Combustion, Performance and Emissions of a Diesel Power Generator with Direct Injection of B7

and Port Injection of Ethanol

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ABSTRACT

This work investigates a diesel engine operating with diesel oil containing 7% biodiesel (B7) and hydrous ethanol with concentrations varying from 5% to 30%. The experiments were conducted in a 49 kW diesel power generator, equipped with an electronic ethanol injection unit installed in the intake manifold and without any modifications in the diesel oil injection system. The results showed a decrease of in-cylinder pressure and net heat release rate with the use of ethanol at low loads and an increase at high loads, in comparison with B7. Increasing ethanol injection caused increased ignition delay and decreased combustion duration. Fuel conversion efficiency was raised up to 13% with the use of ethanol. The use of 30% ethanol in the fuel caused a reduction of carbon dioxide (CO_2) emissions up to 12% and nitric oxide (NO) up to 53%. Carbon monoxide (CO), total hydrocarbons (THC) and oxides of nitrogen (NO_X) emissions increased with ethanol addition. The replacement of 20% of diesel fuel by ethanol showed the lowest penalties on NO_X emissions.

Keywords: ethanol; diesel engine; combustion; emissions; power generation.

2

1 INTRODUCTION

In recent years, the allowable limits of internal combustion engine pollutant emissions has been reduced as a consequence of increasing concern for the environment and air quality. Ethanol is a renewable fuel that can be obtained from plants such as sugar cane and corn. Several authors highlight ethanol characteristics such as the high latent heat of vaporization, high oxygen content and high burning rate, giving it high potential to reduce oxides of nitrogen (NO_X) and particulate matter (PM) emissions from diesel engines [1-3]. Brazil is a major ethanol producer [4] and its main source is sugar cane, which has one of the highest ratio of renewable energy produced to fossil energy used in the production of the fuel.

There are different techniques to use ethanol in diesel engines: blended to diesel oil, fumigated in the intake manifold, dual injection of ethanol and diesel oil, spark ignition conversion and surface ignition [5]. Diesel oil demand can be replaced by up to 50% using ethanol fumigation, up to 90% using dual injection and up to 25% using diesel oil-ethanol blends [6]. Compared with diesel oil, the main limitations of the use of ethanol in diesel engines are the ethanol higher auto ignition temperature, the lower cetane number, which difficult the compression ignition, the lower low heating value (LHV), the lower miscibility of the fuels, mainly with the presence of water, and the lower lubricant properties [7-10].

Bodisco [11] reported that the addition of fuels to the intake air for combustion engines, called indirect injection, has been investigated since the late 1920s, with the commercialization of the first dual-fuel vehicle in 1939. The author points out that methanol and ethanol are suitable for use as secondary fuel. Ethanol fumigation in the intake manifold is a technique of dual fuel engine operation, where ethanol is mixed with the intake air while diesel fuel is injected directly into the combustion chamber as the pilot fuel. This method requires little engine modifications and ethanol is injected into the intake manifold by low-pressure nozzles [12,13]. The high compression ratio of diesel engines is an advantage when exploring the high resistance to detonation of ethanol [14].

Padala et al. [14] conducted experiments in an automotive single engine diesel engine using ethanol fumigation. Diesel fuel was injected directly into the combustion chamber and an ethanol injector was installed in the intake manifold with electronically controlled time and duration of injection. Tests were carried out varying the ethanol content up to 70%, compared to diesel oil, based on the energy required for each demanded engine load. The results showed an increase in the ignition delay with the ethanol addition, reduction of CO emissions and increase in emissions of NO_X and THC.

3

Zhang et al. [15] conducted experiments in a four-cylinder diesel engine with direct injection of diesel oil and rated power of 88 kW. An electronic control unit was developed for controlling the injection of ethanol in the intake manifold. The replacement percentages used, in energy basis, were 10% and 20% ethanol. The results showed an increase in fuel consumption, with ethanol fumigation, and reduction of fuel conversion efficiency at low loads and increase at higher loads, when compared with diesel oil performance. Regarding emissions, the authors reported an increase of total hydrocarbons (THC) and carbon monoxide (CO) emissions, as higher percentages of ethanol were used, mainly at low loads. NO_X emissions were reduced, attributed to ethanol cooling effect, but carbon dioxide (CO_2) emission increased.

Tutak [16] presented experimental results using E85 (85% ethanol and 15% gasoline) fumigation in a diesel engine, replacing up to 90% diesel oil by E85. For replacement levels above 50%, the engine presented combustion problems. The results showed that E85 fumigation caused lower fuel conversion efficiency at low loads and higher fuel conversion efficiency at high loads. The increase in E85 fumigation increased ignition delay, peak pressure and peak heat release rate. The author evaluated emissions using E85 and found, compared to pure diesel fuel, a reduction in NO_X emission at low and medium loads and an increase at full load. The results showed increased CO and THC emissions with increasing percentages of ethanol.

Britto et al. [17] tested a single cylinder diesel engine with compression ratio adjustment with a diesel injector located in the center of the combustion chamber and two ethanol injectors positioned upstream of the cylinder head inlet. The results showed increased CO and THC emissions, attributed to the ethanol cooler effect and non- homogeneous air-ethanol mixture.

This work presents the use of diesel oil containing 7% of biodiesel (B7) directly injected into the engine combustion chamber and hydrous ethanol injection into the engine intake manifold and its effect on combustion, performance and emissions. Ethanol injection was electronically controlled, with concentrations of 5%, 10%, 15%, 20%, 25% and 30%. The experiments were performed in a 44 kW diesel power generator.

2 METHODOLOGY

2.1 Experimental Setup

The main characteristics of the 44 kW diesel engine used in this study shown by Tab. 1. The original engine geometry characteristics and the mechanically controlled diesel oil direct injection system

4

were not changed for the tests. An electronic injection system was adapted to the engine intake system [18] to control ethanol injection in the dual-fuel operation. The electronic injection system consists mainly of a magnetic sensor and a knock sensor, for engine phase synchronization, and microcontrollers, to control the injection timing, according to engine load demand. A common rail system was developed and adapted to the engine intake manifold for multiport ethanol injection, ensuring that the same amount of fuel is available for each cylinder. The ethanol injectors were positioned 130 mm from the intake valves. The tests were conducted with fixed injection timing of diesel fuel, at 23 crank angle (CA) degrees before top dead center (BTDC), and ethanol injection at top dead center (TDC), in the beginning of the intake stroke.

Table 1 Diesel engine and generator details

EQUIPMENT	PARAMETER	TYPE OR VALUE
	Cycle	Four strokes
	Diesel oil injection	Direct
	Bore × stroke	102 mm × 120 mm
ENGINE	Number of cylinders	4, in line
	Total displacement	3.922 L
	Intake system	Naturally aspirated
	Rated power	44 kW
	Number of poles	4
	Voltage	220 V
GENERATOR	Number of phases	3
	Rated power	55 kVA
	Frequency	60 Hz

The intake air mass flow rate was measured through an orifice plate, with maximum uncertainty of ± 2.3 kg/h. The temperatures of ethanol, diesel oil, ambient air, inlet air, orifice plate inlet, exhaust gas and cooling water temperature were monitored using K-type thermocouples. The maximum uncertainty of the measured exhaust gas temperature was ± 7ºC; the remaining temperatures were measured with maximum uncertainty of ± 2ºC. The inlet air humidity was measured through a thermo-hygrometer with

uncertainty of ± 2.5%, and the ambient pressure was determined through a Torricelli barometer with resolution of ± 1.3 kPa. Diesel oil and ethanol consumption were measured through platform balances, with resolutions of 5 g and 1 g, respectively.

