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A comparative study of performance and regulated emissions in a medium-duty diesel engine fueled with sugarcane diesel-farnesane and sugarcane biodiesel-LS9

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Keywords
Sugarcane diesel-farnesane; Sugarcane biodiesel-LS9; Diesel engine; Regulated emissions, European Transient Cycle.

Abstract
Two sugarcane biofuels and mineral diesel fuel are tested under full load conditions**,** under the same values of performance and under the European Transient Cycle on an engine test bench, without any modifications to the ECU. The target is to compare engine performance and emissions. At full load, engine performance varies due to the variation in LHV. Under the same values of performance, the sugarcane biodiesel-LS9 provides the lowest THC emissions. The higher CN and exhaust gas recirculation of the sugarcane biodiesel-LS9 and the higher H/C ratio of the sugarcane diesel-farnesane compared to the diesel S50 provide a NO_x reduction. Neither the increment in *bsfc* nor the increment of %EGR for the sugarcane biodiesel-LS9 deteriorate the combustion, so its CO emissions are lower. **T**he sugarcane biodiesel-LS9 leads to the lowest NO_x and PM specific emissions under transient operation, followed by the sugarcane diesel-farnesane. The THC and CO specific emissions are higher for the biofuels in comparison to the diesel S50. The main reason for these results is the impact of the properties of the biofuels on the ECU response However, both biofuels produce less harmful emissions at idle conditions, which supports their usage to reduce exhaust emissions in urban areas.

1. Introduction

Air pollution is a serious problem in the world, largely caused by the use of mineral fuels for transportation. This is why the governments of most countries are planning the reduction of the environmental contamination and of the greenhouse gas emissions. An option to help reduce this problem is to design and build engines that operate efficiently using renewable fuels, i.e., an integrated approach between engine technologies and fuel composition, which allows the definition of the best combination to address the environmental problem.

Ethanol and biodiesel biofuels are biomass-derived. Bioethanol can be obtained by the fermentation of sugars and starches (first-generation fuel). The gasoline marketed in Brazil is mixed with

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anhydrous ethanol. The Brazilian Executive Branch can increase the percentage of ethanol in gasoline up to the limit of 27.5% or reduce it to 18% [1], thus developing a wide sugarcane production. Another first-generation biofuel is biodiesel produced from vegetable oils or animal fat.

The second-generation ethanol biofuel is produced from non-food biomass, such as cellulosic matter, including sugarcane bagasse, which is widely used in Brazil [2]. However, the conversion of biomass to ethanol using fast, inexpensive and efficient methodologies to disintegrate and hydrolyze cellulosic biomass is the major challenge in the production of second-generation ethanol [3]. These authors also assure that several procedures have been established for the conversion of cellulose into ethanol, but none of them is completely satisfactory when considering processing time, costs and efficiency.

Third-generation biofuels produced by the company Amyris, with advanced chemical processes, are fuels besides ethanol (such as butanol, cellulosic diesel, renewable diesel, etc.) from sources such as sugarcane, wastes, algaes and cellulosic feedstocks. Other companies in this sector are LS9 and Gevo. LS9 is the most similar to Amyris because of its reliance on synthetic biology to produce renewable fuels and chemicals [2].

From sugarcane, the Brazilian companies Amyris and LS9 produce two types of biofuels for diesel engines: the sugarcane diesel-farnesane and the sugarcane biodiesel-LS9, respectively, both using synthetic biology involving fermentation.

The fermentation process of Amyris is through genetically engineered yeast cells that convert glucose or sucrose into a compound called farnesene, an unsaturated oil with the molecular structure $C_{15}H_{24}$. This oil is subsequently saturated through the addition of hydrogen molecules in a process known as hydrogenation, resulting in farnesane, an isoparaffin $C_{15}H_{32}$ (2,6,10-trimethyldodecane) [4]. In Rio de Janeiro, a study was carried out with the introduction of sugarcane diesel-farnesane in urban freight transport, showing its environmental benefits regarding pollutant emissions with reduction values up to 27% carbon monoxide (CO), 22% nitrogen oxides (NO_x), 16% total hydrocarbons (THC) e 10% particulate matter (PM) relative to mineral diesel fuel. However, there was an increase in mean operating costs of 23.1 % [5]. This financial impact is mainly caused by the price of sugarcane diesel-farnesane, which exceeds the cost of mineral diesel fuel by 70%. This cost difference may decrease with the increase in production scale, government incentives or new technologies, thus reaching values closer to the price of petroleum diesel at the fuel stations.

The fermentation process LS9 yields a mixture of various fatty acids of different structures as intermediate products. The sugarcane biodiesel-LS9 products include methyl and ethyl esters (FAME and FAEE) [6]. These esters obtained by transesterification can be considered 16-carbon structures [7], unsaturated with the molecular structure $C_{16}H_{30}O_2$ [6].

These two sugarcane biofuels could be very important in replacing mineral diesel fuel used for transportation. Therefore, the present paper analyses their effects on the performance and emissions of a diesel engine, which emits high levels of particulates and NOx due to high injection pressures and air-fuel ratios [8]. This is a critical context for diesel engines because these emissions must be reduced without penalizing its performance. Biofuels can contribute positively to this balance, in which the transient operation behavior of the engine is increasingly relevant to both questions, demonstrating their environmental sustainability. The use of alternative fuels is an area that progressively attracts scientific interest [9] and biodiesel is widely used in diesel engines [10] because of its advantages, especially the ones related to environmental care and renewable nature [11].

Publications about sugarcane diesel-farnesane are scarce and even scarcer for sugarcane biodiesel-LS9. Studies of the thermal degradation of sugarcane diesel-farnesane in environments other than the combustion chamber of an engine have already been published, among which stands out the work from Conconi and Crnkovic [12]. In this study, it was verified from the oxidation activation energies that pure sugarcane diesel-farnesane and mineral diesel S50 marketed in Brazil have different combustion chemical kinetics.

The aforementioned study and the one performed by Crnkovic et al. [13] considered environments different from the combustion chamber of an engine. Nevertheless, Soto et al [14] studied the burning process through the application of the technique of thermal degradation assessed by thermogravimetry to determine the activation energy, as well as through the experimental determination

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of engine performance and soot emissions of a diesel engine. Both techniques showed that the more efficient combustion process is obtained for an 80% diesel-20% biodiesel blend.

Oßwald et al. [15] developed a study of the combustion chemical kinetics of farnesane from natural terpenoids in a high temperature laminar flow reactor at atmospheric pressure coupled to a mass spectrometry system, in order to evaluate the necessary properties of farnesane fuel blends intended for the aviation use. One of the favorable results for the farnesane was that it showed little or no tendency to produce soot precursors.

Shock-tube measurements for the gas phase ignition of hydroprocessed biofuels showed a minor reduction in ignition delay at low to moderate temperatures (650-900 K) with the increase of the cetane number (CN) [16, 17]. Based on and continuing these studies, Gowdagiri et al. [18] studied the influence of the CN on the ignition delay of the same fuels in a reciprocating engine, confirming that the moderate reduction in the ignition delay for considerable changes in CN indicates that most of the ignition delay is composed of a physical delay. These authors studied the relationship between NO emissions and H/C ratio and density. Fuels with higher fractions of unsaturated species have higher density and lower H/C ratios and therefore lead to higher NO_x emissions. This correlation with the H/C ratios of fuels observed by Gowdagiri et al. [18] is consistent with another study performed by Cecrle et al. [19] on the same engine using biodiesel and measuring NO_x emissions.

Other works studied the performance and emissions of the reciprocating engine with the use of sugarcane diesel-farnesane, as in Millo et al. [20]. These authors showed a comparison of full load engine performance for mineral diesel fuel and its 30% blend with farnesane, without any modifications of the Engine Control Unit (ECU) calibration. A low decrease in torque and thermal efficiency compared to mineral diesel (about 2% on average) was observed over the entire engine speed range when the farnesane-F30 blend is used. This behavior is justified by the higher mass calorific value of farnesane, which attenuates the negative effect of its lower density. In addition, the authors confirm that the ratio between the lower heating value and the stoichiometric air-fuel ratio, $LHV/(A/F)_{stoichiometric}$, have identical values for both fuels. This suggests that there is a potential to restore the engine performance at full load with the farnesane-F30 blend provided that the ECU is recalibrated properly. The emissions benefits obtained from the use of the farnesane-F30 blend, observed by these authors, support the viability of this renewable fuel.

On the other hand, unlike the present paper in which a medium-duty engine was tested under ETC test, Soriano et al. [21] evaluated the impact of several fuels (diesel fuel, soybean-palm biodiesel, GTL and sugarcane diesel-farnesane) on performance and emissions of a light-duty engine using the default ECU configuration. This engine was tested under five steady state operating modes which cover most of the zone characterized by NEDC (New European Driving Cycle) and part of the engine operation area established in the WLTC (Worldwide harmonized Light duty vehicles Test Cycle), trying to follow the recommendations for an appropriate design of experiments [22]. Taking into account that for the operating modes belonging to the WLTC the ECU is not optimized, the authors found similar engine performance with all fuels. Reductions of THC and particulate emissions were observed downstream of the DOC (Diesel Oxidation Catalyst) with all alternative fuels and for all operating modes which characterize the NEDC. These results highlight the benefits in terms of pollutant emission obtained with farnesane.

As regards to the use of sugarcane biodiesel-LS9, Marques et al. [23] investigated the performance and emissions of the same engine studied in the present research. This work proved that the power of the engine fueled with 100% sugarcane biodiesel-LS9 drops considerably. However, a 20% vol. blend of this biodiesel in mineral diesel fuel delivers a very similar performance to that offered by pure mineral diesel fuel and substantially reduces pollutant emissions (NO_x, particulate material and smoke opacity). According to the authors, the decrease in power observed for the sugarcane biodiesel-LS9 was caused mainly by a decrease in the ratio between effective efficiency and excess air coefficient (η_e/λ), although both fuels have the same values of $LHV/(A/F)_{stoichiometric}$. The work of Marques, Izquierdo and Coutinho [23] also presents an analysis of combustion chemical kinetics aided by thermogravimetry. The authors demonstrated that diesel fuel is composed of hydrocarbons that are more reactive than sugarcane biodiesel-LS9, which favors the pre-ignition of this mineral fuel. This pre-ignition is also helped by the existence of more volatile hydrocarbons (64% v/v) in mineral diesel fuel. However, during the ignition

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phase, the activation energy values of the sugarcane biodiesel-LS9 were lower than those of the diesel fuel because the mineral fuel presented a significant quantity of compounds that are less volatile than the sugarcane biodiesel-LS9, favoring the combustion of the biofuel.

