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Performance and emissions analysis of additional ethanol injection on a diesel engine powered with A blend of diesel-biodiesel

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ABSTRACT

This work shows the performance and emissions profile of a diesel engine operating with ethanol injected into the air of the inlet manifold in time with the high-pressure injection of a mixture of diesel and biodiesel. The ethanol injection uses an electronic management system that detects the high-pressure pulse in the diesel injection line as the injection trigger. The air intake temperature reduction caused by the ethanol injection could be evaluated. The tests were made in an engine at 1800 rpm, connected to an electric generator. The high-pressure injection fuel was always a binary blend of diesel and biodiesel, which was supplemented by injected ethanol, producing 5 different fuel compositions. The first of the tested compositions was the binary blend without ethanol, while the others had increasing alcohol content. The fifth composition used 15% of ethanol but had a 0.4% of the additive di-tert-butyl peroxide mixed in the main fuel. The addition of ethanol led to a reduction in diesel fuel consumption, although the overall energy expenditure was increased. The emissions profile showed a consistent reduction in NOx emissions and opacity with the addition of ethanol, breaking up the traditional inverse relationship between NOx and PM emissions, but shown an increase in CO and THC emissions. The energy analysis showed a decrease in engine efficiency with the addition of ethanol. The use of the additive showed a slight increase of engine efficiency and the reduction of the CO and THC emissions. There was a significant reduction in the air intake temperature with the use of ethanol, suggesting that part of the reduction of NOx may be attributed to this temperature reduction. It was proven that the ethanol addition can be an important method to reduce the amount of NOx in the exhaust gases of diesel engines.

Introduction

Biodiesel is a promising substitute for mineral diesel as it is renewable, non-toxic, easily biodegradable and contains physicochemical properties that are very close to that fossil fuel. Therefore, biodiesel has been widely proposed as a substitute for mineral diesel fuel, presenting good results in terms of flammability and lower environmental impacts of exhaust emissions in compression ignition engines. This fuel virtually does not generate residues of sulfur and has lower emissions pollutants due to the presence of the oxygen in its molecules (Candeia et al., 2009; Janaun and Ellis, 2010 and Lin et al., 2007). However, the use of biodiesel leads to a consistent increase in emissions of nitrogen oxides (NOx) (Lapuerta et al., 2008). This can be an obstacle to future growth in the biodiesel market and its application in engines (Zhu et al., 2011). Several solutions to this problem are generally used, such as exhaust gas recirculation (EGR), a change in the injection parameters, the reduction of the compression ratio and/or the intake pressure (turbocharging pressure) (Martins, 2013), but the addition of oxygenated compounds such as ethyl alcohol or dimethyl ether (DME) may also produce the same result (Jie et al., 2010 and Yilmaz and Sanchez, 2012).

The possibility of using oxygenated compounds such as ethanol motivates the study of these compounds in diesel engines, considering that this fuel can be produced from biomass on a large scale in countries with high agricultural potential, such as Brazil. As Montero and Stoytchevap (2011) commented, ethanol can increase the percentage of biofuel in the blend as well as improve the gas emissions profile when compared to mineral diesel.

The use of ethanol as a fuel for diesel engines is justifiable but it requires some dedicated solutions. In general, ethanol injection in diesel engines can be achieved by using ethanol-diesel blends, by diesel and ethanol twin direct injection or by the ethanol fumigation (by a carburetor or low-pressure injection system) in the air intake of the engine.

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The use of ternary mixtures of diesel, biodiesel and ethanol can simultaneously reduce NOx and particulate matter (PM) emissions (Zhu et al., 2011). However, this requires a limited amount of ethanol (up to 10% v/v) due to the miscibility problems of ethanol in diesel fuel, unless solubility additives are used (Lapuerta et al., 2007). Guarieiro et al. (2009) tested the stability of 18 binary and ternary blends of diesel, vegetable oils, biodiesel and ethanol for a minimum period of 90 days and found that the mixtures of ethanol with purity of 95% were not stable due to the polarity of the water molecule present in this compound. They also found that the blends were stable up to fractions lower than 10% of ethanol with purity of 99.5%.

The use of ethanol fumigation technique (low-pressure injection or carburation) in the air intake of the engine requires few modifications in the engine. Lu et al. (2008) studied the effect of ethanol when injected into the air intake of the engine with various percentages of ethanol and several equivalence ratios of the overall fuel/air mixture. The results showed a substantial reduction in the NOx emissions when ethanol was used, but with an increase of the total unburned hydrocarbons (THC) and carbon monoxide (CO) emissions. Tsang et al. (2010) tested ethanol fumigation in a 4-cylinder direct injection diesel engine, observing the reduction in overall NOx, opacity and particulate matter mass, but with the increasing in the THC and NO_2 emissions.

This work shows and discusses the results of performance tests with ethanol fumigation in the air intake of a diesel engine by the use of a low-pressure electronic injection system. The aim is to reduce NOx and particulate matter (PM) emissions while increasing the percentage of renewable fuel in the combustible mixture. The timing of the ethanol injection was obtained by monitoring the diesel injection pressure, simplifying the adaptation of such a system in diesel engines with mechanical injection.

Materials and methods

Engine-electric generator

The tests were conducted in a single cylinder 4-stroke diesel engine, manufactured by Agrale® with indirect injection and maximum power of 10 HP (see Fig. 1). Engine speed and load were controlled by a three-phase generator manufactured by Kohlbach®. Engine and generator specifications are described in Tables 1 and 2. The tests were run in conditions of fixed engine load (1580 ± 10 W) and speed (1800 ± 5 rpm), as the amount of main fuel supplied by the high-pressure injector was being regulated by the generator.

