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EXPERIMENTAL INVESTIGATION OF DIESEL-ETHANOL DUAL-FUEL COMBUSTION FOR AGRICULTURAL TRACTORS APPLICATION

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Abstract. *Dual-fuel combustion using biofuels is a potential technique both to reduce pollutant emissions and to increase overall efficiency in compression ignition engines. This work aimed to study experimentally the performance and emissions of a single-cylinder engine fueled with diesel-ethanol mixtures under the dual-mode dual-fuel technology. The tests were carried out at a point of the Non-Road Steady Cycle (NRSC), which is used in the homologation process of agricultural tractors in Brazil. Ethanol was port fuel injected using an electronic central unit while diesel was direct-injected by a mechanical fuel pump system. An injection trade-off considering the fuel´s energy content helped to achieve the best mixture proportion. In-cylinder pressure analysis was carried out in order to understand the combustion development and behavior. Gaseous (NO_x, HC, CO and CO_2) and smoke emissions were assessed for each dual-fuel test condition. Diesel-ethanol dual-fuel combustion showed an interesting potential to maintain satisfactory efficiency levels and to reduce NO_x and smoke pollutant emissions. The use of ethanol from sugarcane in diesel engines represent an advantage to reduce the carbon footprint and to beneficiate the Brazilian energy matrix.*

Keywords: *ethanol, diesel, dual-fuel engines*

1. INTRODUCTION

Brazil's fuel energy matrix is highly dependent on biofuels, as practically all national vehicles use biofuels in some proportion, being used pure or blended with other fuels. As examples, biodiesel that is blended in a 10% proportion with conventional Diesel or ethanol that is blended with gasoline or used in pure form.

Brazil is the most known case of success of a large biofuel use in transport, having cars that depend only on ethanol since the decade of 1970 and vehicles that run on both fossil and biofuels since the early 2000's. Brazil was a pioneer country when it comes to biofuel use in Diesel engines, being one of the first countries to invest in ways of producing fuel from vegetable oils.

The heavy-duty vehicles are responsible by almost half of the total CO_2 emissions from the road transport even representing just a small portion of the global vehicle fleet (García e*t al.*, 2019*)*. This fact concerns governments and encourage them to search for new ways to mitigate these emissions in compression ignition engines that are used in heavy applications.

In this context, the dual-fuel technology is an interesting alternative to promote a good balance between efficiency and pollutant emissions, allowing the use of conventional Diesel engines with just some modifications. According to Pedrozo and Zhao (2018), a diesel-ethanol dual-fuel combustion mode can reduce NO_x emissions up to 90% and can improve the combustion efficiency when compared with pure conventional diesel combustion. To achieve a dual-fuel system technology it is necessary the installation of a low-cost port-fuel injection system where the low reactivity mixture of air and ethanol is formed.

In this sense, it is possible to pursue a clean and renewable fuel matrix. The challenge today is to reduce its carbon footprint, developing better fuels and engine technologies that provide a better efficiency and less emissions. This paper shows a study based on the use of the dual-fuel technology with two of the most used fuels in the country, Diesel blended with biodiesel and ethanol, aiming to verify the possibility of both fuels combined in a compression ignition engine for agricultural tractors application.

2. MATERIALS AND METHODS

Tests were carried out in an experimental test bench developed at LMT-UNIFEI (Thermal Machines Laboratory – UNIFEI) with a single-cylinder compression ignition engine coupled to a hydraulic dynamometer as shown in Figure 1.