As a precedent for this work, a numerical analysis of the combustion of sugarcane biodiesel-LS9 and sugarcane diesel-farnesane was carried out on the same engine tested in this work, using Computational Fluid Dynamics (CFD). A simplified computational geometric domain was built to represent the real combustion chamber [24]. Two injection strategies were simulated: simple and triple. The modeling made good predictions for average ignition delay, average pressure and temperature and maximum combustion temperature, all as a function of the engine speed. Specifically, it was found that the ignition delay has a clear dependence on the physical properties of the fuel and not only on its cetane number. The simulation made good qualitative predictions for the sugarcane biodiesel-LS9 and the sugarcane diesel-farnesane. From this study, the papers [25, 26] were presented using simple injection strategies, i.e., a single injection for both sugarcane biofuels. Due to the properties of each of them, it was observed that the sugarcane diesel-farnesane presented better performance compared to the sugarcane biodiesel-LS9.

In the work presented here, three fuels are studied through experimental tests in a medium-duty diesel engine. The reference fuel is diesel fuel available at the Brazilian fuel stations, known as S50, which contains 5% biodiesel. The other two fuels are biofuels of the same origin: sugarcane. According to the previous review, these neat biofuels have not been previously tested under the European Transient Cycle (ETC). For this reason, the objective of this study is to determine and to compare the performance and emissions of the engine fueled with mineral diesel S50 as a reference, sugarcane biodiesel-LS9 and sugarcane diesel-farnesane, under full load conditions and under the European Transient Cycle (ETC). The engine has no modifications to the management system of the Engine Control Unit (ECU). The comparison between the engine performance provided by biofuels and that of the reference fuel allows stablishing their possible usability without ECU modifications, while favorable emissions tests would support the potential of the tested biofuels to replace mineral diesel fuel. In order to determine engine performance at full load, the following responses are experimentally determined: brake specific power, torque, fuel consumption, brake specific fuel consumption, brake thermal efficiency, air mass flow rate and intake air pressure. In the case of the transient test, the brake effective power, torque, the cumulative consumption, fuel consumption and air flow rate are obtained, as well as the instantaneous regulated emissions CO, THC, NOx and the specific emissions of CO, THC, NOx and PM.

2. Materials and methods

2.1. Experimental test bench

The research is executed on a medium-sized diesel engine under the Proconve P7 standard. This Brazilian standard follows the Euro V standard and is part of a Brazilian air pollution control program for automobiles. The engine has two-stage forced induction by means of turbochargers, with an intercooler for each stage and exhaust gas recirculation system (EGR). The specifications of the engine are shown in Table 1.

Table 1. Engine specifications.

Engine Model	MAN D0834 LF05
Operation	Diesel, 4 stroke, 2 turbochargers
Cylinders/Displacement	4 in line/4.58 liters
Bore	108 mm
Stroke	125 mm
Valves	4 per cylinder
Compression ratio	16.5:1
Max power/speed	166 kW/2400 min^{-1}
Max Torque/speed	850 Nm/1100 - 1600 min^{-1}
Injection System	Electronic Common Rail
Emissions Standards	Proconve P7 (Euro V)

The engine experimental tests were funded by MAN-Latim and executed at MAHLE Metal Leve S.A., Brazil, Technical Center, Experimental Engineering – Engine Laboratory, on an experimental test bench equipped with a Motoring AVL APA 220 Dynamometer that can measure up to 220 kW power,

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930 Nm torque and 10000 min^{-1} engine speed. The engine emissions were measured through the analysis of the exhaust gas collected before the Diesel Oxidation Catalyst (DOC). The gaseous emissions were measured with an AVL AMA i60 gas analyzer. The particulate emissions were measured with an AVL Smart Sampler. This device is a dilution system of a partial flow of the engine exhaust with filtered air, designed for the gravimetric sampling of exhaust particulates by collecting them in a filter. It allows measurements under steady state or transient conditions. For transient simulation, it performs a fast control of the flow rates and of the dilution ratio of the dilution tunnel. Fig. 1 shows the experimental test bench used in this study and Table 2 shows its main the specifications.

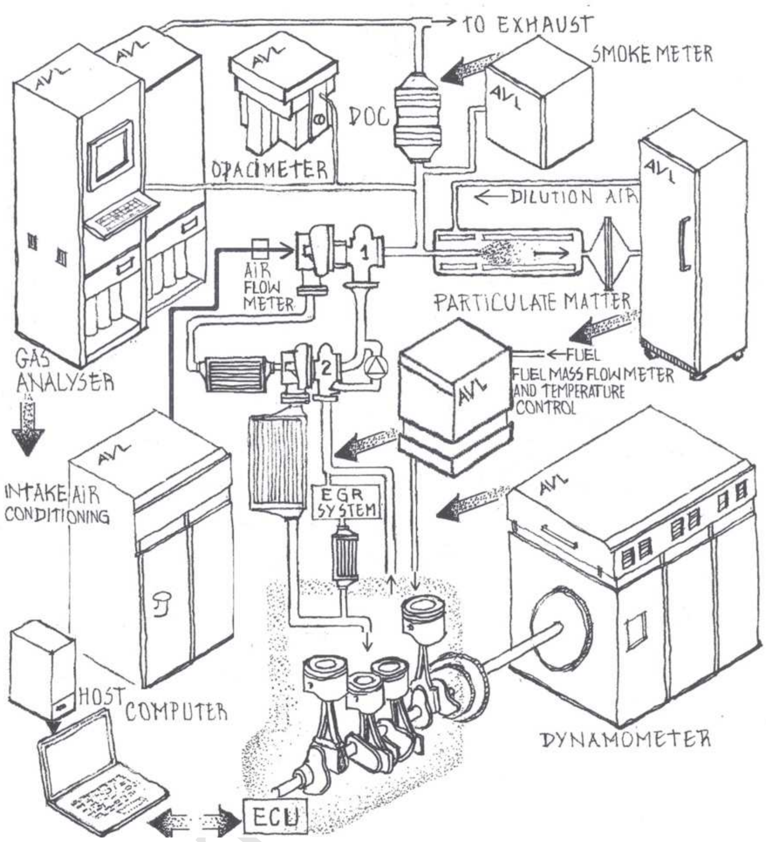

Fig. 1. The engine test bench.

Table 2. Specifications of the experimental devices.

Equipment	Parameter	Range [units]	Accuracy
AVL Smart Sample SPC 472	-	-	-
AVL 735S / 753S	Fuel mass flow Fuel supply temperature	0 ÷ 125 [Kg/h] -10 ÷ 40 [^oC]	±0.12%
ABB Sensy flow FMT700-P	Air flow ratio	1:40	±2 % air mass
AVL Intake air conditioning CONSYSAIR 2400	Temperature Pressure Moisture	-40 ÷ +60 [^oC] -400 ÷ 200 [mbar] 4 ÷ 34 [g H_20/kg dry air]	± 0.5 [^oC] ± 1 [mbar]
Vaisala HMT333	Intake air temperature Intake air humidity	-40 ÷ +60 [^oC] 0 ÷ 90% /90% ÷ 100%	± 0.2 [^oC] ± 2%/± 3%
AVL DynoExact 220 US (Active dynamometer)	Torque Speed Power	0 ÷ 934 [Nm] 0 ÷ 10000 [rpm] 0 ÷ 220 [kW]	± 0.1%
AVL Emissions test bench — FID: Flame Ionization Detector	Emissions measurem — Total hydrocarbon (THC) CH_4	THC: 10 ÷ 20000 [ppm] CH_4: 30 ÷ 20000 “	± 0.5%
AVL Emissions test bench — CLD: Chemilumines-cence Detector	Emissions measurem — NO, NO_X	3 ÷ 1000 [ppm]	± 0.5%

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IRD: Infrared Detector	CO, CO_2	CO: 50 ÷ 5000 [ppm] CO_2: 0.1 ÷ 6 Vol%	± 0.5%
PMD: Paramagnetic Detector	O_2	0 ÷ 25%	± 0.5%
LDD: Laser Diode Detector	NH_3	5 ÷ 1000 [ppm]	± 0.5%
QCL: Quantum Cascade Laser	N_2O	5 ÷ 1000 [ppm]	± 0.5%
AVL 415S (filter paper blackening)	Soot concentration	1 ÷ 10 FSN [Filter Smoke Number]	±0.02 [mg/m³]
Micro Balance XP2U	Particulate matter	0.00003 ÷ 2.1 [g]	±0.0001 [mg]
AVL Opacimeter	Opacity N	0 ÷ 100%	±0.01%

2.2. Properties of the fuels

The properties of the tested fuels are listed in Table 3.

Table 3. Properties of the fuels.

Property	Unit	Diesel S50	Sugarcane diesel-farnesane	Sugarcane biodiesel-LS9
Mass fractions of carbon	[-]	0.870	0.848	0.755
Mass fractions of hydrogen	[-]	0.125	0.152	0.119
Mass fractions of oxygen	[-]	0.001	0.0	0.126
Mass fractions of sulfur	[-]	0.004	0.0	0.0
Approximate summarized formula	[-]	$C_{15.2}H_{27.3}$*	$C_{15}H_{32}$	$C_{16}H_{30}O_2$
H/C		1.796	2.133	1.875
Cetane number	[-]	49	58.7	69.9
Density at 20 °C	kg L^{-1}	0.843	0.770	0.875
Viscosity at 40 °C	mm^2 s^{-1}	3.11	2.71	3.326
Lower heating value (LHV)	MJ kg^{-1}	42.03	43.43	35.80
Lower heating value	MJ L^{-1}	35.43	33.44	31.32
Stoichiometric air-fuel ratio $(A/F)_{st}$	[-]	14.50	14.98	12.23
$LHV/(A/F)_{st}$	MJ kg_{air}^{-1}	2.899	2.899	2.927
Distillation 10% vol	°C	196.0	250.5	280.0
Distillation 50% vol	°C	260.0	252.0	294.0
Distillation 90% vol	°C	340.0	254.0	306.0

*the oxygen was not considered due to its low mass fraction.

2.3. Experimental tests and engine performance analysis

In the present Section, engine performance responses are obtained under full load conditions and under the ETC cycle. At full load conditions, each response is calculated as the mean value of three experimental tests, the uncertainty of which, with a confidence level of 70%, is determined according to the “guide to the expression of uncertainty in measurement” [27]. Spurious data are removed taking into account the Chauvenet’s criterion.