Fig. 1. Engine tested.

Table 1
Engine properties.

Property	Engine
Maximum power (kW)	7.6 (NBR-1585)
Speed (rpm)	1800–2500
Compression ratio	20:1
Number of cylinders	1
Injection type	Indirect
Injection pressure (MPa)	15
Sweep volume (cm^3)	567

Electronic management system for ethanol injection

An electronic system was developed to allow ethanol injection in the intake manifold of the engine. It was decided to inject once per engine cycle, so there was the need for synchronization with the cycle. As it was difficult to access the camshaft to place a sensor in it, it was decided to use the high-pressure diesel injection pulse to synchronize with the ethanol injection. The flowchart of the electronic control system is shown in Fig. 2.

The developed electronic central unit (ECU) has two printed circuit boards (PCBs), each one controlled by its respective microcontroller. PIC microcontrollers manufactured by Microchip® model 18 F4520, with maximum frequency of 40 MHz, 13 analog inputs with 10-bit digital resolution and two priority levels for external and internal interruptions were used. The first printed circuit board (PCB-1) was responsible for the engine rotation velocity measurement and displayed this value on a seven segments display. It also measured the inlet temperatures before and after the ethanol injection and displayed these temperatures on an alphanumeric LCD display and transmitted all the measured data to a PC.

The second printed circuit board (PCB-2) was responsible for setting the timing and duration of the ethanol injection. This was achieved by detecting the pressure pulse from the injection of diesel fuel and doing the required calculations for achieving the required timing and duration of the injection pulse. The value of engine speed was discretized at regular intervals of 30 rpm and converted into an internal protocol for communication between the two microcontrollers.

The temperature measurement was obtained using NTC thermal sensors. An amplification circuit was used to measure the temperatures with sensitivity up to 0.12 V/°C in the range of 15 to 30 °C (range of expected temperatures).

There was communication with a PC in order to allow the remote configuration of the injection settings, the advanced angle of start of injection relating to the top dead center (TDC at the beginning of the intake stroke) and the injection duration (in milliseconds). All circuits were properly packed in a neat plastic case. Front and back views of this case are shown in Fig. 3.

Sensors and calibration

RPM Sensor

An inductive sensor model LM-12-3004-PA manufactured by JNG sensors® was used for the rotation speed measurement. The sensor was mounted on the engine base in orthogonal position to the shaft,

Table 2
Generator properties.

Property	Generator
Maximum power (kVA)	6.0
Nominal speed (rpm)	1800
Voltage	3~/220 V/60 Hz

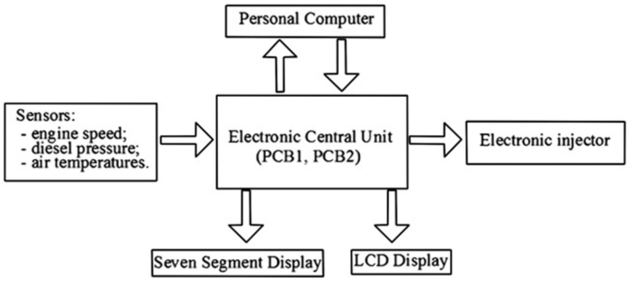

Fig. 2. Flowchart of electronic management system.

Fig. 3. Front (a) and back (b) views of ECU case.

where two metallic plates diametrically opposed were fixed. This tachometer was calibrated in order to validate its reading with a digital portable photo-tachometer (TC-5035 model), manufactured by ICEL®, with digital resolution of 0.1 rpm. The maximum percentage error was 0.33% at 1500 rpm, making the use of correction factors in order to evaluate the rotation speed during the ethanol injection unnecessary.

Diesel injection pressure sensor

In order to monitor the diesel pressure in the injection line, a low-cost sensor 3PP6-12 model, Mobil line, manufactured by Ideal Sensors® was used. This sensor has a range of 0–30 MPa, with a voltage supply of 5.0 VDC and a proportional output to pressure in the range of 0.5 to 4.5 VDC. This sensor was calibrated in the static form using nitrogen gas between 0 and 30 MPa at regular intervals of 5 MPa. A Bourdon pressure gauge with an accuracy of 1% was used as the validation system. By adjusting the calibration line, it was possible to obtain the relationship between the output voltage in volts and the pressure in MPa.

The input signal from the diesel pump pressure sensor of the engine presented a continuous voltage level of approximately 0.5 V. Thus, the pressure pulses were conditioned by a specific circuit in order to become rectangular pulses and be interpreted as a digital signal by the microcontroller of the PCB-2.

Air intake temperatures

For the measurement of the air intake temperatures, two NTC sensors provided by Digikey model AL03006-1248-73-G1 with nominal resistance of 2.0 kΩ at 25 °C were used. The calibration curve of the sensors was obtained in the range of 5.0 °C to 80.0 °C in a thermostatic bath, as described by Ferreira and Pepe (2008).

Assembly details

The assembly details of the test system are shown in Fig. 4, with the positioning of the pressure sensor in the diesel high-pressure feed line of the injection system, as well as both temperature sensors used to measure the temperature reduction in the intake air.

A low-pressure electric injection pump and a pressure regulator adjusted to 0.25 MPa were used for the ethanol injection system. A Bourdon gauge type measured the ethanol pressure during the tests.

An electronic injector manufactured by Honda®, KVB-T01 model, with a 12VDC supply voltage, was used for the ethanol injection. This injector is standard on Honda motorcycles of 108 cm^3. The electronic injector was installed in the intake pipe at a distance 100 mm away from the inlet of the engine, on a curved section in order to increase the turbulence and thus ensure a better air and ethyl alcohol mix.