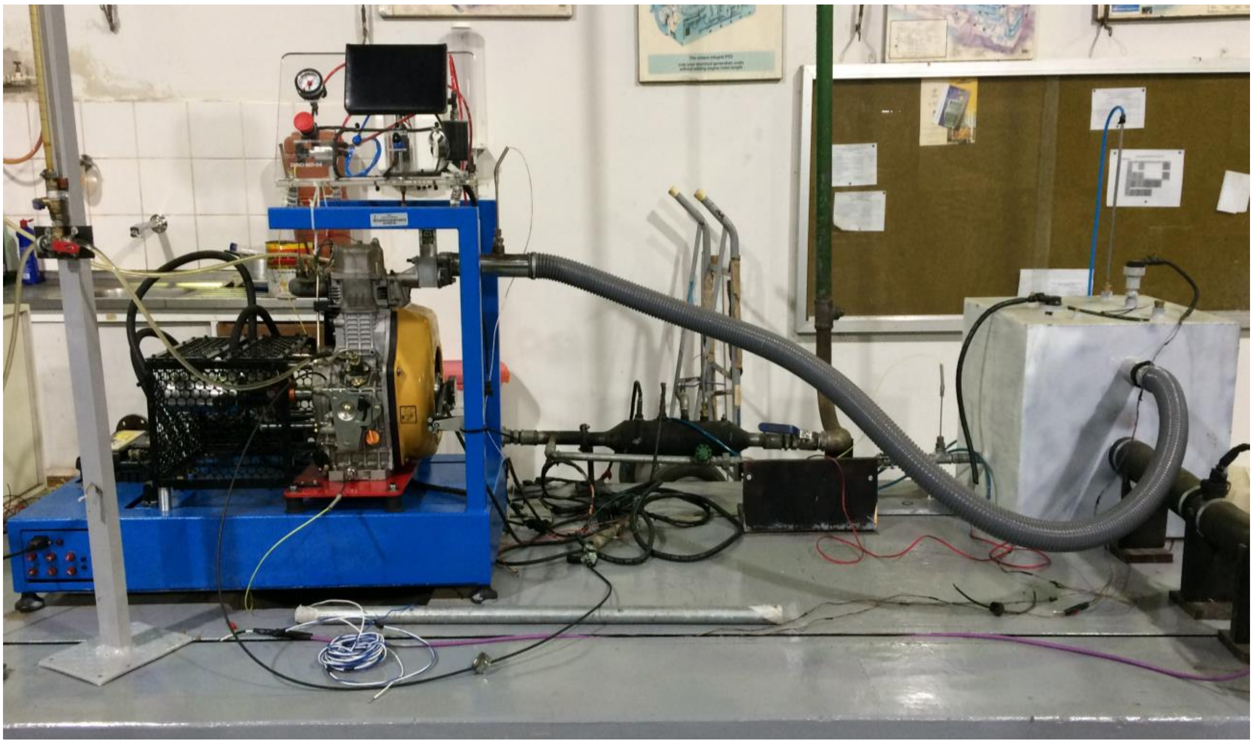

Figure 1. Experimental test bench developed at LMT-UNIFEI.

2.1 Engine and instrumentation

The chosen engine is a four-stroke single-cylinder compression ignition engine, naturally aspirated, air cooled, with diesel being direct-injected by a mechanical fuel pump system. Its application is usually for energy generation in small scales. The engine cylinder head has been modified to enable in-cylinder pressure instrumentation and measurement allowing a combustion analysis, while an electronic port-fuel injection system for dual-fuel technology use was developed and installed in the intake manifold. For port-fuel injection, a Bosch Injector with a maximum of 17mL flow was used. Table 1 presents engine specifications.

Table 1. Single-cylinder engine specification.

Property	**Description**
Type	4-stroke compression ignition
Displacement volume	418 cm^3
Stroke	72 mm
Bore	86 mm
Connecting rod length	118 mm
Crank radius	38 mm
Volumetric compression ratio	19.0:1
Number of valves	2
Intake/Exhaust valve diameter	36 mm/29 mm
Cooling	Air-cooled
Number of injector holes	4
Static start of injection (SOI)	22º±1º before TDC
Diesel injection pressure	19.6 MPa
Lubrification	SAE 15W-40 oil
Aftertreatment	None
Maximum torque	27 Nm(1)
Maximum power	7.1 kW(2)

(1) at 2800 rpm. (2) at 3600 rpm.

To measure the crankshaft torque a specified hydraulic dynamometer for internal combustion engines was used. The data acquired by the dyno bench is stored in own only software. Port-fuel injection characteristics are also controlled by the same software that controls the dyno. To synchro the engine and to identify the exact port-fuel injection timing, there is a speed sensor controlled by an electronic module with real time reading. The port-fuel injection software allows to control injection timing and fuel injection angle.

The instrumentation system using a Novus FieldLogger collects ambient temperature, exhaust temperature, intake temperature, oil temperature and exhaust pressure. There is also a Fueltech programmable datalogger module for intake air mass flow, air fuel ratio and intake pressure acquisition.

For temperature acquiring, PT100 and type K thermocouples are used. For exhaust and intake pressure measurement, a SBTP Sabi Control piezoresistive sensor and a Bosch piezoresistive sensor are used, respectively. The air mass flow sensor (Bosch HFM-5 thermal mass) was positioned before a chamber to equalize air pressure and reduce engine vibrations. A lambda sensor was used for measuring the air fuel ratio, which was positioned in the exhaust pipe. A Humirel piezoresistive sensor measures the humidity.

For combustion analysis, an Indimicro AVL module with an optical encoder and a piezoelectric sensor were used to collect crankshaft angle and combustion chamber pressure. All data is analyzed with real time monitoring Indicom software.

Pollutant emissions were measured by a NAPRO gas analyzer that collects the CO_2, CO, HC and NO_x emissions. Additionally, an opacimeter was used to check the smoke levels produced by the engine during the tests. A schematic diagram of the experimental test bench with all components is represented in Figure 2 and more details about the measurement device and its uncertainties are specified in Table 2.