The exhaust gas concentration of total unburned hydrocarbons (THC) was measured by a heated flame ionization detector (HFID) analyzer, with resolution of ± 1 ppm. Oxides of nitrogen (NO_X) and nitric oxide (NO) concentrations were measured by a heated chemiluminescent detector (HCLD) analyzer, with resolution of ±1 ppm. Carbon monoxide (CO) and carbon dioxide (CO_2) concentrations were determined by non-dispersive infrared (NDIR) analyzers, with resolutions of ±1 ppm and ± 0.01%, respectively. In-cylinder pressure was measured by a piezoelectric pressure transducer with resolution of ± 0.5%. A trigger wheel with a magnetic sensor was used to synchronize the pressure data with the engine cycle. The system was set to obtain the pressure data at 0.1 crank angle intervals. Figure 1 shows a schematic draw of the measurement system. The total uncertainty of the results presented is a combination of both the statistical spread and the instrument uncertainty.

BP – Barometric pressure	RU – Relative humidity
ID – Diesel injector	TA – Intake air temperature
IE – Ethanol injector	TD – Diesel oil temperature
KS – Knock sensor	TE – Ethanol Temperature
MS – Magnetic sensor	TEx – Exhaust temperature
PC – In-cylinder pressure	TI – Inlet air temperature
PT – Power transducer	TR – Room temperature
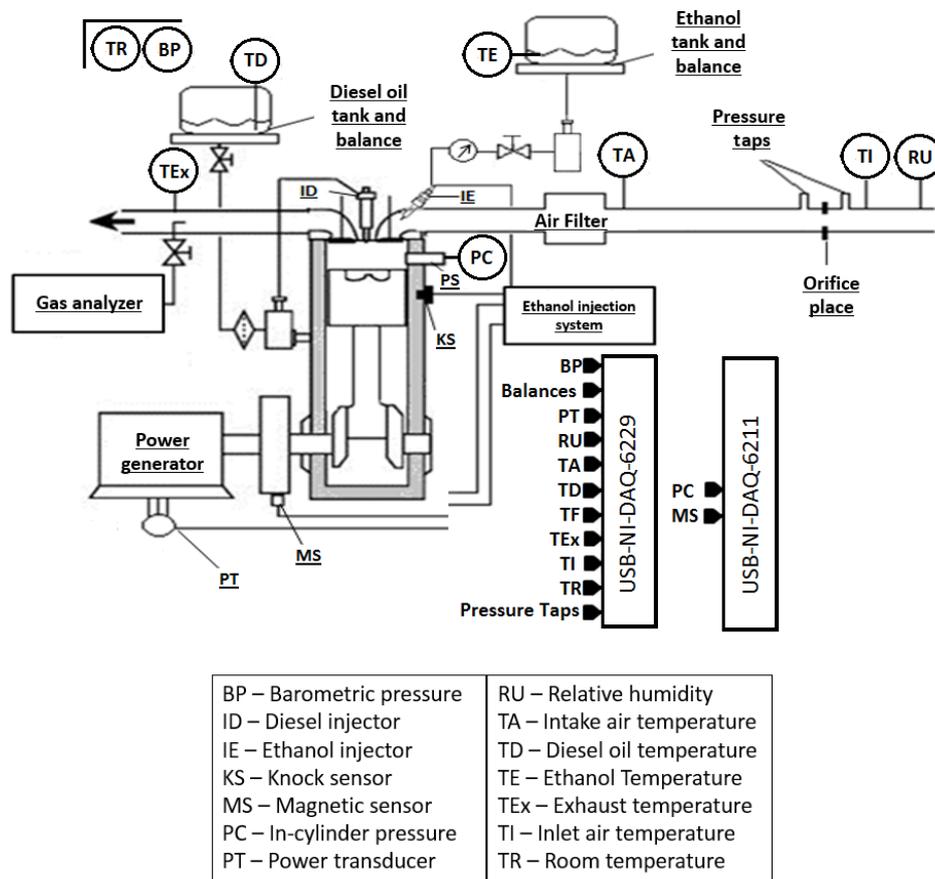
Fig 1 Schematics of the experimental apparatus

2.2 Experimental procedure

The fuels used during the tests were diesel oil containing 7% of biodiesel (B7) and hydrous ethanol (92.6% purity). The replacement percentages used were 0% (B7E0), 5% (B7E5), 10% (B7E10), 15% (B7E15), 20% (B7E20), 25% (B7E25) and 30% (B7E30). The tests were conducted at the constant engine speed of 1800 rev/min. The loads were varied from 0 kW to 37.5 kW and the readings at each load were performed after the engine reached the steady state condition, observing the exhaust gas and the cooling water temperatures. The tests were conducted according to ISO 3046-1:2002 standard [20], which shows the correction of the load power and fuel consumption results to standard conditions.

To set the ethanol injection system, the injected ethanol mass was initially estimated considering the same fuel conversion efficiency for standard operation with diesel oil (B7). Thus, the energy contained in a percentage of B7 mass amount consumed at a specific operating condition in the original engine configuration can be provided by a replacement mass amount of ethanol according to [19]:

$$m_E = P_\% . \frac{m_{B7} . Q_{LHV,B7}}{Q_{LHC,E}} \tag{1}$$

where m_E is the ethanol mass flow rate (kg/h), m_{B7} is B7 mass flow rate (kg/h), $Q_{LHV,B7}$ is the low heating value of B7 (MJ/kg), $Q_{LHV,E}$ is the low heating value of ethanol (MJ/kg) and $P_\%$ is the percentage of ethanol replacement.

The energy content of both fuels, B7 and ethanol, were considered for calculation of the fuel conversion efficiency in dual fuel operation [15,16]:

$$\eta_t = \frac{3600 . W}{m_{B7} . Q_{LHV,B7} + m_E . Q_{LHV,E}} . 100\% \tag{2}$$

where η_t is the engine brake fuel conversion efficiency (%) and W is the engine output power (kW).

The concentration of the exhaust gas component is measured in % or ppm. To convert molar concentration into specific mass concentration per unit of energy produced, in g/kW.h, the following expression is used:

$$c_{,g/kw.h} = \ c_{,ppm \, or \, \%} \cdot (m_{air} + m_{B7} + m_E) \cdot \frac{\rho_{c}}{\rho_{ex}} \cdot \frac{1}{W} \qquad (3)$$

where $C_{,g/kw.h}$ is the component specific composition (g/kW.h), $C_{,ppm \, or \, \%}$ is the component composition

(ppm $\times 10^{-6}$ or % $\times 10^{-2}$), m_{ar} is the intake air mass flow rate (kg/h), ρ_C is the gas component density

(kg/m³) and ρ_{ex} is the exhaust gas density (kg/m³).

The in-cylinder pressure data was used to determine the ignition delay, the combustion duration

and the net heat release rate. The ignition delay is determined from the time interval between the start of

injection and the start of combustion [21]. The start of combustion was determined from the second order

derivative of the in-cylinder pressure diagram [22]. Combustion duration was determined as the crank

angle interval between the beginning of heat release and the crank angle where there is an accumulation of

95% of the total amount of heat released [6,23]. The heat release analysis was made for provide the effects

of the ethanol on combustion characteristics. The apparent net heat release rate was calculated from

application of the first law of Thermodynamics [21]:

$$\frac{dQ_n}{d\theta} = \left(\frac{\gamma}{\gamma - 1}\right) \cdot p \cdot \frac{dV}{d\theta} + \left(\frac{1}{\gamma - 1}\right) \cdot V \cdot \frac{dP}{d\theta} \qquad (4)$$

where $dQ_n/d\theta$ is the apparent net heat release rate (J/°CA), γ is the ratio of specific heats, c_p/c_v, p is the

cylinder pressure (Pa), θ is the crank angle (°CA) and V is the cylinder gas volume (m³).