Initially, the system was tested in order to verify the injection delay, which was done with a reflective-type optical sensor, a 45FSL-2LHE model, PNP type, manufactured by Allen Bradley, with a wavelength of 660 nm and a response time of 30 μs. This sensor was fixed to the engine housing. A reflective tape was attached to the flywheel of the engine so that when the engine was at top dead center (TDC), the tape was positioned in front of the optical sensor. These initial tests were conducted at 3 different speeds: 1700, 1750 and 1850 rpm, considering that the generator operates at a rated speed of 1800 rpm. The injection was programmed in order to occur exactly at the TDC. The maximum delay verified between the start of the injection and the top dead centre was by 840 μs.

Fig. 5 shows the details of engine and generator mechanical coupling and locations of air temperature sensors.

Instrumentation

The mass fuel flow normally introduced by the high-pressure injection pump and ethanol mass flow rate were determined by the

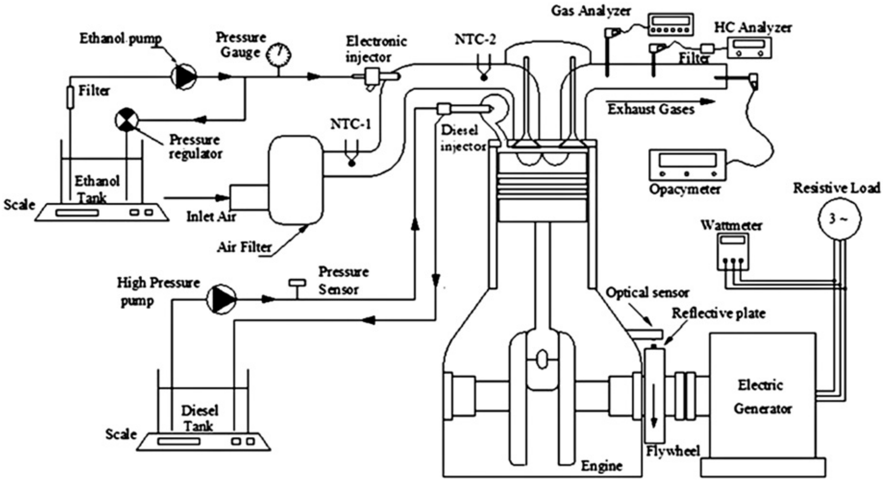

Fig. 4. Details of system for double fuel injection.

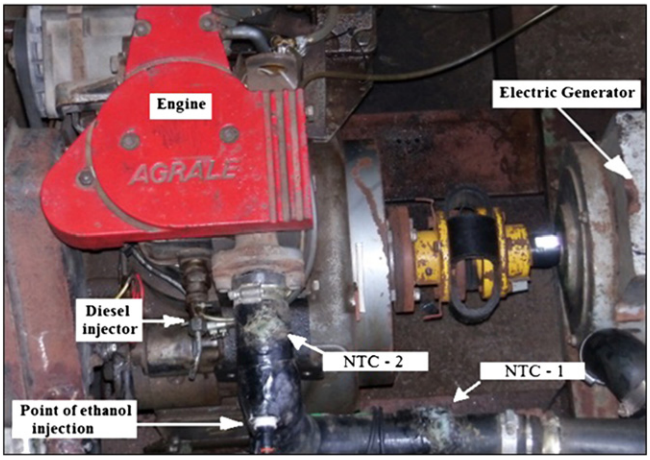

Fig. 5. Engine and generator coupling.

gravimetric method using digital scales. For each fuel five cycles of consumption measurements were performed with a sampling time of 30 min. The generator power was obtained using a digital wattmeter with a sampling frequency of 3 Hz.

The exhaust gases were determined by three gas analyzers. One of them could evaluate the concentrations of CO and NOx in ppm, while the others were able to measure the opacity and concentrations of unburned hydrocarbons (THC). For the latter, fine filters were placed in the collection line in order to retain particulate material from the discharge pipe of the engine. A total of 25 measurements of gas emissions and exhaust gas opacity were done for each fuel. The main characteristics of the employed instruments are shown in Table 3.

Fuels and additive

The class S-500 diesel and soybean biodiesel blend used to make up D70B30 (70% petroleum diesel and 30% biodiesel) were kindly donated by Petrobahia (Petroleum Distributor of Bahia S.A.). The ethyl alcohol 99.3% purity was supplied by Química Moderna®. Table 4 shows the main characteristics of the raw materials used for the fuel compositions. All the tests were carried out with the same batch of fuel, therefore making sure the fuel specification was always the same.

Table 3
Properties of the main instruments.

Property	Instrument model (manufacturer)	Measurement uncertainty
Ambient temperature	Thermometer AN-3070 (Icel)	±(3% + 0.2 °C)
Relative humidity	Digital Thermohygrometer HT-208 (Icel)	±3%
Exhaust gas analysis (CO_2, NOx and CO)	Gas Analyzer Tempest-100 (Telegan Gas Monitoring)	±2%
Electric power (W)	Digital Wattmeter AW-4700 (Icel)	±(3% + 5 °C)
Fuel consumption (diesel + biodiesel)	Digital scale 9094 (Toledo)	±2 g
Fuel consumption (ethanol)	Digital scale BK5002 (Quimis)	±0.2 g
Exhaust gas analysis (THC)	Gas Analyzer PC-Multigás (Napro)	±3%
Opacity	Opacimeter NA-9000 (Napro)	±3%
Exhaust gas temperature	Digital Thermometer/type K thermocouple MT-525 (Minipa)	±(3% + 0.2 °C)

Table 4
Raw material properties for fuels.