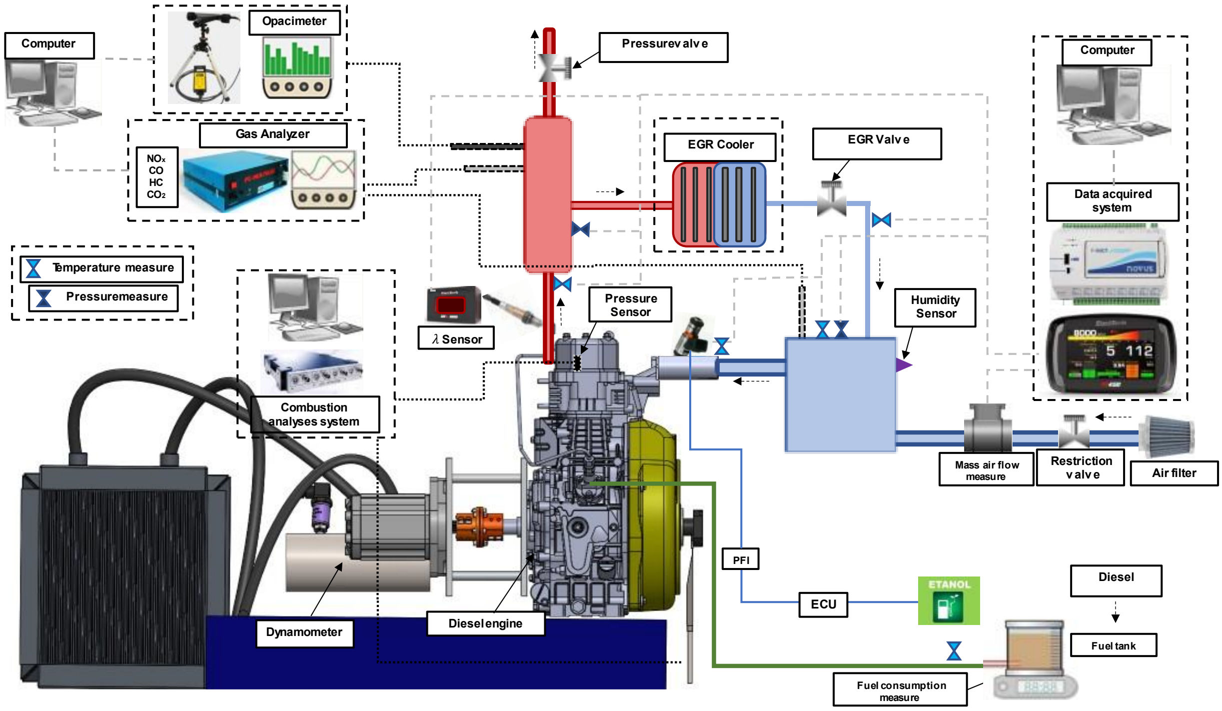

Figure 2. Schematic diagram of the experimental test bench.

Table 2. Measurement device specification.

Variable	Manufacturer	Sensor type/Model	Range	Accuracy
Brake torque [Nm]	INTECHNO	Electric dynamometer / MD04	-	-
In-cylinder pressure [bar]	AVL	Piezoelectric / AVL GH14DK	0-150	±0.3% FSO
Crank angle [°] / Speed engine [rpm]	AVL	Optical encoder / AVL 365	- 50-20000	<±0.03° ±3 rpm
Lambda λ [-]	Fueltech	Wideband sensor/Bosch 4.2	0.65-8	±0.009
Mass air flow [kg/h]	Bosch	Thermal mass/HFM 5	8-480	<±1.0%
Humidity [%]	Humirel	Piezoresistive/HPP811B002	0-100	±3.0%
Intake pressure [bar]	Bosch	Piezoresistive/Bosch 0281002437	0-5	±0.1%
Exhaust pressure [bar]	Sabi Control	Piezoresistive/SBTP	0-2	±0.1% FSO
Ambient temperature [°C]	TC	PT100	0-650	±0.8%
Intake temperature [°C]	TC	PT100	0-650	±0.8%
Exhaust temperature [°C]	TC	PT100	0-650	±0.8%
Oil temperature [°C]	TC	PT100	0-650	±0.8%
NO_x [ppm]	NAPRO	Gas Analyzer/PC Multigas	0-5000	±4.0%
HC [ppm]	NAPRO	Gas Analyzer/PC Multigas	0-20000	±3.0%
CO [% in volume]	NAPRO	Gas Analyzer/PC Multigas	0-15	±2.5%
CO_2 [% in volume]	NAPRO	Gas Analyzer/PC Multigas	0-20	±2.5%
Opacity [m^{-1}]	Altanova	Opacimeter/Smoke Check 2000	0-9.99	±0.5%

2.2 Fuel properties

The high reactivity fuel used in tests was conventional S10 diesel and Table 3 shows its properties. For low reactivity fuel, hydrated ethanol was used as the port-fuel injected fuel which has its properties presented in Table 4. Both fuels are regulated by ANP (Petroleum National Agency), with the conventional S10 diesel purchased during the tests having 10% biodiesel in its composition.