2.3.1. Full load experimental tests

One of the issues impeding the acceptance of biofuels in engines used for transports is their low heating values by the volume unit in comparison to petrol-derived fuels. This is true both for the different kinds of biodiesel fuels that substitute mineral diesel fuel and for the bioethanol that substitutes gasoline. This matter includes the two kinds of biofuels tested in this work, especially in relation to the difference in the properties of the sugarcane biodiesel-LS9. Fuel distribution stations sell fuels by the volume unit, not by the energy unit. However, vehicles do not run on fuel volume but on fuel energy [28]. This is why the vehicle driver, trying to get from the biofuel the same performance achieved by the mineral fuel, unknowingly increases the pressure on the accelerator pedal leading to a higher fuel consumption.

When dynamometer experimental tests are run on engines, the situation related above develops differently based on the nature of the test, i.e., tests at full load at maximum accelerator pedal position

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(fixed position) to measure maximum values of power and torque at different engine speeds. This is the first set of experimental tests presented in this work. Fig. 2 shows the behavior of the power and the torque of the engine operating at full load fueled with sugarcane diesel-farnesane and with sugarcane biodiesel-LS9, both pure. The brake effective power and torque values achieved with mineral diesel S50 are also presented on the figures for reference.

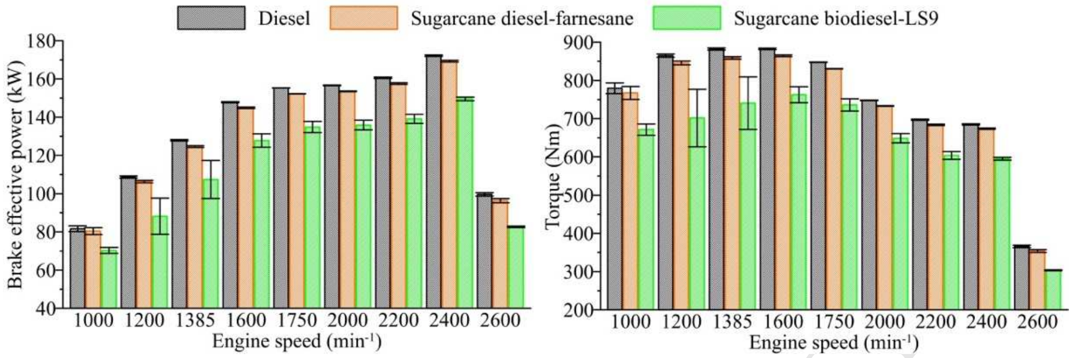

Fig. 2. Engine brake effective power and torque at full load.

These results correlate to the lower heating values (LHV) of the fuels shown in Table 2, as the remarkably lower energy value of the sugarcane biodiesel-LS9 leads to an expressive decrease in power and torque. As for the sugarcane diesel-farnesane, its LHV in kJ by kg of fuel is approximately equivalent to that of mineral diesel S50, such that both fuels show similar values of power and torque. This attenuates the negative effect caused by the lower fuel density. This analysis is not complete if the differences in fuel consumption are not considered, despite the fixed position of the accelerator pedal. Fig. 3 shows these results along with the brake specific fuel consumption (*bsfc*).

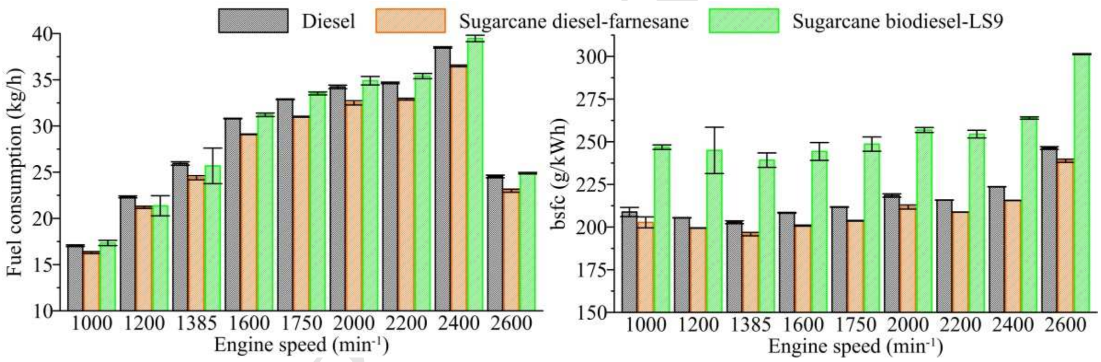

Fig. 3. Engine fuel consumption and brake specific fuel consumption at full load.

The differences in fuel consumption values among the fuels, presented in Fig. 3, are due to the ECU management system. To meet the power demand, it acts on different points of its map to increase the quantity of sugarcane biodiesel-LS9 delivered and decrease the quantity of sugarcane diesel-farnesane delivered, due to the differences in their LHVs in comparison to the reference diesel fuel as seen in Table 2.

In the range of engine speeds from 2000 to 2400 min^{-1}, which corresponds to the maximum power, the consumption of sugarcane biodiesel-LS9 increased 1.9 kg h^{-1} on average in relation to the mineral diesel S50. In the same range, the consumption of sugarcane diesel-farnesane decreased 1.2 kg h^{-1} on average. These values represent a 5.33% increase and a -5.20% drop in fuel consumption respectively. At 1600 min^{-1}, which corresponds to the maximum torque, the consumptions of sugarcane biodiesel-LS9 and sugarcane diesel-farnesane increased 4.74% and decreased 5.49% respectively, compared to the reference diesel fuel.

Also, from Fig. 3, at 2400 min^{-1} (maximum power), it can be inferred that the *bsfc* of the engine fueled with sugarcane biodiesel-LS9 increased 18.3% in relation to the reference diesel fuel. This value corresponds to a -14.8% decrease in the LHV in MJ kg^{-1} of the sugarcane biodiesel-LS9 in relation to the reference diesel fuel. These results do not correspond with most studies published about light and heavy-duty diesel engines, which claim that the increase in *bsfc* of the engine fueled with biodiesel is, on

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average, similar to the difference in the LHV of the fuel [28]. Additionally, the 18.3% increase in *bsfc* corresponds to a -11.6% decrease in the LHV in MJ L^{-1}, which is a larger difference. This result also does not correspond with the study that shows that the differences in viscosity, density and cetane number of the biodiesel are significant because they can influence the combustion duration, leading to values of energy by liter of fuel of the same order of the *bsfc* [29].

For sugarcane biodiesel-LS9 to be positioned within these statistics, the increase in the fuel consumption through the redistribution on the different points of the ECU map to increase the fuel delivery was not enough, nor was its higher viscosity and density values. A greater increase in hourly fuel consumption by the recalibration of the ECU would be necessary and this increase in fuel delivery should cause a greater increment in power, i.e., whenever the brake thermal efficiency does not drop.

Fig. 4 shows the brake thermal efficiency (η_b) of the engine at full load, determined by $\eta_b = 3600/(bsfc\ \mathrm{LHV})$. Through the analysis of the results at 2400 min^{-1}, brake thermal efficiencies of 37% and 38.3% are observed for the engine fueled with sugarcane biodiesel-LS9 and mineral diesel S50 respectively. This represents a 1.3% loss in brake thermal efficiency in the use of this biofuel. This value is much lower than the difference in the LHV of the sugarcane biodiesel-LS9 in relation to the reference mineral diesel S50, which corresponds to 14.8% as seen before. Therefore, the increase in bsfc is not caused by the loss in brake thermal efficiency, but by the lower LHV of the sugarcane biodiesel-LS9.

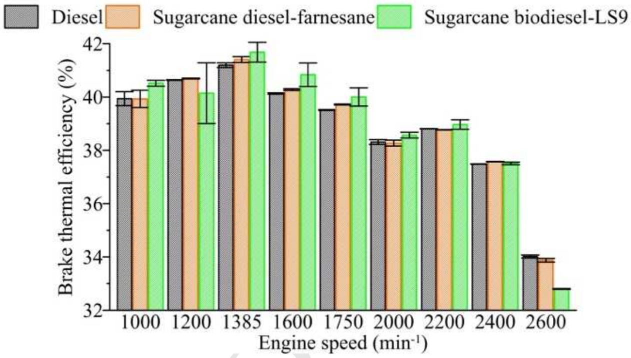

Fig. 4. Engine brake thermal efficiency (η_b) at full load.

Comparing the brake thermal efficiency values of the engine running on sugarcane diesel-farnesane and reference mineral diesel S50, as seen in Fig. 4, there are no significant differences.

Taking into account that η_b and engine mechanical efficiency η_m were obtained at the same engine speed, we will show that the behavior of η_b is conforming to η_m behavior for each of the tested fuels. This fact can be explained by the variation of turbocharger conditions showed in Fig. 5. This Figure illustrates the exhaust gas recirculation mass flow (m_{EGR}), and the percentage of m_{EGR} respect to the total inlet charge (%EGR). A simplified enthalpy balance presented by [30] was used to calculate m_{EGR}.

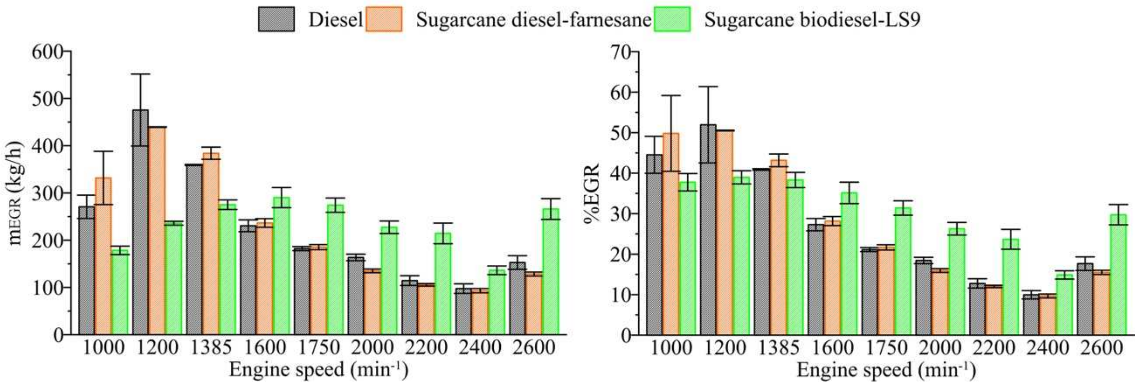

Fig. 5. Exhaust gas recirculation mass flow (m_{EGR}) and %EGR.