Property	Fossil diesel	Soybean biodiesel	Ethyl alcohol
Typical composition	$C_{9.84}H_{17.95}$	$C_{18.74}H_{34.43}O_2$	C_2H_6O
Molecular weight (kg kmol^{-1})	136.3	291.8	46.1
Density at 20 °C	0.853	0.870	0.790
Latent heat of vaporization (kJ kg^{-1})	270	200	840
Cetane number	48	56	6
Lower heating value (kJ kg^{-1})	42,820	36,395	28,300

Five different fuel compositions were tested. The tests started with a binary mixture of 70% v/v mineral diesel and 30% v/v biodiesel (D70B30) without any ethanol injection. After that, three compositions were tested with increasing ethanol content (D70B30-E5, D70B30-E9 and D70B30-E15) added through the air intake pipe by controlling the injection time of the electronic injector.

The premixed ratio (PI), as described in Lu et al. (2008), for each fuel composition was determined by Eq. (1).

$$\mathrm{PI}(\%) = \left(\frac{m_e \mathrm{LHV}_e}{m_e \mathrm{LHV}_e + m_{d_b} \mathrm{LHV}_{d_b}}\right).100 \qquad (1)$$

where

m_e is the is the mass of ethanol injected during each cycle (kg)
m_{d_b} is the is the mass of diesel-biodiesel blend in each cycle (kg)
LHV_e is the ethanol lower heating value (kJ kg^{-1})
LHV_{d_b} is the lower heating value of diesel-biodiesel blend (kJ kg^{-1}).

Table 5 shows the ethanol premixed ratio for each fuel composition as well as the volumetric ratio of alcohol.

The di-tert-butyl peroxide (DTPB), similarly to the 2-ethylhexylnitrate (2-EHN), is a known accelerator for spontaneous ignition and has been used as a cetane improver of diesel fuel in recent researches (Lee et al., 2010). Table 6 shows the properties of (DTPB). The use of DTBP additive in a ratio of 5000 ppm (0.5%) can increase the cetane number of standard diesel fuel in 5.5, from 48.0 to 53.5 (Epa United States Environmental Protection Agency, 2004).

In this study, the additive di-tert-butyl peroxide, supplied by Merck®, was added at a ratio of 0.4% in volume in order to increase the cetane number of the binary blend (D70B30) setting the fifth fuel composition (D70B30-A0.4-E15).

Energy analysis

The tests took place within an ambient temperature of 30 ± 2 °C, while the relative humidity was constant at 55 ± 2 %. For each cycle, the engine was heated for 30 minutes. Lubricating oil was substituted for each fuel composition. The lubricating oil was Lubrax CI-4 (15 W-40) manufactured by Petrobras®.

Table 5
Energy and volumetric ratios of ethanol for each fuel.

Fuel	Ethanol premixed ratio (%)	Ethanol volumetric ratio (%)
D70B30	0.0	0.0
D70B30-E5	3.3	5.1
D70B30-E9	6.1	9.2
D70B30-E15	10.2	15.2
D70B30-A0.4-E15	10.2	15.2

Table 6
Properties of DTBP (di-tert-butyl peroxide).

Property	Value
Composition	$C_8H_{18}O_2$
Flash point (°C)	6
Density (20 °C)	0.79
Autoignition temperature (°C)	165

The combustion reaction is described in Eq. (2). It was accepted that the combustion products contain some CO and unburned hydrocarbons (HC, as diesel, biodiesel and ethanol).

$$x\,C_{9.84}H_{17.95} + y\,C_{18.74}H_{34.43}O_2 + z\,C_2H_6O + a(O_2 + 3.76N_2) \rightarrow b\,CO_2 + c\,CO +$$

$$d\,H_2O + e\,O_2 + f\,NO_2 + g\,NO + h\,C_{9.84}H_{17.95} + i\,C_{18.74}H_{34.43}O_2 + j\,C_2H_6O + k\,N_2 \qquad (2)$$

The x, y and z coefficients are the molar ratios of the diesel, biodiesel and ethanol for each fuel composition in one mol of fuel ($x + y + z = 1$), while the other coefficients ($a, b, c, d, e, f, g, h, i, j$ and k) were obtained using the measured data, and the mass balance of each element.

For the energy analysis, the following assumptions were made:

- The engine operates at steady state.
- The control volume includes the engine and the generator.
- The power consumption by the low-pressure injection system was not considered.
- The inclusion of additive does not modify the heating value of the fuel.
- The blends are ideal solutions.
- The kinetic and potential energy effects were not taken into account.
- The atmospheric air composition was assumed as 21% oxygen and 79% nitrogen on a molar basis.
- Since the air stream is very close to the standard reference state, energy input accompanying it was ignored.
- The SO_2 emissions were not considered.

The engine flows crossing the control surface are presented in Fig. 6.

The balance among the energy flows entering and leaving the control surface can be report in Eq. (3).

$$E_{in} = \dot{n}_c\,|\overline{LHV}| = |\dot{Q}_{CV}| + \dot{W}_{CV} + \dot{E}_{ex} \qquad (3)$$

where

$\dot{E}_{in}$	is the energy flow entering into the control volume [kW];
$\dot{n}_c$	is the molar flow of equivalent fuel (diesel + biodiesel + ethanol) [kmol/s];
$\overline{LHV}$	is the molar lower heating value of equivalent fuel (diesel + biodiesel + ethanol) [kJ/kmol];
$\dot{Q}_{CV}$	is the heat flow crossing the control surface (losses) [kW];
$\dot{W}_{CV}$	is the generator electrical power [kW];
$\dot{E}_{ex}$	is the energy flow following the exhaust gas [kW]

Table 7
Results for air intake and exhaust gas temperatures.