Table 3. S10 diesel properties and methods used.

Fuel Property	S10 Diesel	Method
C [% in mass]	85.33	ASTM D 5291 (Measured)
H [% in mass]	13.12	ASTM D 5291 (Measured)
O [% in mass]	1.55	ASTM D 5291 (Measured) and calculated
S [% in mass]	0.001	ISO 20884 (Measured)
Approximate molecular formula	$C_{14.22}H_{26.24}O_{0.19}$	Calculated
H/C ratio	1.84	Calculated
Stoichiometric air/fuel ratio	14.21	Calculated
Low heating value – *LHV* [MJ/kg]	41.03	ASTM D 240 (Measured) and calculated
Cetane number	50.5	ASTM D 613 (Measured)
Density at 20ºC [kg/m³]	846.6	ASTM D 4052 (Measured)
Kinematic viscosity at 40ºC [mm²/s]	2.88	ASTM D 445 (Measured)
Distillation		ASTM D 86 (Measured)
10% volume [ºC]	195.0	
50% volume [ºC]	262.0	
90% volume [ºC]	339.0	

Table 4. Hydrated ethanol properties.

Fuel Property	Hydrous ethanol
Composition [% volume]	C_2H_6O – 95.1% H_2O – 4.9%
Density at 1 atm and 15ºC [kg/m³]	805.2
Motor octane number	91.8
Research octane number	> 100
Auto ignition temperature [ºC]	363
Lower heating value – *LHV* [MJ/kg]	24.76
Adiabatic flame temperature [ºC]	~1920-2080
Flame speed [cm/s]	39
Stoichiometric air fuel ratio	8.36

2.3 Experimental procedure

The first step of experimental procedure was the system calibration where the port-fuel injector behavior was acquired to know the ethanol injection fuel consumption. A precision scale was used to measure the fuel mass consumption while the injector was opened by choosing an injection timing in the electronic dyno software. It showed a minimum injection timing of 0.7 ms. It was decided to start the calibration points in 0.7 ms until 2 ms, waiting for 60s to measure each mass value. Port-fuel injector pressure is controlled by a manometer and its value was maintained constant during all the tests at 3.8 bar pressure.

After port-fuel injection consumption calibration, tests were started. First of all, the engine worked during a time to set operation conditions such as stable temperatures and this happened at 85º±5ºC oil temperature. Diesel direct injection advance timing (from the engine mechanical fuel pump system) was maintained constant at 22º±1º before TDC, while ethanol port-fuel injection advance timing was also maintained constant at 180° before TDC during all test points. The chosen point and its conditions were decided because they represent a point of the Non-Road Steady Cycle (NRSC), which is used in Brazil for homologation process of agricultural tractors. A constant speed of 1800 rpm and 50% of the corresponded maximum torque were chosen as test condition (for pure diesel and all dual-fuel mixtures). In the exhaust pipe, a 100±5 mbar pressure was chosen and maintained constant in all test conditions following this same line to simulate real exhaust backpressure or/with aftertreatment systems.

Fuels were tested in 50% of the engine maximum load in a pure composition and also in the dual-fuel mode with four different blends energy fractions. Ethanol energy fraction for each blend represents the ratio between the energy content of ethanol and the total energy available in the fuel mixture. It was calculated using both fuel mass flow rate and its lower heating values according to Eq. 1. The ratios tested in this study were EF = 27%, EF = 31%, EF = 40% and EF = 44%.

$$EF = \frac{\dot{m}_{Ethanol} * LHV_{Ethanol}}{(\dot{m}_{Diesel} * LHV_{Diesel}) + (\dot{m}_{Ethanol} * LHV_{Ethanol})} \tag{1}$$

where $\dot{m}$ and LHV are the mass flow rate and lower heating value, respectively.

Fuel mass flow rate for ethanol was measured from the ethanol injection fuel consumption curve for each injection timing, as previously mentioned. For diesel, fuel mass flow rate was calculated using the intake air mass flow and the air fuel ratio – *AFR*-, which means the proportion between air and fuel present in a combustion process, showed in Eq. 2.

$$AFR = \frac{N_{AIR} * M_{AIR}}{N_{FUEL} * M_{FUEL}} \tag{2}$$

where N and M are the number of moles and molecular mass, respectively.

While for diesel *AFR* = 14.21, for ethanol *AFR* = 8.36. With these values and using the real timing lambda factor measured by lambda sensor, it is possible to substitute in Eq. 2 and solve it to conclude the Eq. 3, in order to calculate the diesel mass flow rate.

$$\lambda = \frac{\dot{m}_{Air}}{(\dot{m}_{Diesel} * AF_{Diesel}) + (\dot{m}_{Ethanol} * AF_{Ethanol})} \tag{3}$$

where λ, $\dot{m}$ and AF are lambda, mass flow rate and air fuel ratio, respectively.