3 RESULTS AND DISCUSSION

Figures 2 and 3 show the in-cylinder pressure history and the net heat release rate curves as a

function of crank angle and the amount of ethanol injected into the engine intake manifold for the loads of

0 kW and 37.5 kW, respectively. The curves follow the same trend but with a shift of the peak pressure

with increasing amount of ethanol due to increased ignition delay. At low loads, the lower temperature of

the cylinder gas and the longer ignition delay lead to a peak heat released displaced from the TDC, causing

decreased peak pressure.

At the load of 0 kW, the peak cylinder pressure is generally reduced with increased ethanol content in the fuel (Fig. 2). The peak heat release rate peaks does not show a particular trend with ethanol content in the fuel. Despite the larger amount of accumulated fuel during the ignition delay, the concentrations above 5% of ethanol did not show significant increase of peak heat release rate. This can be explained by a poor combustion due to the higher cooling effect caused by ethanol, dominant at low loads, causing also reduced peak pressure.

At high loads, the ethanol-diesel oil-air mixture is richer and more fuel is burned during the premixed combustion phase (Fig. 2). In addition, the cylinder gas temperature is higher and the ignition delay is shorter. This behavior causes an increase in peak pressure and heat released. With increasing engine load, the effect of higher amount of air-fuel mixture formed during the ignition delay and burned in the premixed phase is more evident, being dominant for pressure increase with the increase of ethanol concentration [25].
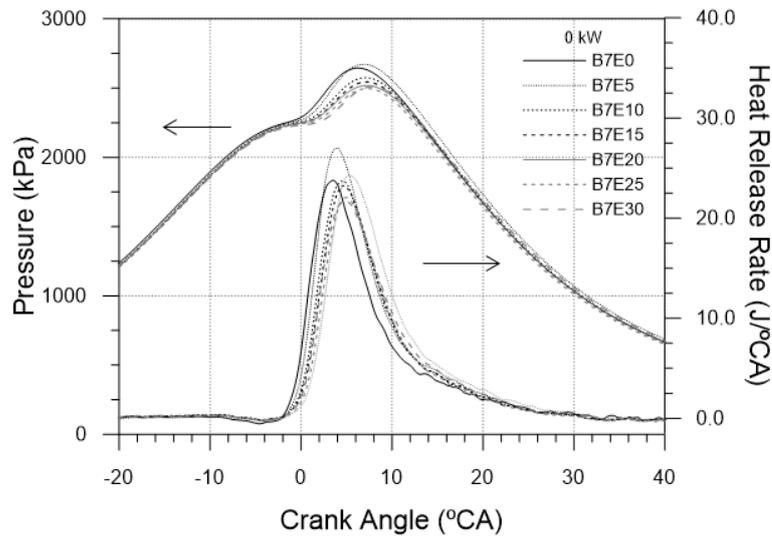
Fig 2 In-cylinder pressure and heat release rate at 0 kW

Figure 3 shows the results of in-cylinder pressure and heat released for the load of 37.5 kW. There is an increase in peak pressure with increasing ethanol content in the fuel. At this load, the difference in the net heat release rate between different ethanol concentrations is amplified, with an increase in the magnitude of the peak pressure and displacement form TDC. This is caused by the fuel accumulated during the ignition delay and the faster burning of ethanol [13]. Several authors have found similar behaviors of in-cylinder

9

pressure [1,6,13]. The increase in heat release can increase local temperatures of the cylinder structure, requiring the evaluation of the engine cooling system efficiency.
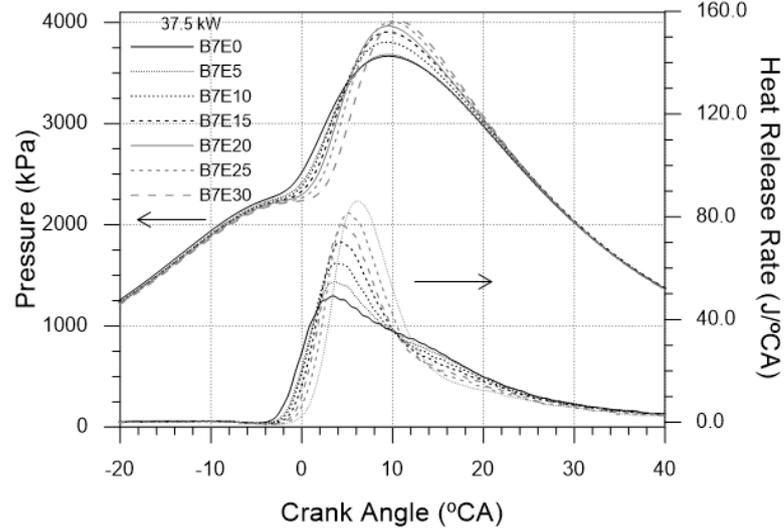
Fig 3 In-cylinder pressure and heat release rate at 37.5 kW

Figure 4 shows increased ignition delay with increasing amount of injected ethanol. The high latent heat of vaporization of ethanol decreases in-cylinder gas temperature, thus reducing the cetane number of the ethanol-diesel oil mixture and, as a consequence, increasing the ignition delay [6,13,14,16,24]. Furthermore, when B7 is injected in the cylinder it is not taken by air only, but by an ethanol-air mixture with lower oxygen concentration when compared with pure air, thus increasing the ignition delay. The in-cylinder gas dilution is increased with increasing ethanol injection [14,16]. The results also shows that engine load influences ignition delay, since it is dependent on the cylinder gas temperature and pressure conditions [21]. For B7E0 operation the ignition delay decreased at high loads, due to high in-cylinder temperature. For ethanol operation, at low loads, the in-cylinder temperature is low and the cooling effect of ethanol fumigation is dominant. At high loads and high ethanol concentration, the gas dilution effect and the low cetane number of ethanol increase the ignition delay. The shortest ignition delay, 21.7ºCA, was verified for B7E0 fuel at 37.5 kW, and the longest ignition delay, 26.2ºCA, occurred to B7E30 at the same load.
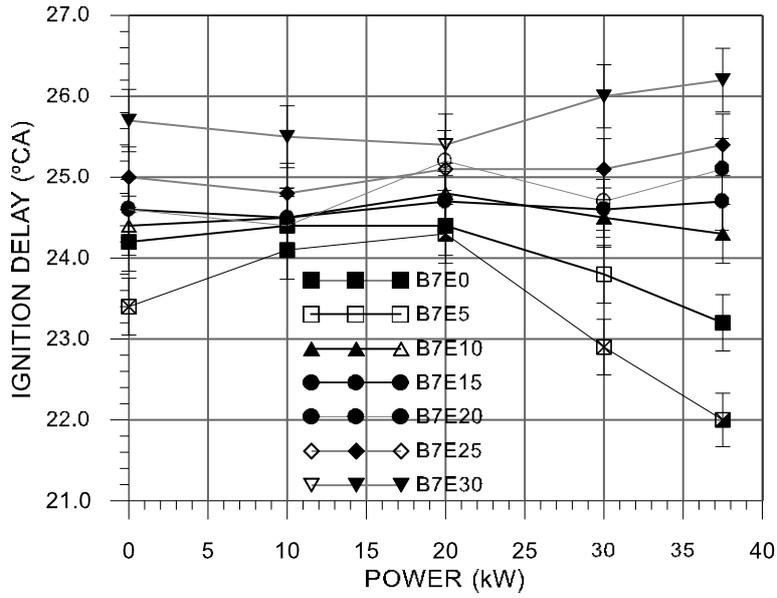
Fig 4 Variation of ignition delay with ethanol concentration and load power