If the m_{EGR} increases, less amount of exhaust gases is delivered to the turbines of the turbochargers and, consequently, turbines work decreases. This fact causes an increase in (η_m) if the ratio

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of power needed to drive the turbochargers (N_m) to indicated power (N_i) decreases (see Eq. 1):

$$\eta_m = 1 - N_m/N_i, \qquad \text{(Eq. 1)}$$

and η_m can be used to determine η_b by applying Eq. 2:

$$\eta_b = \eta_i\, \eta_m, \qquad \text{(Eq. 2)}$$

where the indicated efficiency (η_i) can be determined from the LHV and the indicated specific fuel consumption (ISFC) as Eq. 3 shows:

$$\eta_i = 1 / (\text{LHV} \cdot \text{ISFC}). \qquad \text{(Eq. 3)}$$

According to the previous equations, the increase in η_b observed in Fig. 4 for the sugarcane biodiesel LS-9 is caused by the greater increase in η_m compared to the decrease in η_i. ECU configuration provides a similar engine behavior for this biofuel, which is oxygenated and has less energy content, to diesel fuel by modifying the m_{EGR}, as well as by increasing the fuel supplied to the cylinders, as seen earlier. This analysis which relates η_b to η_m behavior linked to m_{EGR} regulation can be applied from 1600 to 2400 min^{-1}, since at 2600 min^{-1} the fuel supply is cut off (see Fig. 3). This result supports the reliability of the simplified way for determining m_{EGR} proposed by [30], as well as the decrease in η_b observed at 1200 min^{-1} for biodiesel-LS9, since the calculation of m_{EGR} shows the lowest value. In general, from the lowest values of engine speed tested at full load (from 1000 to 1385 min^{-1}) it is reasonable to assume that, for the biodiesel-LS9, the amount of exhaust gas recirculation mass is less than for the other tested fuels in order to avoid an even greater decrease in the amount of intake air mass flow rate, as it will be shown below.

As for the air delivery to the engine, the ECU management and control, to meet the emissions regulations and the CO_2 standards, acts on the EGR system varying the opening of the valve (most of the ECU management and control maps are controlled by the engine speed and by the accelerator pedal position or by the fuel demand). Consequently, the intake air mass flow rate is varied. Fig. 6 shows the intake air mass flow rate for the fuels tested.

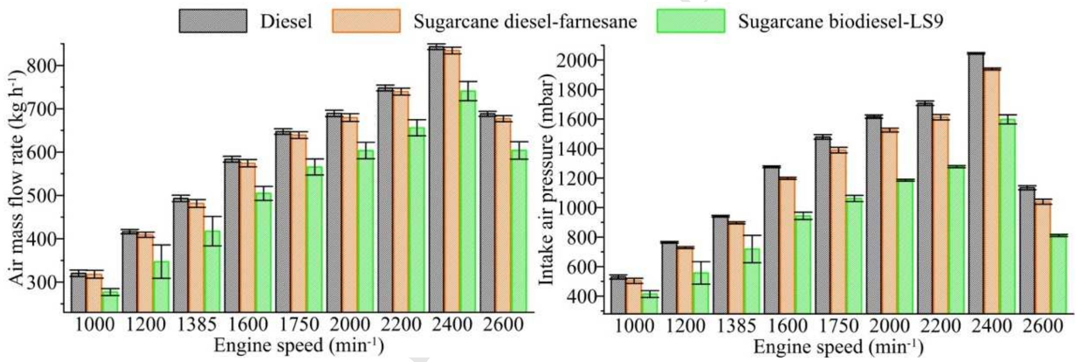

Fig. 6. Engine intake air mass flow rate and pressure.

The reduction in intake air mass flow rate observed for the sugarcane biodiesel-LS9 was due to a larger opening of the EGR valve, which caused a lower energy quantity (and less impulse) available at the inlet of each turbine. This caused a decrease in intake air pressures as seen in Fig. 6. These results indicate that the EGR valve opening varied little with the change from reference mineral diesel fuel to sugarcane diesel-farnesane.

Based on intake air mass flow rate and fuel consumption experimental data, it is appropriate to perform a comparative analysis of the excess air coefficient λ, which can be calculated as the ratio of the actual air-fuel ratio $(A/F)_{actual}$ to stoichiometry air-fuel ratio $(A/F)_{st}$ (see Fig. 7), where $(A/F)_{st}$ is provided in Table 2 for the tested fuels.

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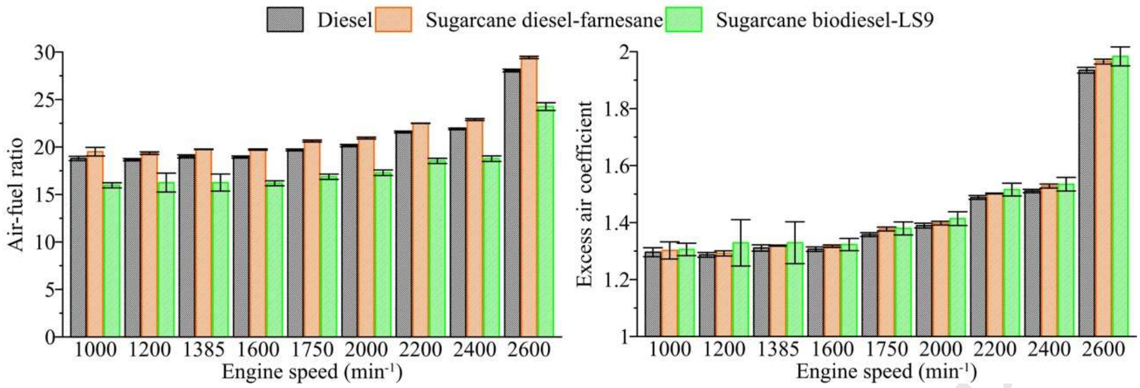

Fig. 7. Actual air-fuel ratio $(A/F)_{actual}$ at full load and excess air coefficient (λ) at full load.

Taking into account that the ratio $LHV/(A/F)_{st}$ is the same for all the tested fuels (approximately 2.9, see Table 2), LHV value of each fuel is the responsible for the *bsfc* variation, since the η_b shows a similar behavior for all fuels. According to Fig. 7, for all operating regimes tested, a slight increase in λ is observed for the biofuels respect to the reference fuel. This small variation is even lower for λ values than for η_b values showed in Fig. 4. Consequently, the ratio $\chi = \eta_b/\lambda$ remains in a small range of variation for the three tested fuels in each operating condition (see Fig. 8)

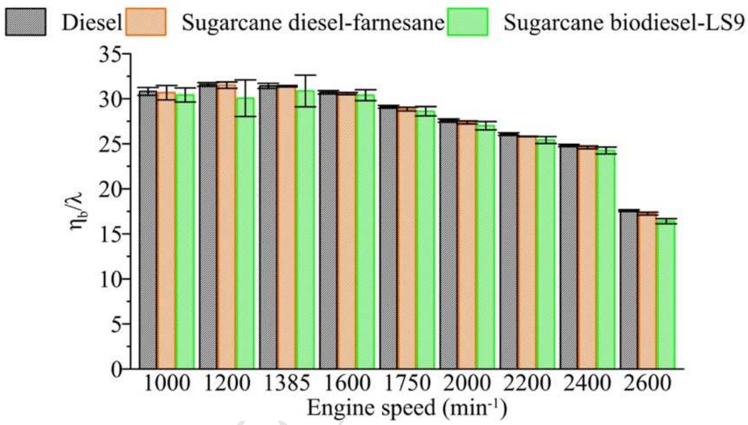

Fig. 8. Ratio of brake thermal efficiency to air-fuel equivalence ratio (χ) at full load.

When λ decreases below a specified limit, the combustion process drastically deteriorates, smoke emissions increase and the engine overheats above acceptable levels. The acceptable maximum value of λ is that corresponding to the maximum value of η_b/λ. The maximum amount of fuel injected per cycle (fueling) is established taking into account that maximum λ is slightly greater than the maximum value of η_b/λ. For the engine under study, λ values at nominal regime vary from 1.3 to 1.54 in the engine speed interval 1000-2400 rpm (see Fig. 7) for the three tested fuels. Those λ values provide an acceptable engine heating. For engines speeds between 1200 and 1750 rpm, the fueling increases (see Fig. 9) and the maximum torque values are recorded (see Fig. 2). The decrease in λ is limited by the smoke formation. This limit corresponds to the maximum values of (η_b/λ), which are within the range of maximum torque (see Fig. 8).

The engine heating can be qualitatively estimated by the exhaust gas temperature (see Fig. 10). For the three fuels tested, the minimum values of exhaust gas temperature correspond to the maximum torque values, and the fueling corresponds to the maximum limit set for the enrichment of the air-fuel mixture so that smoke emissions follow the results showed in Fig. 11. Note that, at low engine speeds, smoke emissions are highly variable, which lead to high uncertainties.

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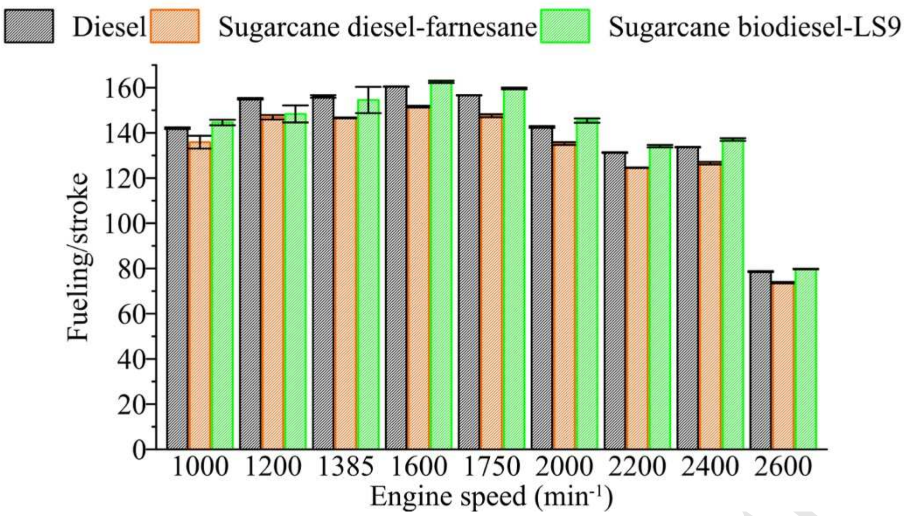

Fig. 9. Fuel injected per cycle.