For combustion analysis, the heat release rate (HRR) was calculated using Eq. 4 (Heywood, 2018), where "γ" is the ratio of specific heats (c_p/c_v), "p_i" is the in-cylinder pressure measured (200 cycles averaged) at each crankshaft angle position (θ), "dt" is the 0.1° resolution of the optical encoder and "V_i" is the instantaneous in-cylinder volume. A constant value of γ was assumed for the compression (γ = 1.35) and expansion (γ = 1.30) stroke, respectively (Merket *et al., 2012)*. Integrating the HRR from the start of combustion (SOC) to the end of combustion, the total apparent net heat released and the mass burnt fraction (MBF) were assessed (MBF 10%, MBF 50% and MBF 90%). The SOC was defined by the moment when the HRR, after the start of injection, assumed positive value before the curve peak, representing effective heat release. Moreover, ignition delay (ID) was defined as the crank angle interval between the start of injection and SOC timings. The combustion duration (CD) was determined as the crank angle interval between SOC and MBF 90%.

$$HRR = \frac{\gamma}{\gamma - 1} p_i \frac{dV}{dt} + \frac{1}{\gamma - 1} V_i \frac{dp}{dt} \tag{4}$$

Measurement uncertainties were calculated and presented in the results section following the methodology presented in Da Costa *et al.* (2021), by dividing the analysis into uncertainty due to the records repeatability (type A) and the uncertainty due to the instrument accuracy (type B). Type A uncertainty was determined by dividing the acquisition sample standard deviation by the square root of the number of records. For directly measured dimensions, type B uncertainty was obtained by multiplying the relative accuracy reported in Table 2 by the average of the test reading sample.

3. RESULTS AND DISCUSSIONS

3.1 Combustion analysis

Tests were carried out at an engine speed of 1800 rpm and in 50% of the engine maximum load (about 10 Nm torque). They were analyzed for pure diesel and for 27%, 31%, 40% and 44% of ethanol energy fraction in diesel-ethanol dual-fuel mode, which was controlled by injection timing.

Figure 3 presents the in-cylinder pressure measured against crank angle for all test conditions, considering an average value over 200 cycles and a resolution of 0.1°. No significant differences were observed in the maximum in-cylinder pressure value, being 72.4 bar, 72.5 bar, 73.5 bar, 73.4 bar and 72.9 bar for pure diesel, EF=27%, EF=31%, EF=40% and EF=44%, respectively. Considering the uncertainties analysis performed, the maximum peak cylinder pressure uncertainty deviation achieved for all data collected was 0.05 bar. The addition of ethanol to the cylinder reduces the ignition capacity of the mixture due to cylinder cooling, with its high latent heat of vaporization, and due to the lower cetane number compared to diesel oil. It can be observed in Figure 3 that the addition of ethanol in the mixture moved the shape and the peak of the in-cylinder curve towards a delayed combustion in comparison to the pure diesel condition.

Figure 3 also presents the heat release rate curve (*HRR*) calculated against crank angle and Figure 4 presents the combustion characteristics (ignition delay (ID), MBF 10%, MBF 50%, MBF 90% and combustion duration for each test condition. As can be seen, with ethanol addition the ID increased and the combustion duration decreased, both progressively. Also, comparing the MBF10-50% and MBF50-90%, even with a later SOC in dual-fuel mode, ethanol accelerated the first and second half of combustion.