Figure 5 shows the combustion duration as a function of engine load and fuel. Increasing ethanol concentration decreased the combustion duration, justified by the higher heat release rate in the premixed combustion phase caused by the addition of ethanol in the air and increased ignition delay. The addition of ethanol increases the amount of oxygen in the mixture, which reduces the pyrolysis process and increases oxidation during combustion, reducing the combustion duration [23].
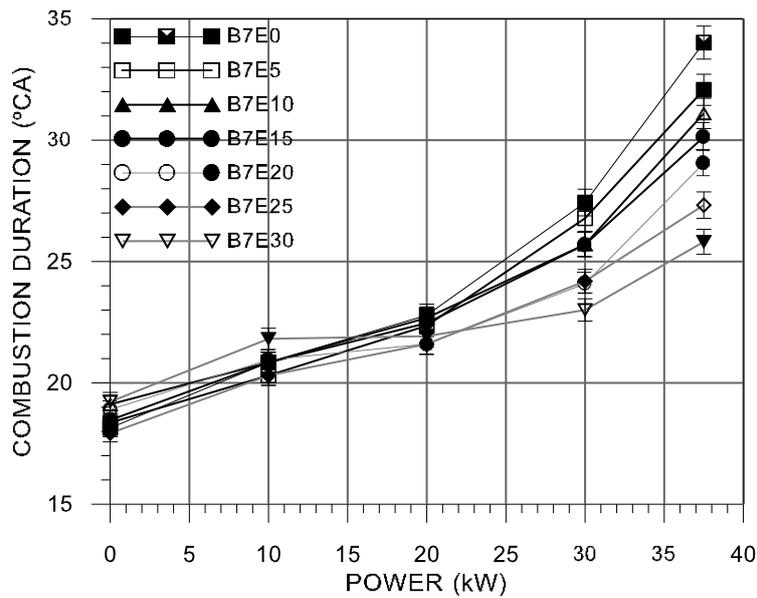
Fig 5 Variation of combustion duration with ethanol concentration and load power

Figure 6 shows the fuel conversion efficiency results for the different ethanol concentrations and engine load. Ethanol fumigation in diesel engine affects its performance by different factors. The gases inside the cylinder are cooled when ethanol is admitted during the intake stroke, which causes poor air-fuel mixture formation and tends to reduce the fuel conversion efficiency. The addition of ethanol tends to increase the ignition delay, as shown in the above results, causing a larger fuel amount to burn during the premixed phase of combustion. Furthermore, the air-ethanol mixture burns faster, compared with B7, which may increase the fuel conversion efficiency by reducing the time for heat exchange between the gases and the combustion chamber walls [13,26]. These factors act simultaneously and are dependent on ethanol percentage in the fuel and engine load.
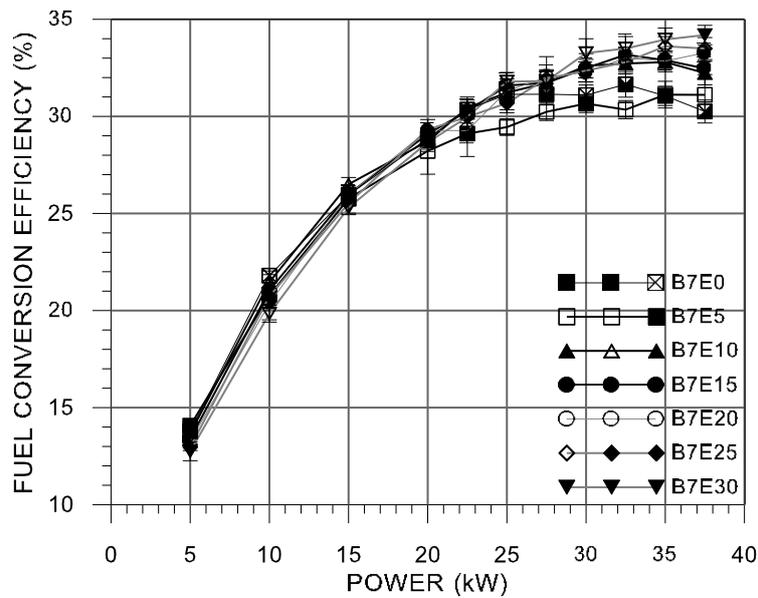
Fig 6 Variation of fuel conversion efficiency with ethanol concentration and load power

In general, there was a reduction of fuel conversion efficiency at low loads and increase at high loads (Fig. 6). At low loads the cooling effects of ethanol vaporization are dominant, decreasing the in-cylinder gas temperature. Besides, the low amount of ethanol injected form an air/fuel mixture that can be too lean to allow combustion to proceed, resulting in reduced fuel conversion efficiency [16]. At high loads, the larger quantity of fuel burned increases the cylinder gas temperature. These effects, together with the faster burning of ethanol and increased amount of fuel burned in the premixed phase of combustion, in

comparison with B7, are determinant to decrease or increase the fuel conversion efficiency [1,13,26,27]. In addition, at low loads, the reduction of the peak pressure and heat release rate could have contributed to reduce the fuel conversion efficiency. At high loads, there was an increase of these two parameters, having, therefore, increasing fuel conversion efficiency. The largest reduction occurred for B7E20 at the load of 10 kW, when the fuel conversion efficiency was 6.6% lower than operation with B7E0, and the largest increase was 13.0%, found for operation with B7E30 at the load 37.5 kW.

Figure 7 shows the behavior of the specific CO_2 emission for the different ethanol concentrations and loads. As the load was increased, CO_2 emission was reduced for all ethanol concentrations. At a given load, in general, increasing ethanol concentration in the fuel also caused a reduction of CO_2 emission. The reductions obtained for CO_2 emissions were up to 12.2%, for B7E25. The decrease of cylinder gas temperature due to the high ethanol latent heat of evaporation inhibits CO oxidation, which leads to a reduction of CO_2 emission [14,15]. The presence of oxygen and a single carbon bond in ethanol molecule increases fuel conversion efficiency (Fig. 6), helping to reduce CO_2 production (Fig. 7).
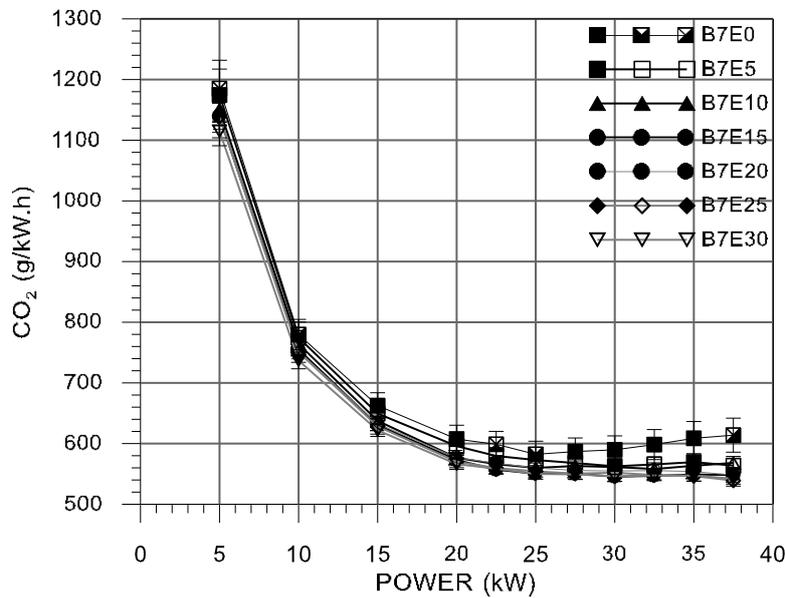
Fig 7 Variation of specific carbon dioxide emissions with ethanol concentration and load power

Figure 8 shows that CO specific emissions was increased with ethanol concentration in the fuel for all engine loads tested. Zhang et al. [15] explain that CO emission tend to increase in situations where there is incomplete combustion, under low temperatures and very lean mixtures, and is controlled by the

