Diesel S10	-1.00	-	-	-	1.36	63.68	21.39	1.07	93.76	52.90
	0.00	0.59	58.98	81.17	0.51	67.56	25.02	0.42	96.96	13.31
	1.00	0.51	67.15	98.77	0.37	88.41	36.80	0.21	112.23	42.03
	2.00	0.48	75.76	128.88	0.30	94.96	59.07	0.15	125.95	73.04
PEG 400	0.00	-	-	-	-	-	-	0.49	97.84	23.63
	1.00	-	-	-	-	-	-	0.36	112.92	49.28
	2.00	-	-	-	1.34	118.94	161.41	0.35	132.19	96.05
	3.00	-	-	-	1.50	123.23	145.93	0.38	138.59	114.08
	4.00	-	-	-	1.59	136.17	217.44	-	-	-
	5.00	-	-	-	1.71	144.86	322.91	-	-	-
	6.00	-	-	-	1.78	148.90	324.86	-	-	-
n-Butanol	3.00	-	-	-	-	-	-	1.43	126.40	100.85
	4.00	-	-	-	-	-	-	1.65	140.39	174.14
	5.00	-	-	-	-	-	-	1.69	151.00	191.89

Table 10. Ignition delay for the tests with diesel oil S10, ethanol / PEG 400 (13% w/w) blend and ethanol / n-butanol (10% w/w) blend, in the RCM

Fuels	SOI (mm)	CR = 16:1		CR = 20:1		CR = 24:1	
		$\bar{X}_{ID}$(ms)	$D\bar{X}_{ID}$ (ms)	$\bar{X}_{ID}$ (ms)	$D\bar{X}_{ID}$ (ms)	$\bar{X}_{ID}$ (ms)	$D\bar{X}_{ID}$ (ms)
Diesel S10	-1.00	-	-	1.36	0.16	1.07	0.12
	0.00	0.59	0.12	0.51	0.10	0.42	0.04
	1.00	0.51	0.11	0.37	0.06	0.21	0.05
	2.00	0.48	0.07	0.30	0.06	0.15	0.04
PEG 400	0.00	-	-	-	-	0.49	0.05
	1.00	-	-	-	-	0.36	0.04
	2.00	-	-	1.34	0.12	0.35	0.05
	3.00	-	-	1.50	0.05	0.38	0.07
	4.00	-	-	1.59	0.07	-	-
	5.00	-	-	1.71	0.08	-	-
	6.00	-	-	1.78	0.11	-	-
n-Butanol	3.00	-	-	-	-	1.43	0.02
	4.00	-	-	-	-	1.65	0.06
	5.00	-	-	-	-	1.69	0.08

As stated in the "Materials and methods" section, n-butanol can be considered a derivative of petroleum (fossil fuel - non-renewable source) due that it can be obtained as a co-product during petroleum refinement. However, also it is produced via renewable sources through the ABE process, it is called biobutanol. Therefore, the ethanol/biobutanol blend is considered a renewable fuel that can substitute diesel oil in total or partially. The main objective of this study is to determine the possibility and future challenges involved in the use of n-butanol as an ignition improver additive for ethanol for use in CI engines.

Tables 8, 9, and 10 list the results related to the tests with the ethanol / n-butanol (10% w/w) blend. Through Table 8, it's possible to observe the results of the tests, under the conditions described in Table 5, there was no combustion when the blend was tested with the CR of 20:1; this phenomenon was already observed during the preliminary tests, and it was confirmed by this test group. Besides, no combustion was observed for the test with SOI 2.0 mm before TDC. This phenomenon could be caused by the high enthalpy of vaporization of the blend. Therefore, it was necessary to use a greater SOI (SOI farther from TDC) to obtain the fuel burn. Besides the SOI used in this test group and the ones listed in Table 8, some tests with SOI 6.0 and 7.0 mm before the TDC were also conducted. However, similar to that in the tests with the ethanol / PEG 400 blend, no combustion was observed in most cases, and at other times, the combustion processes did not have repeatability and the pressure peak was higher than 180 bar.

Table 8 indicates that the pressure peak increased to 11.06% when the SOI changed from 3.0 to 4.0 mm before TDC. Moreover, the increase in the pressure peak was 19.46% when the SOI changed from 3.0 to 5.0 mm before TDC. This fact showed that once more that one of the characteristics of ethanol blends combustion changes at the SOI. A greater advance of the SOI causes greater pressure peaks and a higher PRR.

Figure 8 shows the pressure behavior for the tests under the conditions listed in Table 5. In this figure, besides high-pressure peaks, high-pressure rise rates appear after fuel ignition. The high-pressure peaks are characteristics of ethanol combustion processes; however, higher pressure rise rates compared to those obtained in the tests with diesel oil S10 and ethanol/PEG 400 blend under the CR of 24:1 were observed in the tests with ethanol/n-butanol blend (Table 9). It is possible that this characteristic may be a property of n-butanol combustion, which is why this value tends to increase when the CR is increased.