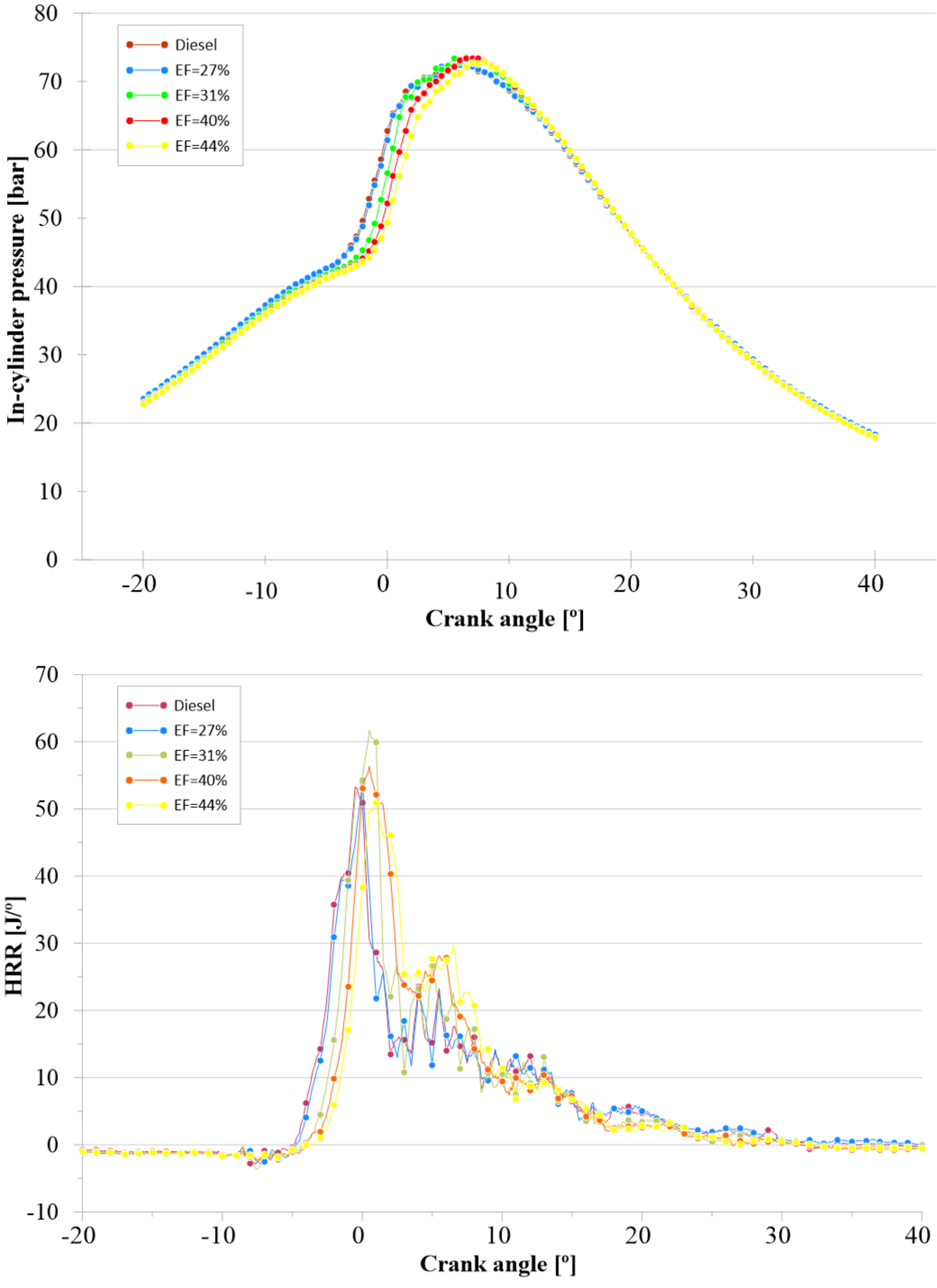

Figure 3. In-cylinder pressure and *HRR* behavior for each energy fraction.