13

local equivalence ratio, being an indicative of combustion quality. CO oxidation occurs at high combustion temperatures. At low loads, the addition of ethanol leads to a reduction of cylinder gas temperature, resulting in quality loss and lower combustion oxidation of CO to CO_2 in the expansion stroke, increasing the CO emission [12,15]. With the addition of ethanol, there is a reduction of excess air in the cylinder, which has a strong influence to increase CO emission [3]. At high loads, the effect of ethanol vaporization to decrease cylinder gas temperature is reduced, due to the higher combustion temperatures attained, thus reducing the differences of CO emissions between the ethanol blends and B7.
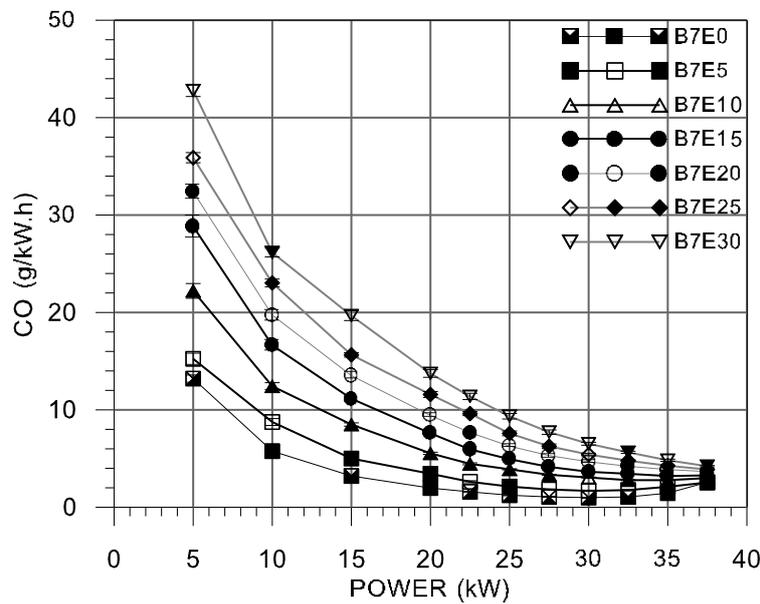
Fig 8 Variation of specific carbon monoxide emissions with ethanol concentration and load power

Figure 9 shows the specific THC emission for the different ethanol concentrations. As it occurred for CO emissions, THC emission increased with the addition of ethanol. Unburned hydrocarbon formation is strongly dependent on combustion quality; thus, the results indicate the use of ethanol provokes poor combustion. Several factors may have caused the increase of THC emission, such as temperature reduction due to ethanol injection, which causes lower rates of vaporization and mixture formation. For Chauhan et al. [9], at low loads, the indirectly injected ethanol is unable to impinge on surfaces and be distributed, causing poor fuel distribution, large amounts of excess air, low exhaust temperature and lean air-fuel mixtures tend to escape, resulting in higher unburned fuel. The air-ethanol mixture trapped in cylinder crevices during the compression and combustion processes, under increasing cylinder pressure, returns to

14

the cylinder during the expansion and exhaust strokes, under decreasing cylinder pressure, and partially undergoes a post-flame oxidation process. The unburned amount escape though the exhaust valve and contribute to the increase of hydrocarbon emissions, a characteristic phenomenon of port fuel spark ignition engines [13,27]. At high loads, the temperature increase allows more burning of the air-fuel mixture, improves the quality of combustion and reduces the increase of hydrocarbon emission [13]. Several researchers, such as Surawski et al. [3], Chauhan et al. [9], Tutak [16] and Yao et al. [27], also report the increase of CO and THC emissions.
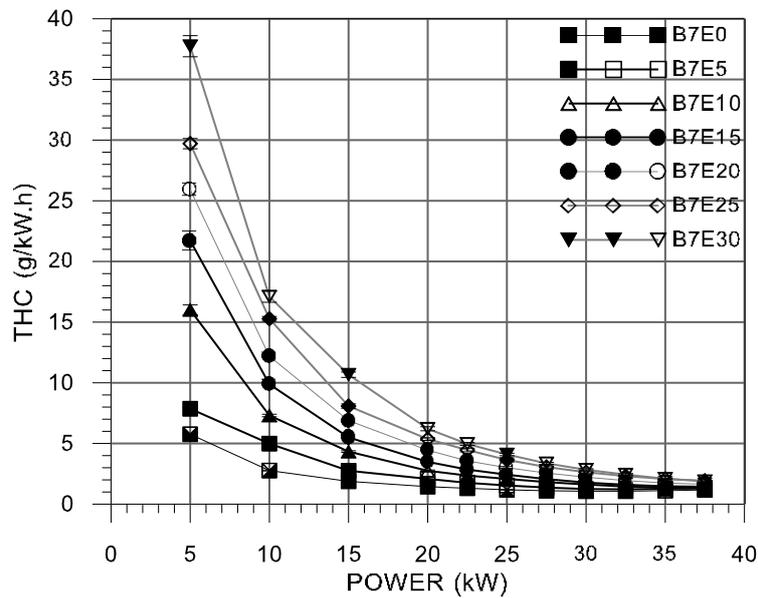
Fig 9 Variation of specific total hydrocarbon emissions with ethanol concentration and load power

Figure 10 shows that NO emission was reduced for all loads when ethanol was used as fuel. The reductions attained were up to 53.2%, for B7E30. Surawski et al. [3] also found reduction of NO emission for all engine power range with ethanol concentration up to 40%. Several factors affect NO_X formation and emissions in a diesel engine. The oxygen content and low cetane number of ethanol lead to high temperature peaks and cause increased NO_X, but, on the other hand, the high enthalpy of vaporization and low flame temperature of ethanol lead to decreased NO_X emissions. The effects of ethanol in NO emissions are dependent on engine operating conditions and fuel characteristics [28]. Several authors associated reduction of NO emissions to decreased gas temperature caused by ethanol fumigation [15]. This effect is more significant at low loads, since at high loads richer air-fuel mixtures tend to increase combustion temperature.
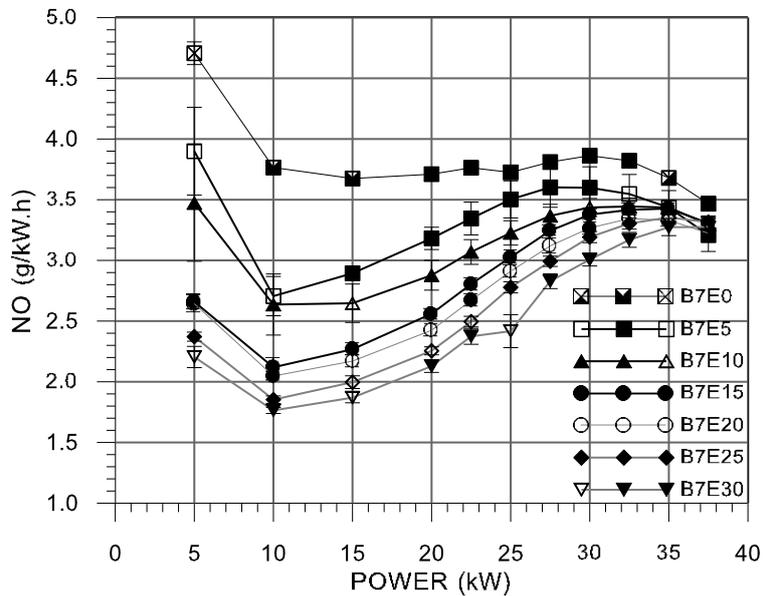
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Fig 10 Variation of specific nitric oxide emissions with ethanol concentration and load power

Despite the reduction of NO emission with ethanol fumigation, it was observed an increase of specific NO_X emissions for all loads (Fig. 11). The increase in NO_X emission is due to the increase of NO_2 formation, as Padala et al. [14] and Tutak [16] showed it. This behavior can be caused by the cooling effect of ethanol, as the decrease in temperature favors NO_2 formation. Heywood [21] explains that one of nitrogen dioxide formation mechanism is by the combination of NO formed in the flame front with hydroperoxyl radical (HO_2), also producing hydroxyl (OH). NO_2 would subsequently combine with oxygen element (O) to form NO and molecular oxygen (O_2), unless the NO_2 formed in the flame is quenched by mixing with cooler fluid.