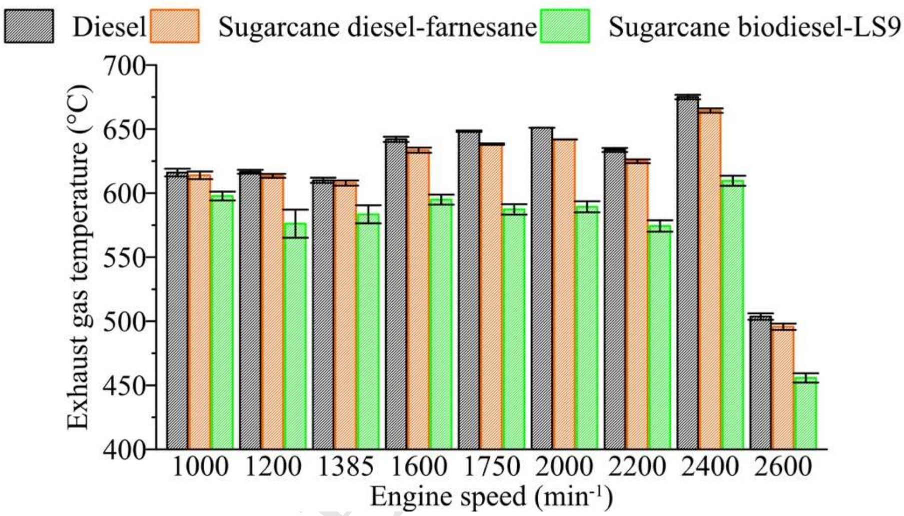

Fig. 10. Exhaust gas temperature at full load.

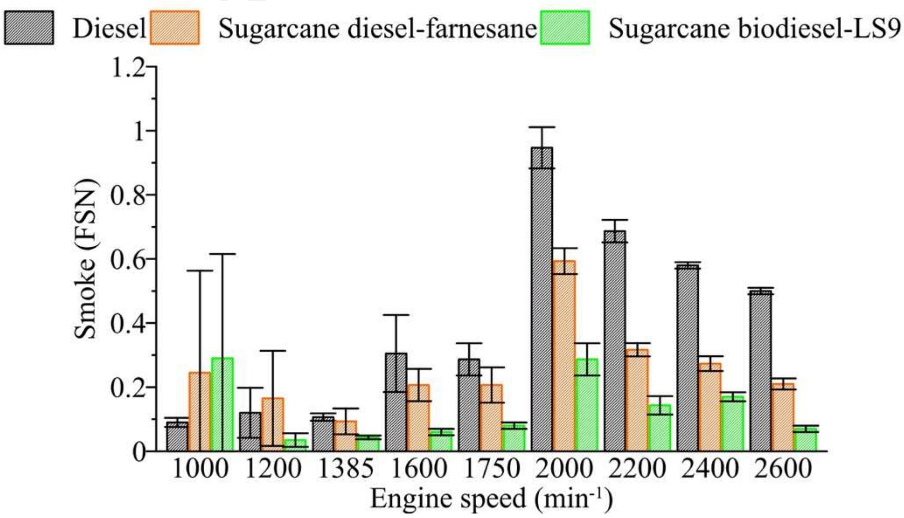

Fig. 11. Smoke emissions at full load.

According to Fig. 5, the biodiesel-LS9 shows the largest amount of exhaust gas recirculation. Nevertheless, biodiesel-LS9 oxygen content promotes a more complete combustion, which compensates its contribution to decrease the LHV of the fuel. On the other hand, at high torque operating regimes,

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which correspond to low and medium engine speeds (see Fig. 2), the amount of exhaust gas recirculation decreases below that of the diesel fuel. In this case, the second turbocharger (see Fig. 1), which is smaller than the first one, increases the amount of exhaust gas recirculation. This affirmation is supported by data provided in Fig. 5, where it can be observed that m_{EGR} values are the highest for the three tested fuels. In this situation, the ECU strategy didn't lead to increase the fuel consumption (kg/h) above that of the diesel fuel (see Fig. 3) but the ECU varied biodiesel-LS9 fueling (see Fig. 9) with the target of satisfying the limit of smoke formation. Consequently, the resulting air/fuel ratio promotes a more complete combustion leading to the smoke emissions and exhaust gas temperature showed in Fig. 11 and Fig. 10, respectively.

The considerable differences in smoke emissions and exhaust gas temperature observed at full load, between the reference diesel fuel and the biodiesel-LS9, supports the idea of modifying the ECU calibration in order to improve engine emissions and performance under these operating conditions. On the other hand, these differences are not significant for the case of sugarcane diesel-farnesane, therefore, this biofuel does not require an ECU recalibration.

The performance results presented above are not only a direct consequence of the physicochemical properties of the fuels. They are also a consequence of the ECU management. Thus, these results are not definitive for abandoning the use of the sugarcane biodiesel-LS9, despite the fact that the engine was designed to achieve a maximum power of 166 kW at 2400 min^{-1} and a maximum torque of 850 Nm from 1110 to 1600 min^{-1} (Table 1). To obtain this performance with pure sugarcane biodiesel-LS9, a change in the fuel delivery parameters or in the construction characteristics of the engine would be required, which would not prevent its use.

As shown in Table 2, the sugarcane biodiesel-LS9 has high cetane number, density and viscosity values, factors that justify the use of the biodiesel for many authors, as claimed in [28]. These authors also summarize that the biodiesel does not cause any loss in power unless its maximum values are demanded. For this reason, a comparative study of performance and emissions in experimental tests carried out under the same operating mode can be accepted. These tests can be executed in transient cycle conditions, which may prove the existence or inexistence of significant differences in engine performance under each test condition for all the tested fuels, as well as verify if specific emissions satisfy the standard's limitations. Nevertheless, it is appropriate to perform a previous analysis of the three tested fuels considering the same values of performance. These results are shown and discussed in Section 2.3.2.

2.3.2. Experimental results under the same values of performance

In order to obtain a better understanding of the effect of each biofuel over the engine, it is suitable to perform a comparison for the same effective power delivered by the engine. This analysis allows determining how the fuels affect exhaust emissions.

Six operating regimes were selected from the ETC test, which provide the same values of power for the three fuels tested (see Fig. 12), i.e. the criterion for selecting the operating modes was based on finding similar values of torque and engine speed for each fuel.

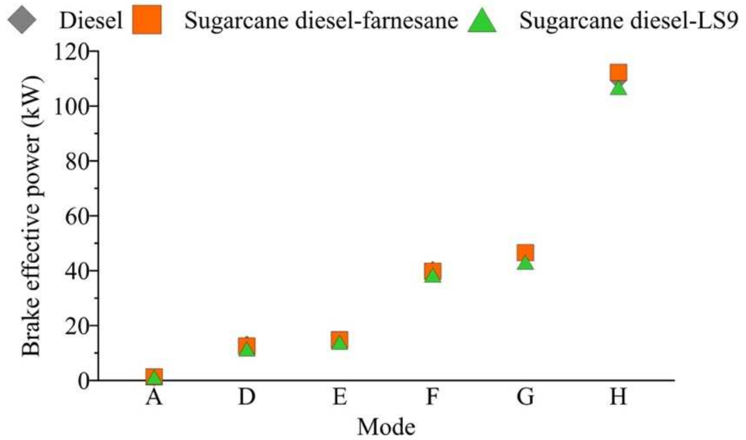

Fig. 12. Engine effective power at several operating modes for the three tested fuels.

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Three tests were performed per fuel. Table 4 shows the mean values of the parameters which characterize the selected operating modes.

Table 4. Parameters defining each of the steady state operating modes selected.

Mode	Engine speed (min^{-1})	Engine Torque (Nm)	Effective power (kW)
A	1065	9.8	1.2
D	2019	57.6	12.5
E	1928	68.6	14.4
F	1916	203.7	39.6
G	1723	260.7	45.3
H	1877	564.4	109.5

The *bsfc* for all tested fuels at each operating mode are shown in Fig. 13. The very low engine power delivered at mode A causes the highest values of *bsfc* from all the operating modes considered. Comparing both biofuels, *bsfc* of biodiesel-LS9 shows higher values, which is in concordance with its lower LHV. Only modes A and G show an inverse tendency due to its fuel consumption is slightly higher. Considering the same brake thermal efficiency, the variation in *bsfc* is caused by the variation in LHV of the fuels.

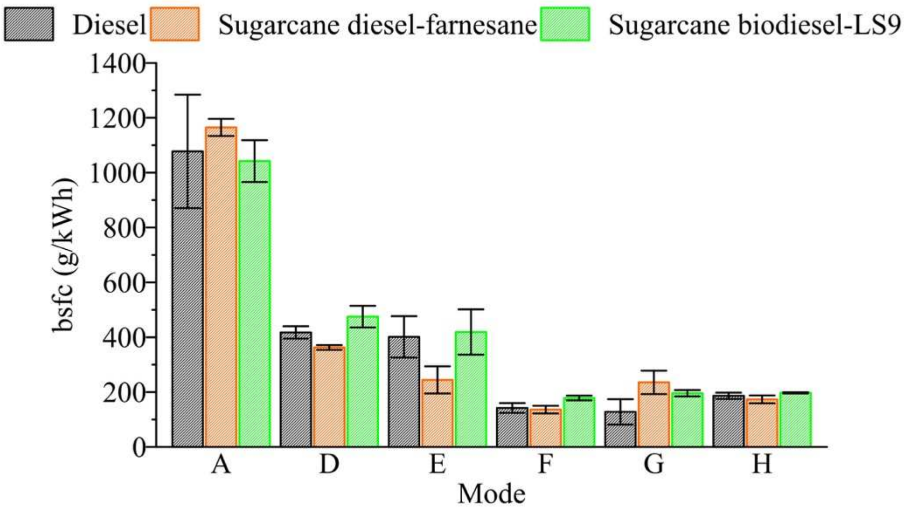

Fig. 13. Brake specific fuel consumption at each operating mode for the 3 fuels.

THC and NOx specific emissions for all tested fuels at each operating mode are shown in Fig. 14. The greatest increase in THC emissions is found for the mode A (the lowest load), which is caused by the low combustion temperature since it deteriorates the fuel atomization, evaporation, air-fuel mixing and the subsequent combustion process. This analysis is supported by the data provided in Fig. 15, which shows the lowest exhaust gas temperature in mode A.

According to previous authors [31], in comparison to mineral diesel fuel, the main factors contributing to decrease THC emissions are: the presence of oxygen in the fuel and the absence of aromatic compounds. This is the case of the biodiesel-LS9. Therefore, it is reasonable to observe that, for all operating modes tested, biodiesel-LS9 shows the lowest THC emissions, followed by the sugarcane diesel farnesane, which is also free of aromatic compounds.

Both biofuels can be considered as pure compounds. This affirmation is supported by the distillation results given in Table 3, which show a fairly homogeneous evaporation process. The sugarcane diesel-farnesane evaporates faster since this process occurs at lower temperatures. This is an advantage form the point of view of THC emissions reduction, but, based on the results obtained, it can be said that the oxygen content of the fuel is more significant in the THC emission reduction (note that the THC emissions of the biodiesel-LS9 are slightly lower).