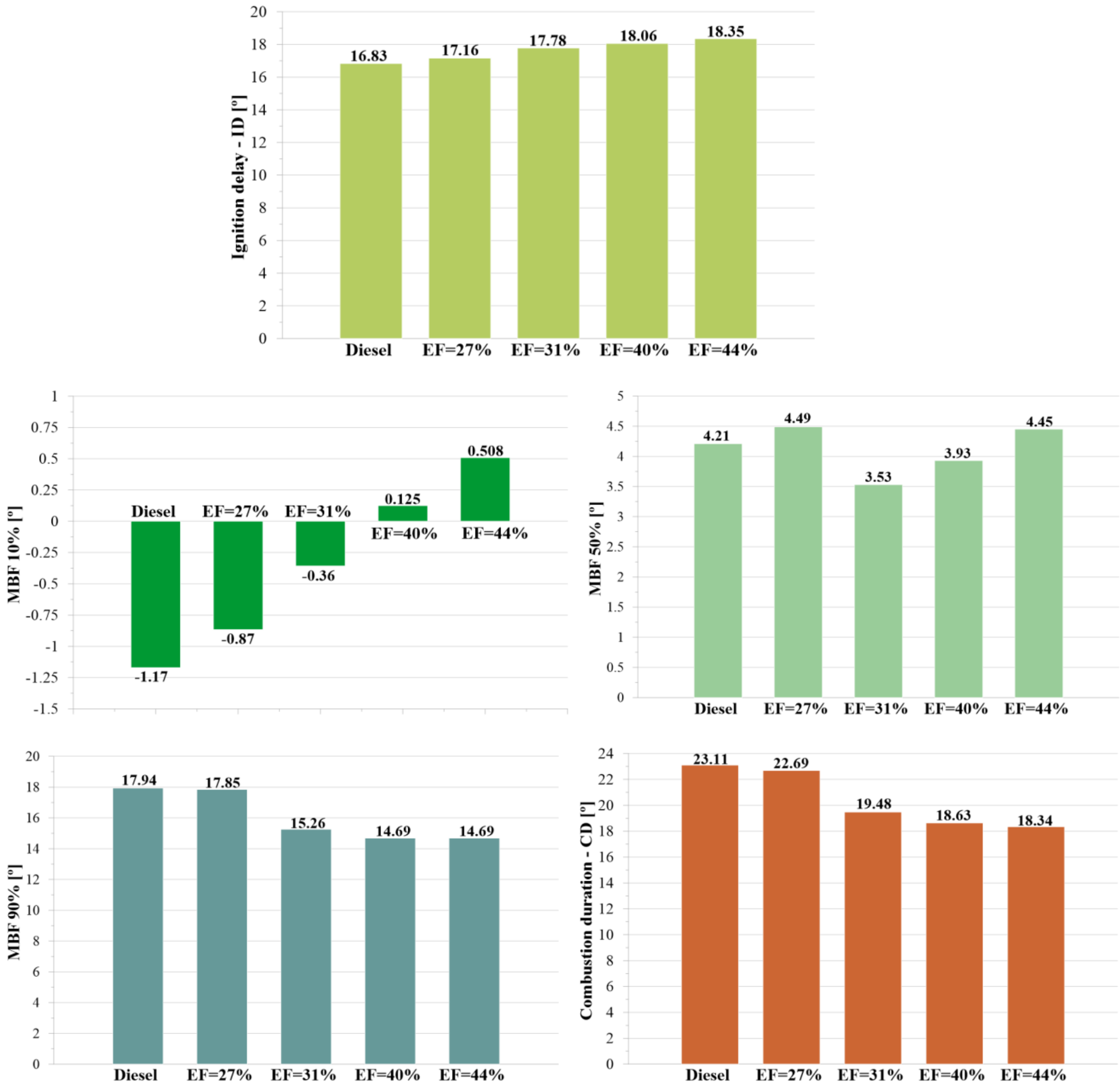

Figure 4. Combustion characteristics for each energy fraction.

3.2 Pollutant emission analysis

Figure 5 shows the NO_x behavior that decreases with higher ethanol energy fractions. Comparing pure diesel and an energy fraction of 44%, a NO_x reduction of 20% was observed. In a first moment, from pure diesel to an EF of 27%, its concentration reduced about 24%. But the main decrease was from an EF of 27% to 31% when NO_x concentration reduced 37%. From the pure diesel to an EF of 44%, there was a reduction of 71%. NO_x reduction is related to ethanol fuel psychochemical properties, the lower diesel concentration (so with an increase in the fuel's H/C ratio, NO_x emissions decrease), and the lower in-cylinder temperatures at a given crank angle, as the ethanol addition (with high latent heat of vaporization) delayed initial combustion development. Additionally, smoke levels presented a considerable decrease with energy fractions enhancement, as can be seen in Figure 5. Soot emissions decreased due to longer ignition delay with better mixing of fuels, and also due to more complete combustion (in consequence of ethanol´s hydroxyl).

However, CO and HC levels increased when ethanol energy fraction also increased. Figure 5 presents the CO behavior and it is possible to conclude that there was a 168% increase from pure diesel to EF of 44%. But a constant behavior was observed from pure diesel to EF of 27%, showing that the more significant increase was in the next point when the EF was in 31%. Regarding HC, Figure 5 shows a 259% increase in HC levels from pure diesel to EF of 44% (higher than CO increase). HC levels start to increase significantly after EF of 31%. The increase of HC and CO levels with dual-fuel technology in this case, can be attributed to premixed fuel trapped in the crown volumes of the stock diesel piston.

Beside that, CO_2 emissions presented a reduction of 7% with higher ethanol energy fractions as shown. It was the lower percentage of reduction that was observed.