Fuel	Upstream temperature (°C)	Downstream temperature (°C)	Exhaust gas temperature (°C)
D70B30	29.5 ± 0.2	29.5 ± 0.2	264.6 ± 0.4
D70B30-E5	29.5 ± 0.3	19.2 ± 0.3	249.5 ± 0.5
D70B30-E9	30.8 ± 0.2	15.0 ± 0.2	241.8 ± 0.5
D70B30-E15	30.7 ± 0.2	14.2 ± 0.2	234.5 ± 0.5
D70B30-A0.4-E15	31.0 ± 0.2	14.1 ± 0.2	237.5 ± 0.5

For the determination of heat exchanged between the engine and the environment, the first law of thermodynamics was applied to the engine in a steady state the mass and energy rate balances for the single-inlet, single-exit can be used to obtain the following relationship on a per mole of fuel basis (see Eq. (4)) (Canakci and Hosoz, 2006 and Moran and Shapiro, 2009).

$$\frac{\dot{Q}_{CV}}{\dot{n}_c} - \frac{\dot{W}_{CV}}{\dot{n}_c} = \left(\overline{h}_p - \overline{h}_R\right) = \sum_P n_{out}{}^{out}\left(\overline{h_f^0} + \overline{\Delta h}\right)_{out} - \sum_R n_{in}{}^{in}\left(\overline{h_f^0} + \overline{\Delta h}\right)_{in} \qquad (4)$$

where

$\overline{h}_p$	is the molar combustion enthalpy in combustion gases per mole of fuel (kJ/kmol);
$\overline{h}_R$	is the molar combustion enthalpy in reactants per mole of fuel (kJ/kmol);
n_{out}	is the coefficient for each product in the reaction equation;
n_{in}	is the coefficient for each reactant in the reaction equation;
$\overline{h_f^0}$	is the enthalpy of formation for each compound in the combustion equation;
$\overline{\Delta h}$	is the enthalpy change due to a change of state at a constant composition.

Enthalpy change was determined by Eq. (5) (Canakci and Hosoz, 2006).

$$\overline{\Delta h} = \overline{h}(T) - \overline{h}(T_{REF}) \qquad (5)$$

where

$\overline{h}(T)$	is enthalpy at the temperature of the considered reactant or product (kJ/kmol);
$\overline{h}(T_{REF})$	is enthalpy at the reference temperature (kJ/kmol);

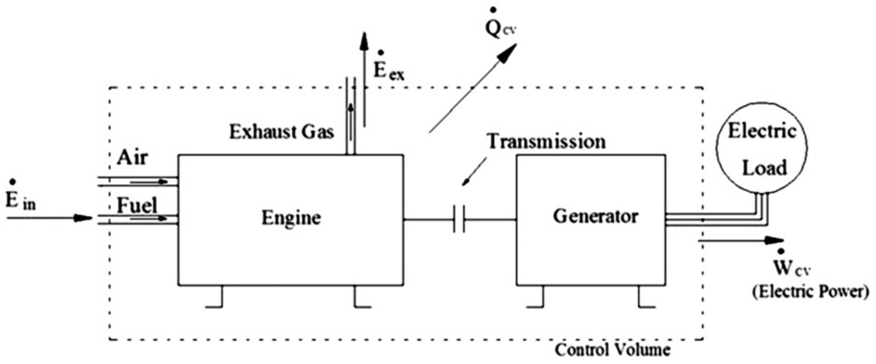

Fig. 6. Engine flows in the control volume.

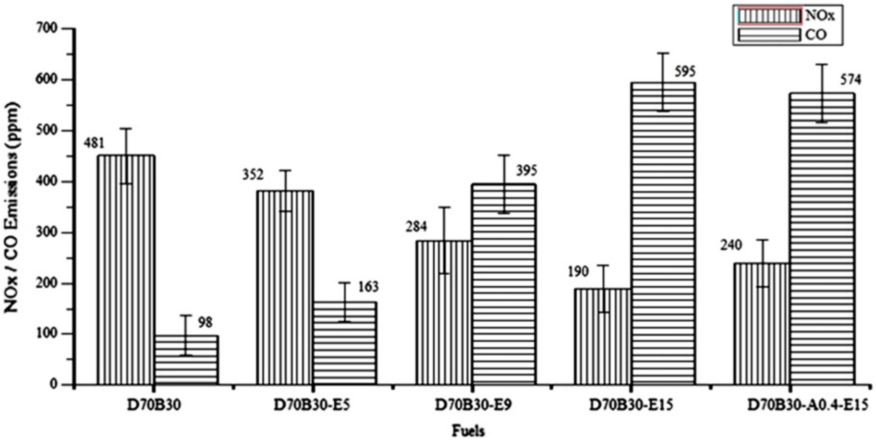

Fig. 7. CO and NOx emissions for each fuel.

Formation enthalpies of the fuels were determined from the general reaction equation for the complete combustion. The energy in the exhaust gas $(\dot{E}_{\mathrm{ex}})$ was determined by Eq. (3).

The engine-generator energy efficiency (ε) defined as the ratio of useful energy produced by the generator and the contained energy in the consumed fuel was determined according to Eq. (6).

$$\varepsilon(\%) = \left(\frac{\dot{W}_{\mathrm{vc}}}{\dot{n}_{\mathrm{c}}|\mathrm{LHV}|}\right)\cdot 100 \tag{6}$$

Results and discussion

The average results for air intake temperature reduction, exhaust gas temperature, exhaust gas emissions, fuel consumption and energy analysis are shown in the next section. The ethanol injection was scheduled to start 5° after top dead center (TDC), thus respecting the delay for shutting down the exhaust valve of the engine. The measurement uncertainties presented in the next sections were evaluated for a reliability of 95%.