Regarding mineral diesel S50, it is noteworthy the wide range of distillation temperatures and its high value at 90% of the distillation process. On the other hand, in the range from 10% to 50% of the distillation process, temperatures are significantly lower, which can be an advantage for the mineral diesel S50, as it will be demonstrated in the next section.

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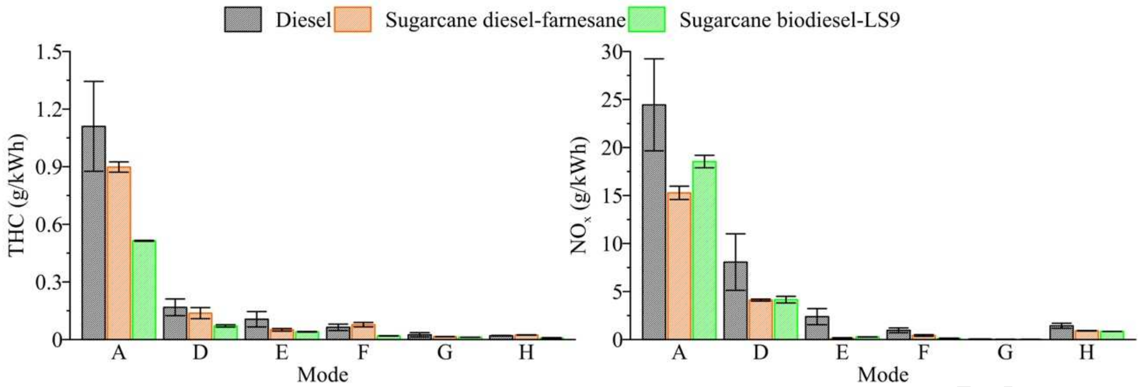

Fig. 14. THC and NO_x specific emissions.

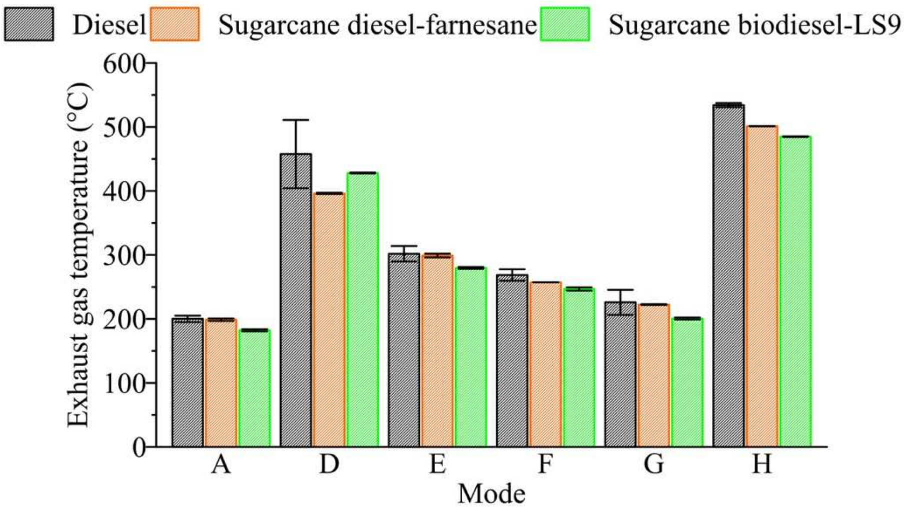

Fig. 15. Exhaust gas temperature.

In regard to specific NOx emissions, the high values obtained at low load (Mode A) are mainly a consequence of the low effective power. The biodiesel-LS9 leads to slightly higher NOx emission than the sugarcane diesel-farnesane, except for the mode A where this difference is significantly higher. This result can be explained by the presence of oxygen in biodiesel-LS9. Compared to the reference mineral diesel S50, the higher cetane number of biodiesel-LS9 (see Table 3) and the higher H/C ratio of sugarcane diesel-farnesane promotes a reduction of the adiabatic flame temperature [18], which leads to a reduction in NOx emissions. This reduction in the adiabatic flame temperature can be evidenced based on exhaust gas temperature experimental results provided in Fig. 15, where the temperature of the biofuels is never higher than that of the diesel S50.

Regarding specific CO emissions (see Fig. 16) a similar behavior to that of THC emissions can be observed (see Fig. 14), due to both emissions are a consequence of an incomplete combustion. CO formation in the cylinder increases mainly due to the decreasing in CO oxidation in the expansion and exhaust processes [32].

The biodiesel-LS9 shows lower CO emissions than diesel S50. Except in the case of mode A, these results are a consequence of the greater *bsfc* of the biodiesel-LS9 compared to that of the diesel S50 (see Fig. 13), which is directly proportional to fuel consumption since effective power is the same for all the tested fuels, as well as a consequence of the EGR regulation performed to satisfy the limit of air/fuel ratio stablished for promoting a complete combustion.

CO emissions of the sugarcane diesel-farnesane are within the range of the diesel S50 and the biodiesel-LS9, except in the case of mode E. A reduction in combustion temperature of the sugarcane diesel-farnesane, compared to the biodiesel-LS9, may lead to increase CO formation. This decrease in temperature can be caused by a decrease in the oxygen introduced in the cylinders [33] due to EGR regulation. Nevertheless, the exhaust gas temperature of the sugarcane diesel-farnesane is slightly higher than that of the biodiesel-LS9, except in the case of mode D, which can be explained by the lower LHV due to the presence of O_2 in the biodiesel-LS9.

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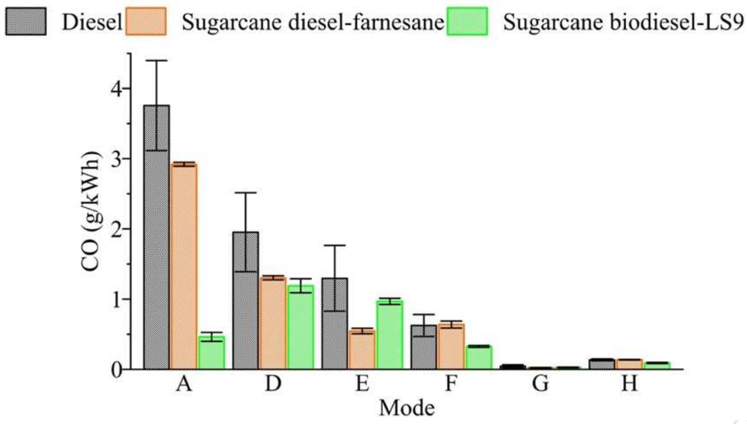

Fig. 16. CO specific emissions.

2.3.3. Transient cycle experimental tests ETC

The dynamometer experimental tests under European Transient Cycle (ETC) conditions provide greater precision and statistical confidence to determine small differences in performance and emissions. It is a driving cycle in which the position of the accelerator pedal is varied, i.e., a more specific analysis of the vehicle driving is done, in which fluctuations in acceleration and deceleration are considered. Different driving conditions are represented by three parts of the ETC, which are urban, rural and freeway driving. The duration of the entire cycle is 1800 s, in which the duration of each simulated driving is 600 s. The first part, from 0 to 600 s, represents urban driving at a maximum simulated speed of 50 km h^{-1}, with frequent starts, stops and idling. The second part, from 600 to 1200 s, represents rural driving, starting with a stage of steep acceleration. The average simulated speed is approximately 72 km h^{-1}. The third part, from 1200 to 1800 s, represents freeway driving at an average simulated speed of approximately 88 km h^{-1} [34].

Fig. 17 shows results for brake effective power, torque and engine speed as a function of time from the ETC experimental tests performed with the three fuels used in this study.

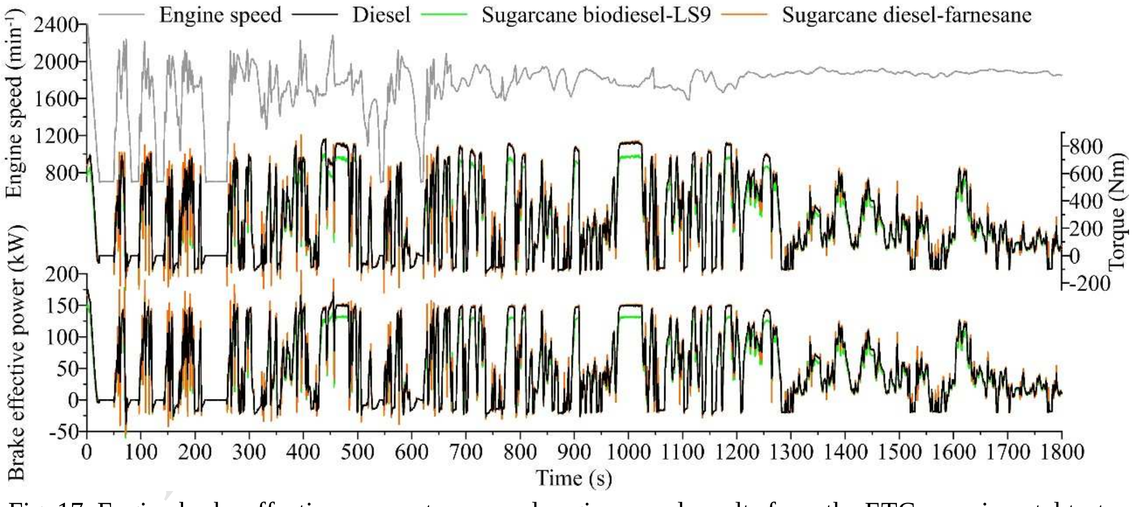

Fig. 17. Engine brake effective power, torque and engine speed results from the ETC experimental tests.

As shown in Fig. 17, the engine can reach each test condition with the sugarcane diesel-farnesane. With the use of sugarcane biodiesel-LS9, the engine cannot meet the maximum load values reached by the reference mineral diesel fuel, despite the difference in cumulative fuel consumption.

Aided by Fig. 18, it can be noticed an evident difference in cumulative fuel consumption during the entire test. For sugarcane diesel-farnesane, a lower cumulative fuel consumption is observed. This is in part due to the difference in the chemical formula of the fuel. The H/C ratio of the sugarcane diesel-farnesane is higher than that of the reference mineral diesel fuel (2.13 vs. 1.80), the reason why its LHV is slightly higher (43.43 MJ kg^{-1} vs. 42.03 MJ kg^{-1}) (Table 3). Consequently, the cumulative fuel

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consumption of the sugarcane diesel-farnesane is lower for all the test conditions. On the other hand, the opposite happens to the sugarcane biodiesel-LS9 due to its low calorific value.