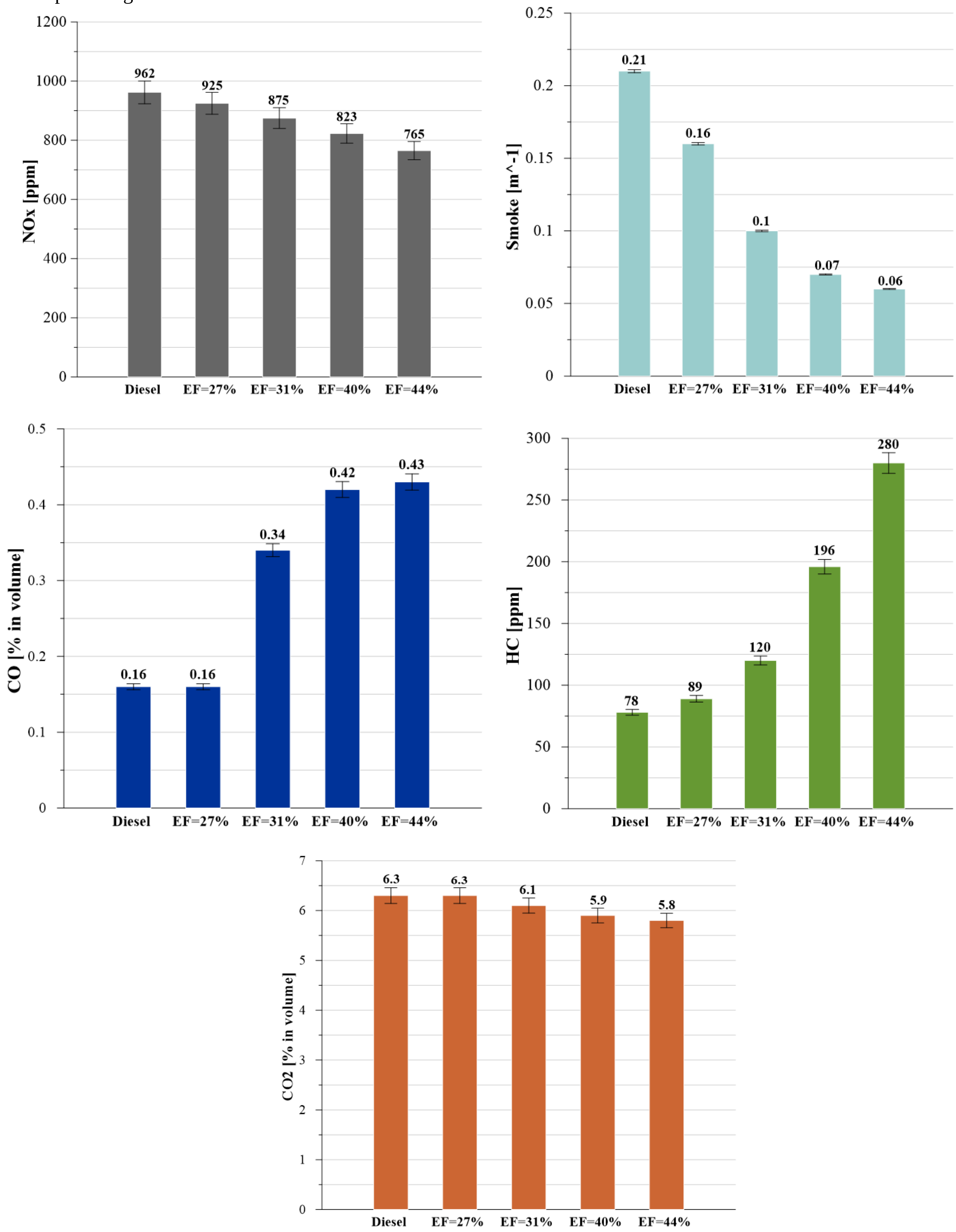

Figure 5. NO_x, smoke, CO, HC and CO_2 emissions for each energy fraction.

3.3 Efficiency analysis

Figure 6 shows the fuel conversion efficiency for all test conditions. A reduction in the fuel conversion efficiency due to the lower heating value of ethanol compared to Diesel when Dual-Fuel is used was observed. Also, the reason for this lower efficiency is the reduction of the in-cylinder temperature in early stages of the cycle (Liu *et al.*, 2018).

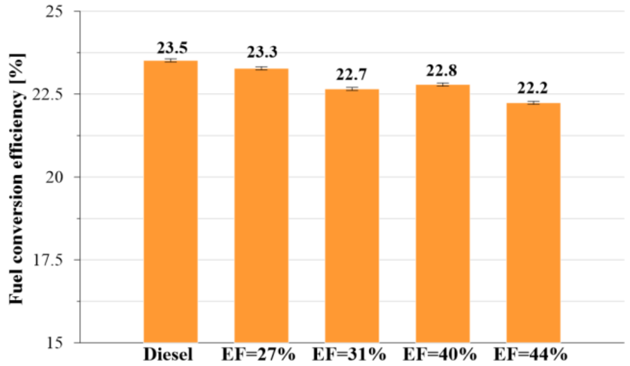

Figure 6. Fuel conversion efficiency for each energy fraction.

4. CONCLUSIONS

The sugarcane derived fuel (ethanol) lower heating value and the lower in-cylinder pressures at the end of the compression stroke obtained in the diesel-ethanol dual-fuel mode (caused by the ethanol higher latent heat of vaporization) in comparison with the pure diesel operation, promoted a longer ignition delay and reduced slightly engine efficiency. Furthermore, higher HC and CO emission levels were assessed.

However, a reduction of total combustion duration and a significant reduction in NO_x emissions and opacity of the exhaust gases were observed with dual-fuel technology, improving the NO_x-smoke trade-off, the main concern of emission homologation nowadays. Increased HC and CO emissions are believed to be manageable by a diesel oxidation catalyst.

In this sense, ethanol, a well-established and environmentally friendly fuel (specially for Brazil), presented interesting behavior for dual-fuel technology use with diesel engines, as it can reduce NO_x and smoke levels, and collaborate to reduction of CO_2. The latter can also be intensified if a life cycle analysis is considered.

5. ACKNOWLEDGMENTS

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