- Start of ignition: The start of ignition for the ethanol/n-butanol blend occurred after the start of ignition for the ethanol/PEG 400. Therefore, it is easy to note that the ID for the tests with ethanol/n-butanol is higher than the ID for the tests with ethanol/PEG 400. This fact was attributed to the cetane number for the fuels; the cetane number of the ethanol / n-butanol blend is lower than that of the ethanol / PEG 400 blend.
- Pressure peak value: Due to the higher ID for the tests with the ethanol/n-butanol blend, the conditions (P-T) at the moment of the start of ignition are poor in comparison with the conditions at the start of ignition of the ethanol/PEG 400 blend; this causes a lower pressure peak; however.

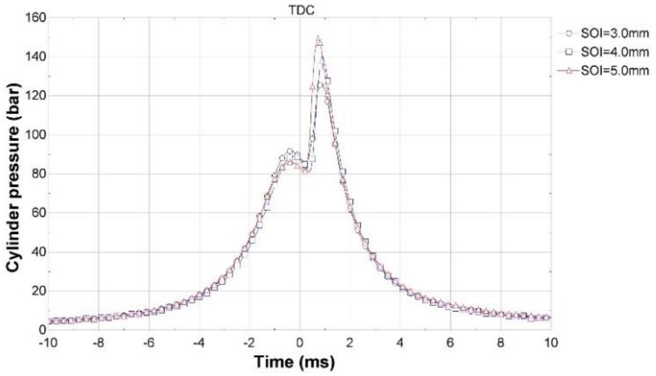

Figure 8. Cylinder pressure for the tests with ethanol / n-butanol (10% w/w) blend, 2400 rpm and CR = 24:1, in the RCM

Moreover, the maximum pressure rises ratios for tests with SOI 3.0, 4.0, and 5.0 mm before the TDC were determined; they are shown in Figure 8. These values were 100.85, 174.14, and 191.89 bar/ms, which occurred 0.59, 0.68, and 0.51 ms after TDC, respectively. These results indicate the possibility of a high noise level and the increase in component wear when the ethanol/n-butanol blend is used in the engine.

4. CONCLUSIONS

During the tests, it was possible to observe that fuels based on a mixture of ethanol and self-ignition additives (PEG400 and n-butanol as additives) can't be used directly in a compression ignition process, without modifying the compression ratio and the start of injection, when compared to diesel oil. According to what was observed during the final tests, it was necessary to increase the compression ratio from 16:1 to 20:1, in such a way that the ethanol and PEG mixtures would burn. In the case of mixtures of ethanol and n-butanol, a greater increase (24:1) was needed.

Observing the results, it can also be mentioned that, for all tests with similar compression ratios, the maximum pressure, as well as the maximum rate of pressure variation, product of the fuel combustion, was always higher when using ethanol mixtures. compared to tests using diesel oil. This is one of the characteristics of the ethanol burning process. Therefore, in the engine calibration processes, it is of great importance to observe pressure peaks, in order to avoid deterioration of the engine elements, as well as the generation of high noise levels.

In tests with ethanol blends, it can be said that the combustion process, for a given compression ratio, presents a range of SOI in which the ignition of the mixture takes place and, in addition, within this range, it can be find a SOI that gives a minimum ignition delay. Ethanol is a highly corrosive fuel and has low lubricity when compared to diesel oil. During the tests it was necessary to change the injection pump twice and once the fuel injector. Therefore, it is vitally important to use additives that reduce corrosivity and increase lubricity.

Due to the amount of energy contained in ethanol, the compression ignition process requires injecting approximately 80% more by mass of ethanol powered than diesel. The greater mass of additive ethanol can be made available, via an increase in injection time, injection system pressure or the diameter of injector holes. But when there are limitations, it would be interesting to be able to perform multiple injections.

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NOMENCLATURE

ABE	Acetone-Butanol-Ethanol
CI	Compression ignition
CN	Cetane number
CO_2	Carbon dioxide
CP	Combustion pressure
CR	Compression ratio
DI	Direct injection
$D\overline{X}$	Mean deviation
HCCI	Homogeneous charge compression ignition
ID	Ignition delay
LHV	Lower heating value
NO_x	Nitrogen oxides
P	Pressure, bar
PEG	Polyethylene-glycol
PFI	Port fuel injection
PM	Particular matter
PPR	Pressure rise rate
RCCI	Reactivity controlled compression ignition
RCM	Rapid compression machine
SOI	Start of injection, mm
TDC	Top dead center
VEL	Vehicular Engineering Laboratory
w/w	Referent mass fraction
$\bar{X}$	Mean

Subscripts

ID	Ignition delay
max	maximum