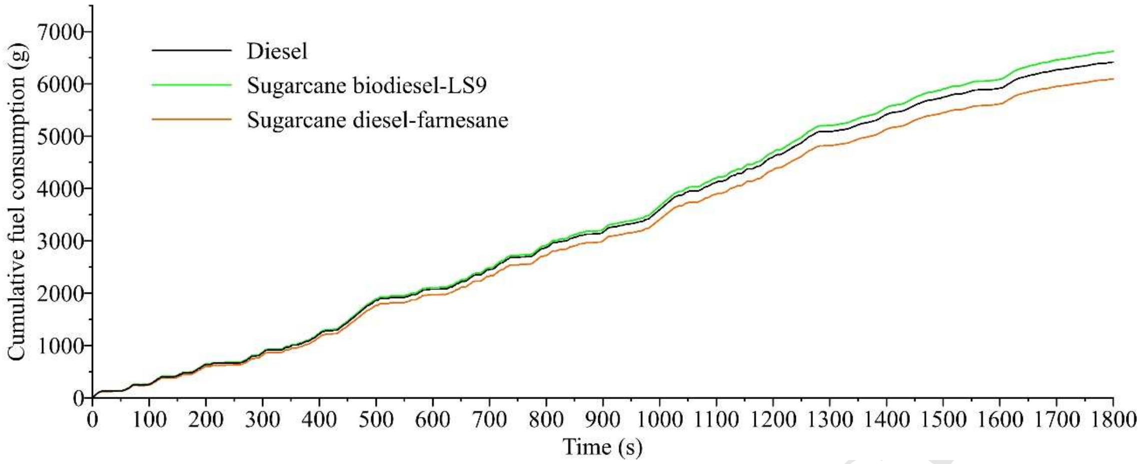

Fig. 18. Engine cumulative fuel consumption from the ETC experimental tests.

The trend shown by the cumulative fuel consumption during the urban driving cycle is maintained during the other two cycles. However, a noticeable difference is observed, as the sugarcane biodiesel-LS9 and the mineral diesel fuel lead to a higher cumulative fuel consumption. The sudden increases in fuel consumption are related to the way the ECU map reaches the specified engine speed [35]. These consumption values are caused by the engine speed and load, as higher loads lead to a higher variation in fuel consumption. At the same engine speed and load, lower LHV values cause higher fuel delivery.

Figs. 19, 20 and 22 reveal the instantaneous concentrations profiles of CO, THC and NO_x measured during the ETC experimental tests. In Fig. 19, it can be noticed that the sugarcane biodiesel-LS9 leads to a slight decrease in the CO concentration during the urban and rural cycles. During the freeway cycle, it causes higher concentrations compared to the other fuels. The concentration values for the sugarcane diesel-farnesane and the reference mineral diesel S50 are similar during the entire test cycle, with a small decrease during the freeway cycle observed for the biofuel.

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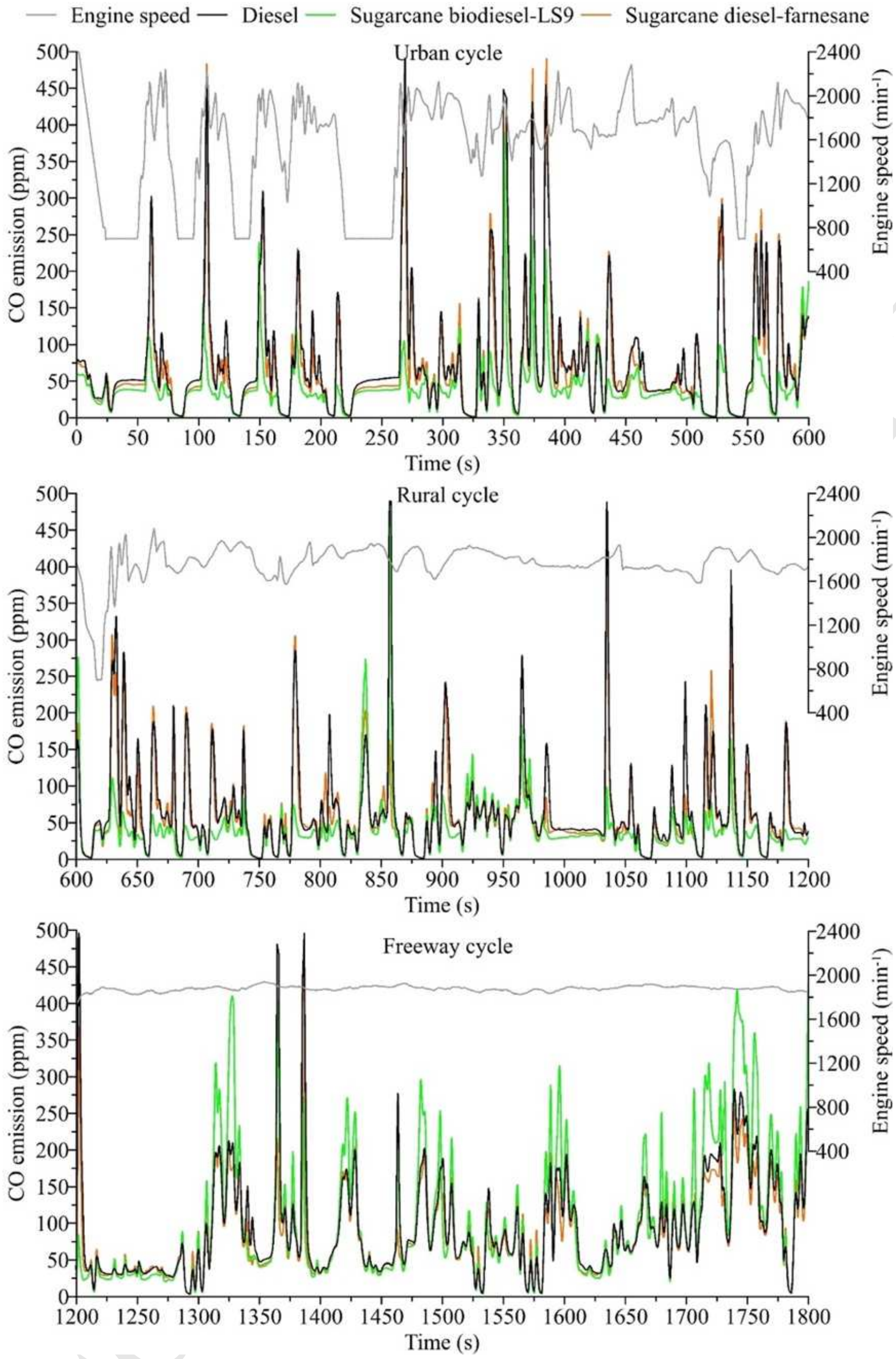

Fig. 19. Instantaneous concentrations profiles of CO measured during the ETC experimental tests.

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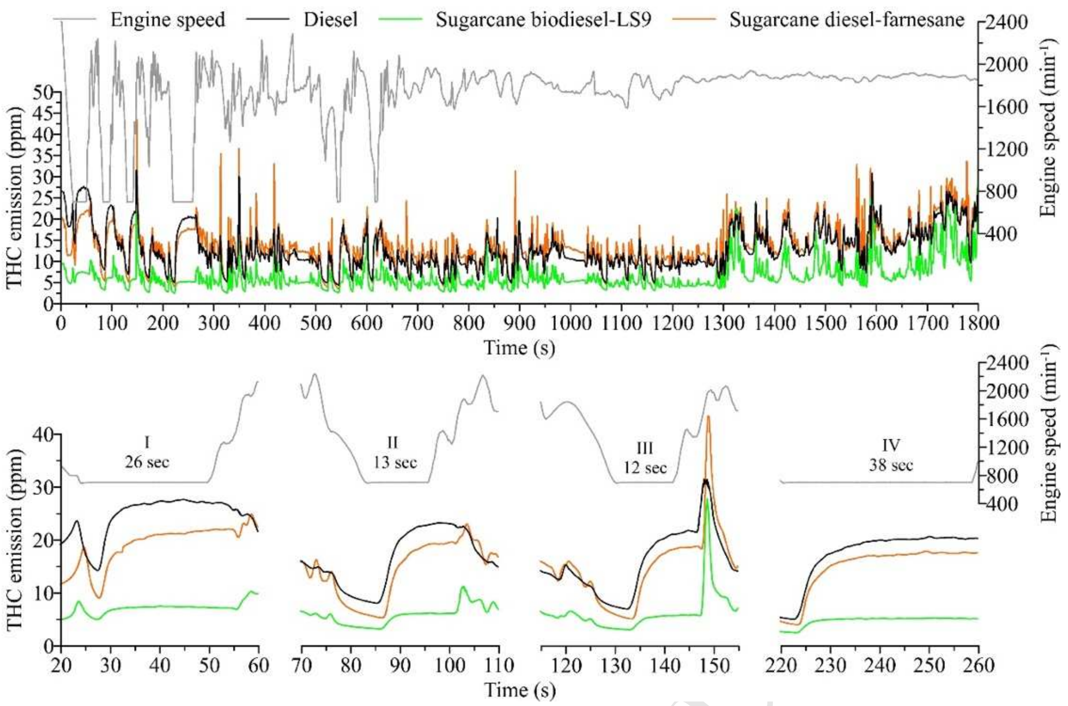

Fig. 20. Instantaneous concentrations profiles of THC measured during the ETC experimental tests, highlighting the most significant idling stages.

In regard to THC emissions (Fig. 20), for the entire test cycle, the sugarcane biodiesel-LS9 leads to a notable reduction. As for the sugarcane diesel-farnesane, a considerable increase in THC emissions in comparison to the reference mineral diesel fuel can be observed during the entire ETC experimental tests, except for a few time intervals such as when the engine is idling. In Fig. 20, the most significant idling stages in the cycle are highlighted: stage I that lasts 26 s, stage II that lasts 13 s, stage III that lasts 12 s and stage IV that lasts 38 s. In these stages, the behavior of the THC emissions is inverted, i.e., the sugarcane diesel-farnesane produces less THC emissions than the reference diesel fuel. Additionally, according to Figs. 8 and 10, at idle conditions both biofuels produce less CO and NOx than diesel S50.

The fact of lower THC, CO and NOx emissions for the sugarcane diesel-farnesane and sugarcane biodiesel-LS9 at idle conditions, compared to the diesel S50, supports the potential of these renewable fuels to significantly reduce exhaust emissions in urban areas, where engine idle conditions are common.