Zhu et al. [23] explain that the NO oxidation efficiency depends on the formation of hydroperoxyl radical. The authors cite studies that shows that ethanol could produce HO_2 through the thermal degradation behavior of alcohols with NO, and that HO_2 is promptly formed during oxidation of oxygenated fuels. Thus, ethanol can be a source of HO_2 and increase the oxidation of NO into NO_2, especially at low and medium loads. Furthermore, their results showed significant increases of unburned fuel emission, which favors the formation of NO_2 due to the presence of ethanol, a good oxidizing [15], and due to the cold fuel condition, which inhibits the conversion of NO_2 to NO [21].

16

From an analysis of Figs. 10 and 11 for each fuel curve when the engine operated with ethanol, there is tendency of increasing NO emissions for loads below 10 kW and no single trend is observed for higher loads. From the difference of NO_X and NO concentrations, it can be inferred that NO_2 concentration always increase with decreasing load. At low loads the relatively low temperatures attained in the cylinder do not favor the decomposition of NO_2 into NO, while, as the load and combustion product gas is increased, a larger amount of NO_2 is decomposed into NO.
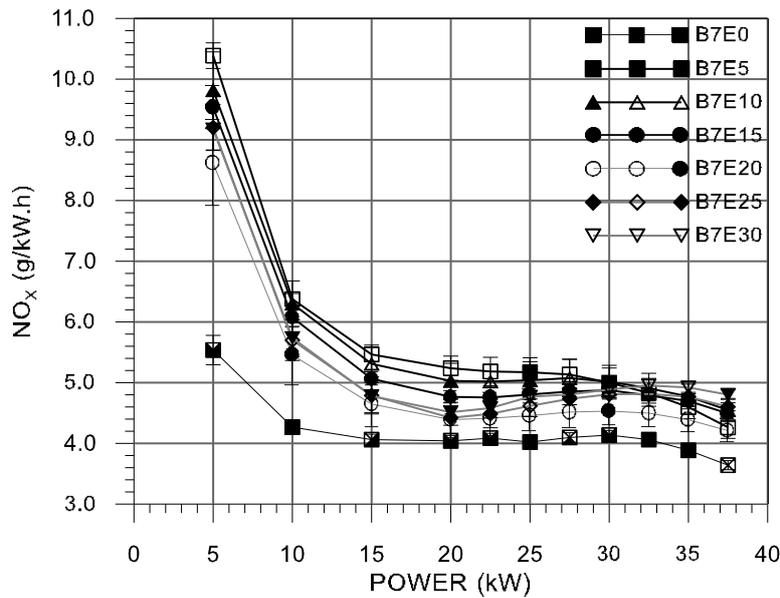
Fig 11 Variation of specific oxides of nitrogen emissions with ethanol concentration and load power

The studied diesel engine does not have any exhaust gas aftertreatment technology. Thus, when operating with diesel oil only, the engine does not meet the current Brazilian emissions standards for diesel engines, PROCONVE (Air Pollution Control Program for Motor Vehicles) P7, based on Euro V European standards for heavy-duty engines, which requires systems as exhaust gas recirculation (EGR) or selective catalytic reduction (SCR). Ethanol addition to B7 increased CO_2, THC and NOx emissions, however aftertreatments systems could lead to a reduction in these emissions. SCR system is a well-stablished technique to achieve very low levels of NO_X [29-31]. Tadano et al. [29] showed reductions up to 93% in NO_X using SCR systems in a diesel engine. EGR systems are also very effective in reducing NO_X emissions [32-33], but increases smoke emissions, which can be reduced by ethanol addition to the fuel [33-35].

Another extensive used aftertreatment system is diesel oxidation catalyst (DOC), used to oxidize THC and CO emissions to H_2O and CO_2, respectively, for diesel oil with low sulfur content [36]. This

17

system is able to reduces CO and THC emissions to the standards limits [36-42]. Zhang et al. [41] and Wei et al. [42] also showed that alcohol fumigation in diesel engines increased CO, NO_2 and THC emissions, but, after catalytic conversion, the concentrations of these components were significantly reduced, up to 38% for THC and 16% for CO [41], and for levels near diesel fuel operation with DOC system [42]. For diesel fuel operation, Wei et al. [42] showed reductions up to 90% in CO and THC emissions for high loads.

4 CONCLUSIONS

Ethanol was added to B7 in a diesel engine by fumigating in the intake manifold and without any change in the original engine geometry characteristics and in the mechanically controlled diesel oil direct injection system. Ethanol addition increased ignition delay and reduced combustion duration. Peak cylinder pressure and heat release rate were reduced at low loads and increased at high loads. The fuel conversion efficiency was reduced at low loads up to 6.6%, using 20% ethanol in the fuel, and increased up to 13.0% at high loads, for 30% ethanol. CO_2 and NO emissions were reduced in all engine load range tested, reaching up to 12% CO_2 reduction and 53% NO reduction with 30% ethanol concentration in the fuel. CO, THC and NO_X emissions always increased with the use of ethanol. The partial substitution of 20% of diesel fuel by ethanol fumigation at moderate to high loads in an engine with unmodified diesel injection settings is recommended for the lowest penalties on NO_X emissions. In general, ethanol fumigation in a stationary diesel engine may require optimization of diesel injection and/or exhaust gas aftertreatment to meet emissions standards.

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23

LIST OF FIGURE CAPTIONS

LIST OF TABLE CAPTIONS

Table 1 – Diesel engine and generator details.

BP – Barometric pressure	RU – Relative humidity
ID – Diesel injector	TA – Intake air temperature
IE – Ethanol injector	TD – Diesel oil temperature
KS – Knock sensor	TE – Ethanol Temperature
MS – Magnetic sensor	TEx – Exhaust temperature
PC – In-cylinder pressure	TI – Inlet air temperature
PT – Power transducer	TR – Room temperature