Air intake temperatures and exhaust gas temperature

The measured values for the air intake temperatures at the upstream and downstream points of the injection and the exhaust gas temperature for each fuel tested are shown in Table 7. As can be observed, the use of ethanol resulted in a significant reduction in the air intake temperature caused by the high latent heat of vaporization of this fuel, as shown in Table 4. The exhaust gas temperature follows a similar trend to that observed by post-injection temperature. However, the reduction in exhaust gas temperature is higher than the reduction measured in the intake air temperature. This may be partly attributed to the fact that a part of ethanol only vaporizes into the combustion chamber.

Exhaust gas emissions

NOx and CO

The results for nitrogen oxides (NOx) and carbon monoxide (CO) emissions are shown in Fig. 7. NOx emissions are mostly related to the maximum temperatures obtained within the combustion chamber during combustion, but oxygen in the fuel may also increase the NOx emissions in diesel engines. As it can be seen in Fig. 7, the addition of ethanol caused a significant reduction in NOx emissions. This reduction can be partly attributed to the cooling effect produced by the high latent heat of the ethanol. But the addition of a fuel with a very low cetane number is expected to significantly increase ignition delay, so producing the peak pressure at a later stage, resulting in a decreasing of combustion chamber pressure and temperature as discussed by Park et al. (2011). This author observed that the increasing of ethanol content in diesel oil increases the ignition delay. As the ethanol is supplied not mixed with the injected fuel (diesel + biodiesel), the oxygen in ethanol molecule probably does not participate on the spray combustion and, therefore, does not contribute to increase the NOx production.

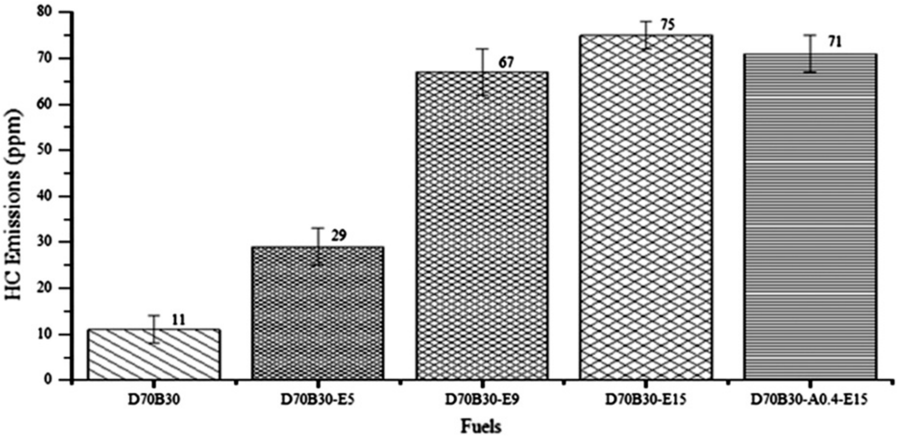

Fig. 8. Results for THC emissions.

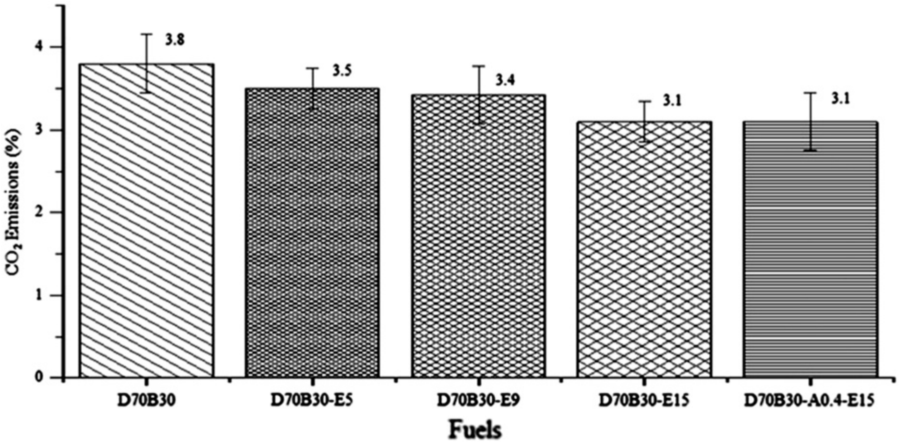

Fig. 9. Results for CO_2 emissions.

Additionally, the high latent heat of vaporization of ethanol induces temperature reduction in the combustion chamber, as herein measured and as described by Zhu et al. (2011) who tested the effect of ethanol in biodiesel blends. According to other authors (Randazzo and Sodré, 2011), since the small low heating value of ethanol in comparison with diesel oil, higher fuel amounts are required to produce the same power from the engine, thus intensifying fuel vaporization and reducing the temperatures attained in the combustion chamber, thereby reducing the NOx emissions.

Carbon monoxide is generally formed when the engine operates in a rich environment of fuel/air mixture. The carbon monoxide emissions in Fig. 7 show a consistent increase with the addition of ethanol in the fuel composition probably due to the fact that the ethanol was pre-mixed in a very weak mixture that may not burn away from the direct fuel spray. Also, the normally rich region of the fuel spray in air is not injected in a mixture air-ethanol, therefore further enriching this region.

DTBP does not have nitrogen in its composition such as others cetane improvers (2-EHN), but the induced reduction of the ignition delay of the combustion will increase the combustion pressure and temperature. This may be the reason why the use of the additive led to a very slight reduction in carbon monoxide emissions and a slight increase in NOx emissions.

Unburned hydrocarbons

In general, unburned hydrocarbons can be formed in diesel engines by incomplete combustion, rich combustion operation, deposits of hydrocarbon on the walls of the combustion chamber or from the fuel left in the interior of the injector tip (Martins, 2013). The results for the total unburned hydrocarbons (THC) emissions are shown in Fig. 8, where an increase in THC emissions with the addition of ethanol can be noticed. As described by Lu et al. (2008), probably during the intake and compression strokes, part of the ethanol pre-mixed with air is kept into the clearance of piston ring and dead zone of the combustion chamber. By having a weak ignitability, these mixtures are only partially oxidized or no oxidized at all during the combustion process, favoring the formation of unburned hydrocarbons during the expansion stroke and during the exhaust blow out (Martins, 2013). The use of the additive resulted in a slight reduction in hydrocarbon emissions probably by promoting compensation for the decrease in ignition delay.

CO_2

Fig. 9 shows the results for CO_2 emissions measured in % (v/v). A noticeable reduction in CO_2 emissions with increasing ethanol content can be seen. It occurs because the ethanol has a lower C/H ratio when compared to fossil diesel and biodiesel fuels. These results are in agreement with those obtained by Randazzo and Sodré (2011).

Opacity

The opacity of the exhaust gases of diesel engines is mainly caused by the formation of particulate material composed of solid carbon soot generated in rich mixture zones inside the cylinder during combustion reaction.

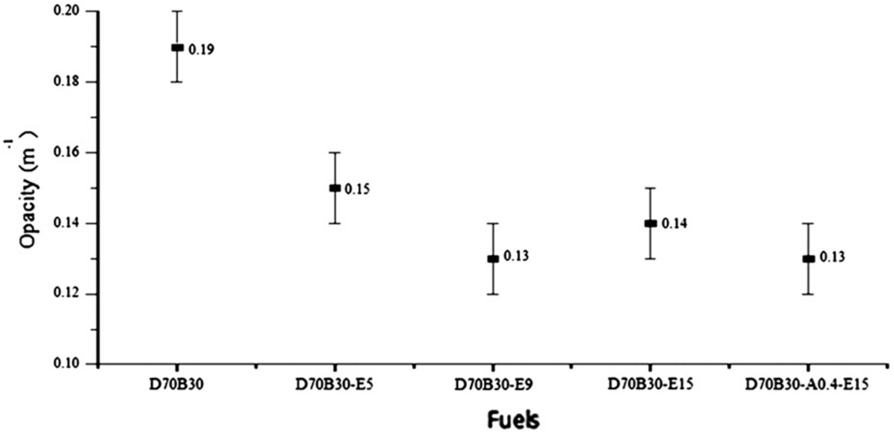

Fig. 10. Opacity of exhaust gases.

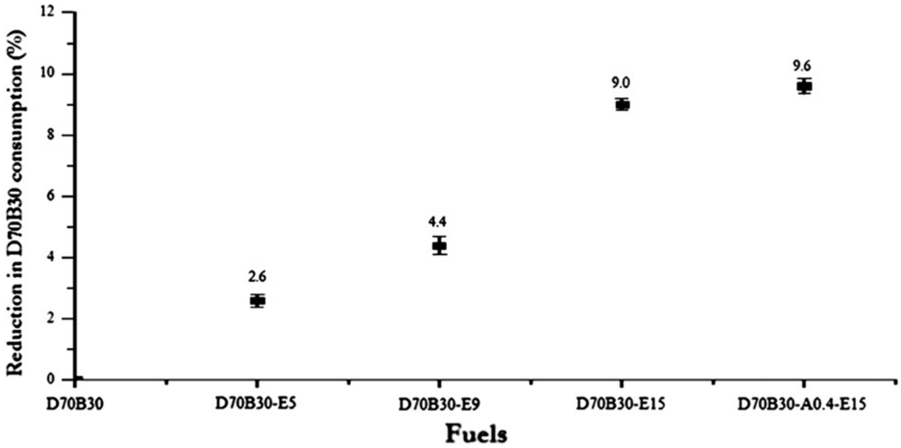

Fig. 11. Reduction in diesel-biodiesel blend consumption.

Table 8
Results for energy analysis.

Energy flow (kW)	Fuels				
	D70B30	D70B30-E5	D70B30-E9	D70B30-E15	D70B30-A0.4-E15
Input energy	9.29	9.35	9.44	9.46	9.41
Electric power	1.58	1.58	1.58	1.58	1.58
Losses	4.71	4.74	4.72	4.68	4.63
Exhaust gas heat	3.00	3.03	3.14	3.20	3.20

The opacity measured by the coefficient of light absorption is shown in Fig. 10. With the increased ethanol content, it is possible to note a reduction in the opacity until a concentration of 9% of ethanol. A further increase of ethanol to 15% did not produce further improvements. The use of ethanol can lead to oxygen-rich areas during the combustion process within the cylinder. According to Zhu et al. (2011), ethanol can reduce the soot precursors due to the production of OH radicals by the ethanol, thereby reducing the formation of particulate material. The incorporation of the additive led to a slight reduction in the opacity of the exhaust gases, although the measurement uncertainty makes reliable conclusions impossible.

Tsang et al. (2010) verified that the opacity reduction with the use of ethanol is less noticeable under low load conditions, which were the conditions used in the present test.