Fig 1 Schematics of the experimental apparatus
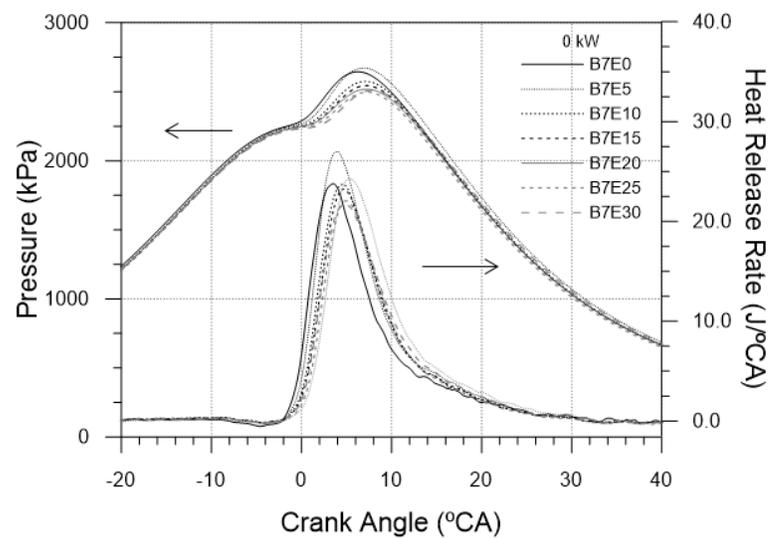
Fig 2 In-cylinder pressure and heat release rate at 0 kW
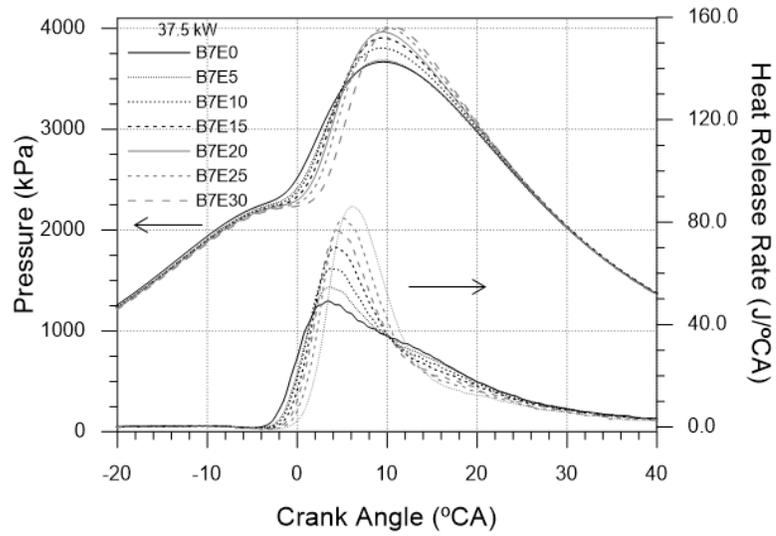
Fig 3 In-cylinder pressure and heat release rate at 37.5 kW
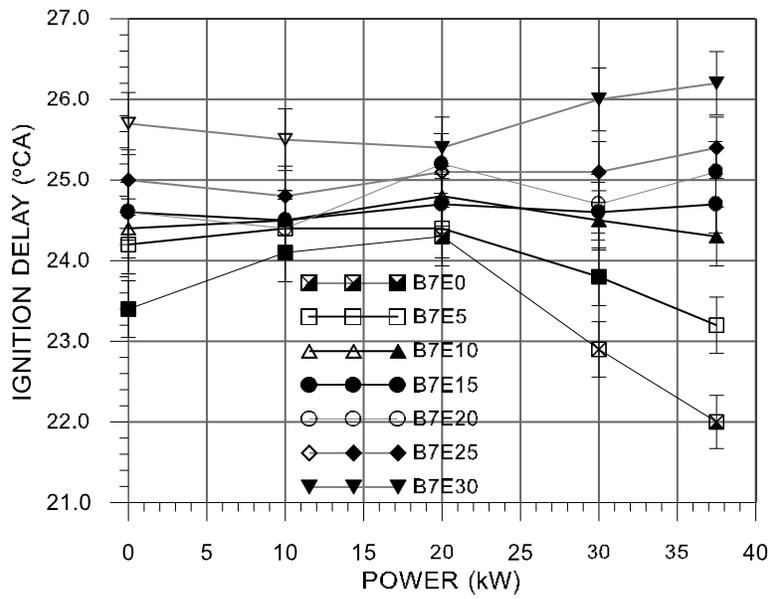
Fig 4 Variation of ignition delay with ethanol concentration and load power
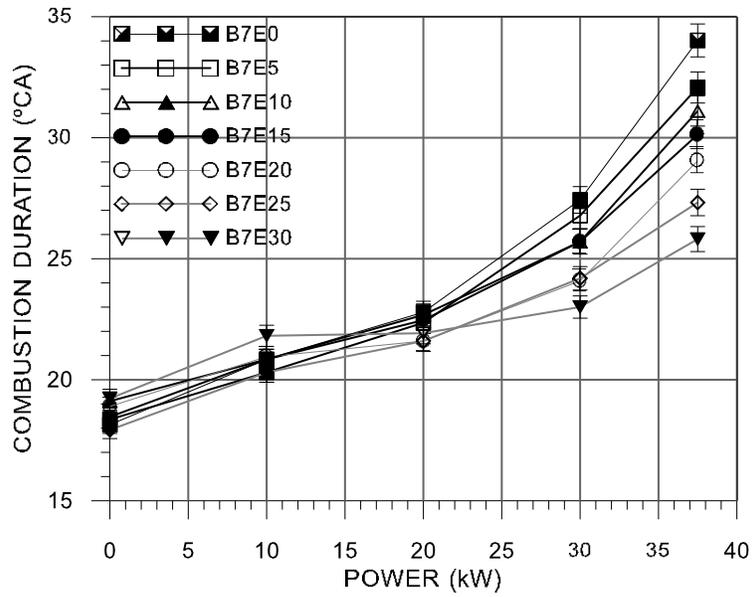
Fig 5 Variation of combustion duration with ethanol concentration and load power
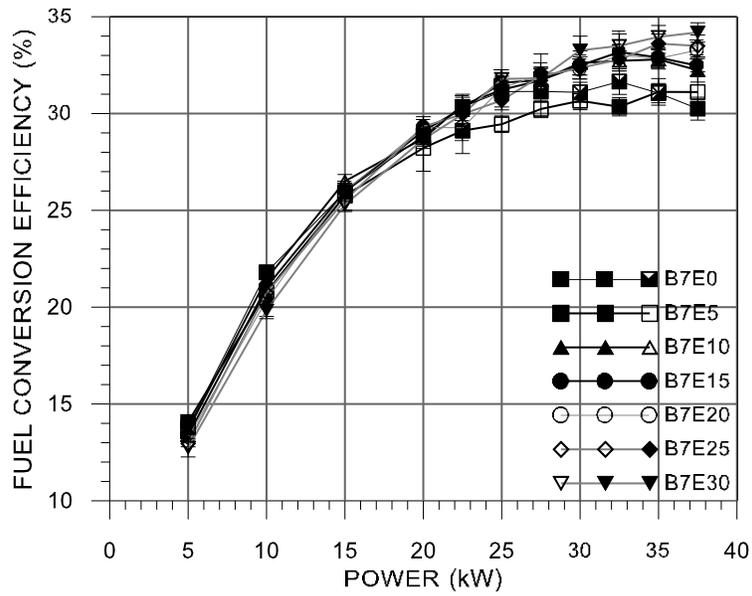
Fig 6 Variation of fuel conversion efficiency with ethanol concentration and load power
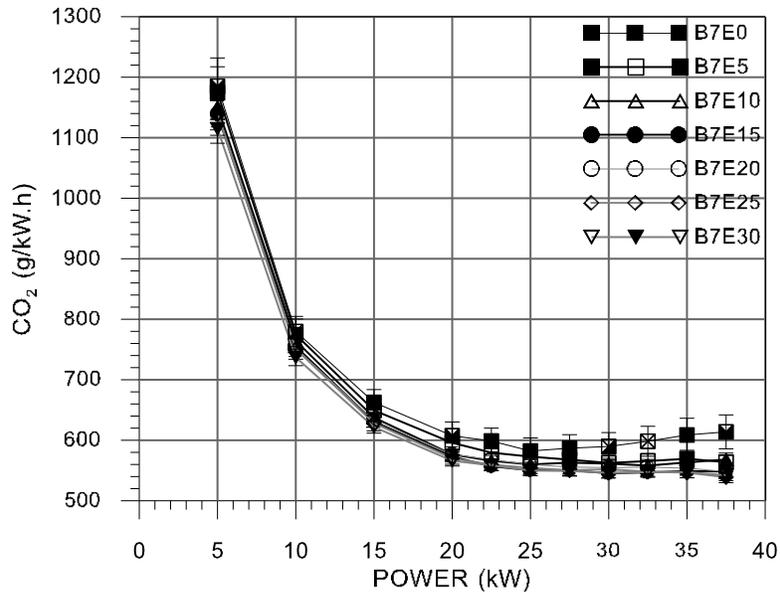
Fig 7 Variation of specific carbon dioxide emissions with ethanol concentration and load power
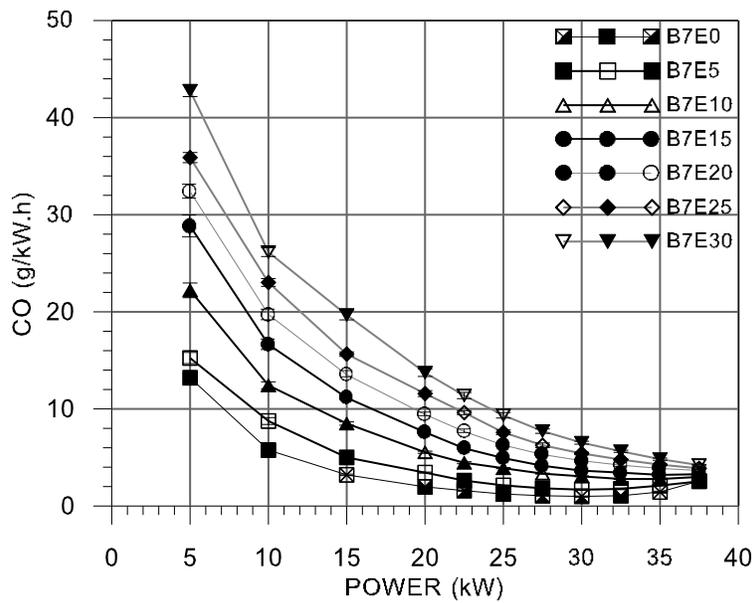
Fig 8 Variation of specific carbon monoxide emissions with ethanol concentration and load power
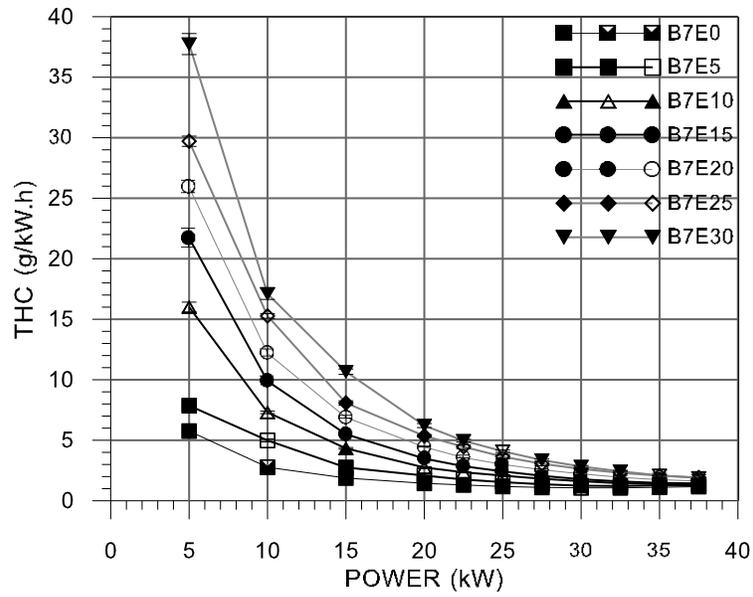
Fig 9 Variation of specific total hydrocarbon emissions with ethanol concentration and load power
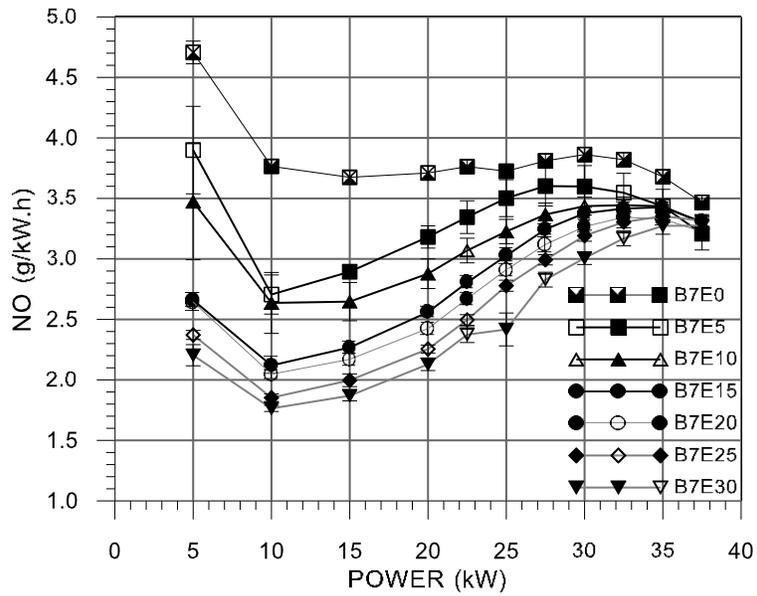
Fig 10 Variation of specific nitric oxide emissions with ethanol concentration and load power
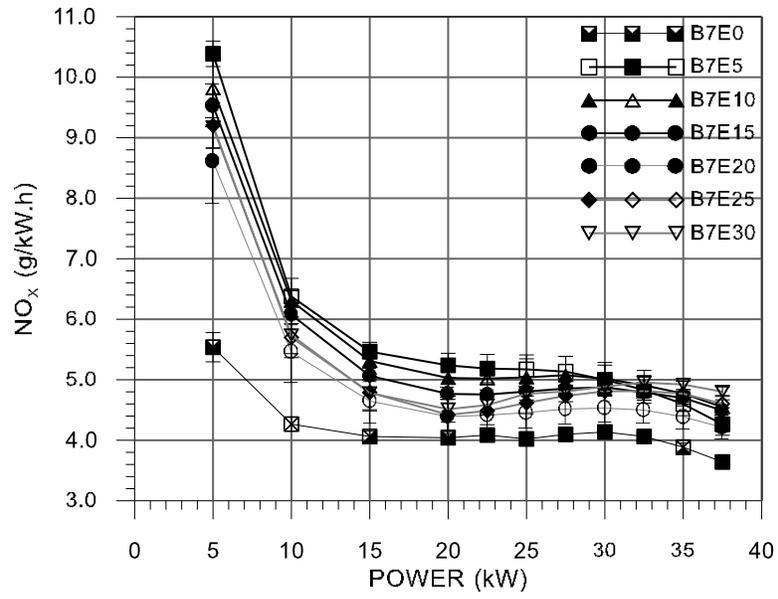
Fig 11 Variation of specific oxides of nitrogen emissions with ethanol concentration and load power

Prof. Francisco R. Cunha
Editor-in-Chief
Journal of the Brazilian Society of Mechanical Sciences and Engineering

Dear Editor,

Thank you for the Reviewers' useful comments and suggestions on the structure of our manuscript. We have modified the manuscript accordingly, and detailed corrections are listed below point by point:

Reviewer #1:   The previous comments and indications are well reflected on the revised manuscript. Therefore, I would like to recommend this revised version as an accepted paper for the journal of BSME after the correction according to following editing comment.
Comments
In the manuscript, please remove the section numbers of 5. Acknowledgement and 6. References in pages 19 as Acknowledgement and References.

Authors' response: The number 5 was removed from the Acknowledgments section title, and the number 6 was removed from the References section title, on page 19.

Reviewer #2: I agree with revisions done by the authors and suggest the paper be accepted for publishing.

The manuscript has been resubmitted to your journal. We look forward to your positive response.

Sincerely yours,

Prof. J. R. Sodré, Ph.D.
Corresponding author