Fuel consumption

Fig. 11 shows the percentage reduction in diesel–biodiesel blend consumption as a result of the introduction of ethanol in the air intake pipe of the engine for each fuel. With increasing ethanol levels, the required amount of fuel (diesel + biodiesel) to maintain the same level of power was obviously reduced.

The use of the additive produced a small further reduction in the consumption of diesel and biodiesel. The improvement in the combustion conditions may have contributed to a more efficient combustion reaction, reducing the required flow rate of the injected binary blend (D70B30).

Energy analysis

Table 8 shows the energy flows entering and leaving the engine for the five tested fuels.

There is a steady increase in the input energy as the ethanol injection level increases. As said before, the mass flow of high-pressure fuel (D70B30) was reduced, but this reduction in energy content is lower than the surplus energy given by the ethanol. This can only be explained by the fact that the combustion efficiency deteriorated as the ethanol was added in larger percentages.

Although there is a decrease in the discharge temperature when ethanol was injected (see Table 7), the energy contained in the exhaust gases shown a slight increase due to the increase of carbon monoxide and unburned hydrocarbons present in the exhaust gases. The

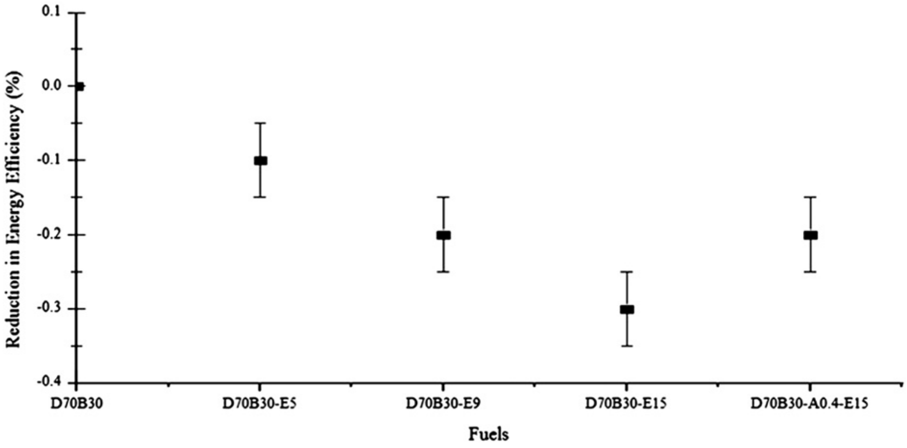

Fig. 12. Reduction in energy efficiency compared to D70B30 fuel.

incorporation of the additive did not modify the energy following the exhaust gas since the slight reduction in THC and CO emissions is compensated by the slight increase in exhaust gas temperature.

The results for the reduction in engine energy efficiency for each fuel tested when compared to B70D30 are shown in Fig. 12. The efficiency calculation refers to overall efficiency of the engine-generator set, thus taking into account the losses of the mechanical transmission added to those caused by the Joule effect in the generator, its controller and electrical power cables.

It is possible to note a reduction in energy efficiency with increasing ethanol content in the fuel composition. As said before, this probably occurred because of the low cetane number of ethanol, which resulted in reduced engine efficiency. The use of the additive resulted in a small improvement in energy efficiency.

Conclusions

The use of ethanol can be a valuable alternative to reduce NOx emissions when using biodiesel as fuel in compression ignition engines. Furthermore, by reducing diesel fuel consumption in these engines, ethanol can help to reduce the consumption of fossil fuel.

This paper presents a description of a system which introduces ethanol mixed with the air intake pipe of the engine avoiding the inconvenient separation of phases when using the ternary blends of diesel, ethanol and biodiesel. Furthermore, the fumigation technique enables the use of alcohol with a higher water content, which would not be possible in case of using blends because water promotes phase separation in such mixtures.

The electronic injection system described in this paper can be implemented with minimal changes in engine construction, considering that the detection of the synchrony of the oil pressure pulse prevents the implementation of a specific sensor on the camshaft.

The results for the performance, exhaust gas and intake air temperatures and the emissions of a diesel engine operating with a binary blend of diesel and biodiesel (D70B30) which was supplemented by the low-pressure injected ethanol are presented.

The results for the air temperature measurements at the upstream and downstream points of injection showed air cooling with ethanol injection up to 17.0 °C. The high latent heat of the ethanol vaporization accounts for this significant reduction which may significantly change the combustion characteristics.

The energy analysis showed a decrease in engine efficiency with the addition of ethanol and the injection of a volumetric ratio of 15.2 % ethanol (10.2 % of ethanol energy ratio) led to a volumetric reduction of 9 % in diesel–biodiesel consumption (reduction of 8.6 % in binary blend energy consumption), thus the mass flow of high pressure fuel (D70B30) was reduced, but this reduction in energy content is lower than the surplus energy given by the ethanol because the combustion efficiency deteriorated as the ethanol was added.

The CO and THC emissions increased with ethanol content. However, NOx emissions fell consistently coupled with a slight reduction in opacity, thus breaking the traditional inverse relationship between NOx and particulate matter emissions (PM).

The use of the additive di-tert-butyl peroxide as ignition improver led to a slight reduction in fuel consumption, CO and THC emissions. It is noteworthy that the amount of additive was minimal in these tests.

The results achieved by using a small amount of additive indicate the need for further research into new low cost additives that reduce ignition delay of the fuel composition in order to compensate for the low cetane number of ethanol.

The use of ethanol injected into the air intake can be a valuable method in the future for controlling NOx emissions in diesel engines when using blends with high levels of biodiesel, favoring the preservation of natural resources by reducing the consumption of mineral diesel oil.

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