In some works [36, 37], reductions in THC and CO emissions were reported for biodiesel tested under transient conditions. It is argued that its oxygen content and higher cetane number promotes a more complete combustion. The results shown here match with this argument (see Table 3). For sugarcane diesel-farnesane, the increase in THC emissions and the non-reduction in CO emissions compared to the reference mineral diesel fuel can be an influence of the difference in boiling points [18], since sugarcane diesel-farnesane is a single compound (2,6,10-trimethyldodecane) and mineral diesel fuel is a mixture of various hydrocarbons. Some of these hydrocarbons are more reactive than farnesane, but others are heavier and less reactive. It appears that the combustion of farnesane is not as affected by the disturbance of the fuel spray caused by the idling conditions as the combustion of diesel fuel, which causes higher THC emissions. The fact that farnesane is a single compound leads to advantages over mineral diesel fuel, such as lower THC emissions at idling conditions (Fig. 20). However, it also leads to disadvantages, since some of the hydrocarbons in mineral diesel fuel are more reactive and volatile than farnesane, helping the pre-ignition, which favors the main stage of the combustion when the fuel spray is better structured under conditions other than idling. Namely, for some steady state operating conditions, such as idling and the aforementioned operating modes, THC emissions for the sugarcane diesel-farnesane are lower than those for the reference mineral diesel fuel. For highly transient regimes, which is the case of ETC tests, the rage of distillation from 10% to 50% shows significantly lower temperatures for the reference mineral diesel fuel compared to the sugarcane diesel-farnesane. This fact can promote the combustion of the reference mineral diesel fuel during the transient process which links two operating

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modes. This is the reason why exhaust gas temperature is higher for the reference mineral diesel fuel (see Fig. 21).

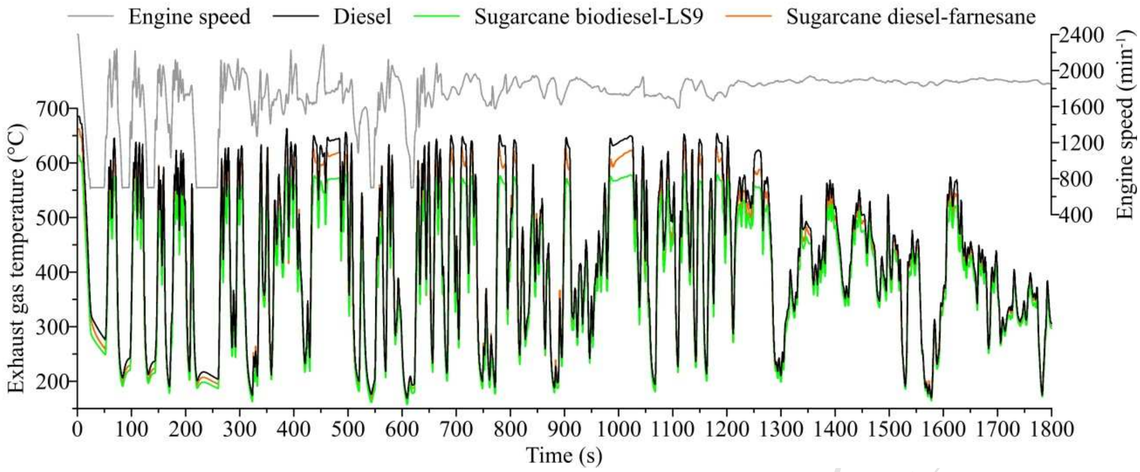

Fig. 21. Exhaust gas temperature during the ETC.

Fig. 22 displays the instantaneous concentrations profiles of NO_x measured during the ETC experimental tests. For the case of the sugarcane biodiesel-LS9, results show that the NO_x concentrations slightly rise during the urban and rural cycles and decrease during the freeway cycle in comparison to the reference mineral diesel fuel. Thanks to the action of the EGR controlled by the ECU, the NO_x emissions are not higher, since the other properties of the sugarcane biodiesel-LS9 would cause a considerable increase in these emissions, such as its higher cetane number, its lower volatility and its higher real in-cylinder flame temperature (mainly due to the difference in the soot radiation heat transfer) [38-41]. The lower NO_x emissions are mainly due to the lower oxygen concentrations and flame temperatures caused by the EGR [42], which is supported by the exhaust gas temperature results showed in Fig. 21. In this figure the lowest data are provided by the sugarcane biodiesel-LS9 for the whole ETC test. As for the sugarcane diesel-farnesane, a slight drop in the NO_x emissions is noticed during the entire ETC experimental tests, in comparison to the reference mineral diesel fuel.

The high NO_x concentration peaks seen for all the tested fuels are justified by the ECU strategy of most engines [43]. In this engine, during the acceleration sequences of the ETC experimental test, it appears that the tendency was to decrease the EGR valve opening to help increase the air-fuel ratio, increase boost pressure and limit the formation of smoke, as a result, during this phase an excess of nitrogen oxides is generally observed [36]. This criterion explains the peaks in NOx emissions observed during the acceleration sequences of the ETC experimental test in Fig. 22.

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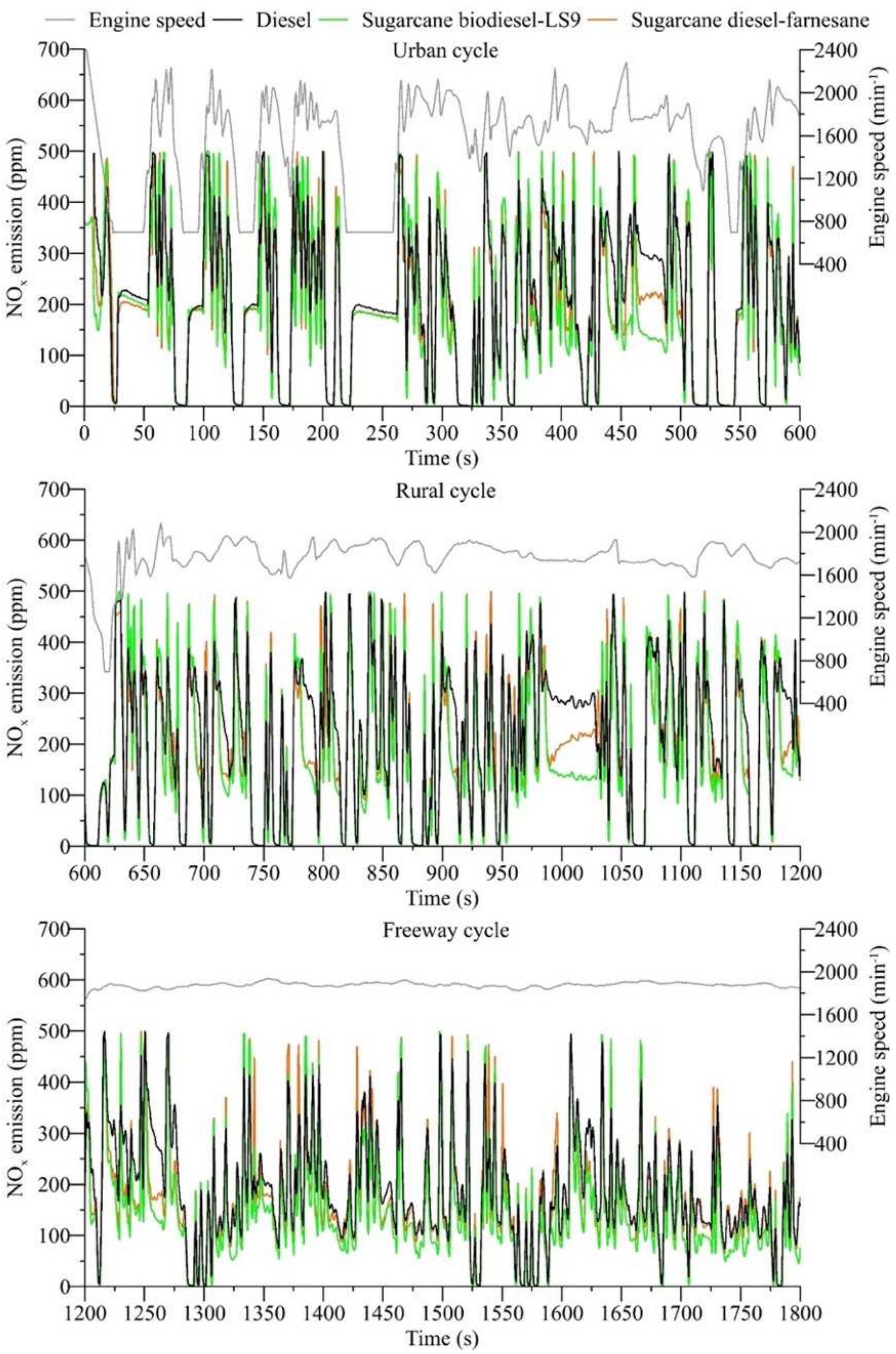

Fig. 22. Instantaneous concentrations profiles of NOx measured during the ETC experimental tests.

It is interesting to notice: (i) the behavior of the CO concentrations for the sugarcane biodiesel-LS9 is opposite to that of NO_x emissions; (ii) the sequence of characteristic transient accelerations in the urban and rural cycles of the ETC experimental tests, which are responsible for the significant peaks in emissions, favors the lower concentrations of CO and THC observed for the sugarcane biodiesel-LS9 (notice that when the engine enters the freeway cycle, which is characterized by an almost constant

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engine speed, the concentrations of CO and THC are slightly increased for this biofuel); (iii) comparing the emissions from both biofuels, the sugarcane biodiesel-LS9 shows more advantages, except for its CO emissions during freeway driving.

Following the same reasoning from Cárdenas et al. [43], as the engine works under the same demand of brake effective power, the emissions results presented above are not only a direct consequence of the fuel properties. The ECU control must be analyzed when the engine is powered with fuels characterized by different physicochemical properties in comparison to the reference fuel used by the automobile manufacturer to calibrate the engine to meet the emissions regulations and the CO_2 standards. Mainly, due to the lower LHV of the sugarcane biodiesel-LS9, it is necessary to increase the quantity of fuel injected to meet the power demand, as seen before, for which the driver must increase the accelerator pedal position. Most ECU management and control maps are controlled by the engine speed and by the accelerator pedal position or the fuel demand. Therefore, when fuels with lower LHVs are tested (as analyzed for the behavior of the cumulative fuel consumption), the engine needs an increased accelerator pedal position to deliver the same power. This fact has made the mechanism work instantly during the ETC experimental tests on different points of the ECU map for the sugarcane biodiesel-LS9 in comparison to the reference diesel fuel. It also occurred during the ETC experimental tests of the sugarcane diesel-farnesane but with a lower fuel demand due to its higher LHV in kJ kg^{-1}. Fig. 23 shows these differences in the behavior of the instantaneous fuel consumptions of both sugarcane biofuels and of the reference mineral diesel fuel. The sudden increases in fuel consumption are related to the way the ECU map reaches the specified engine speed [35], as seen before in the analysis of the cumulative fuel consumption.

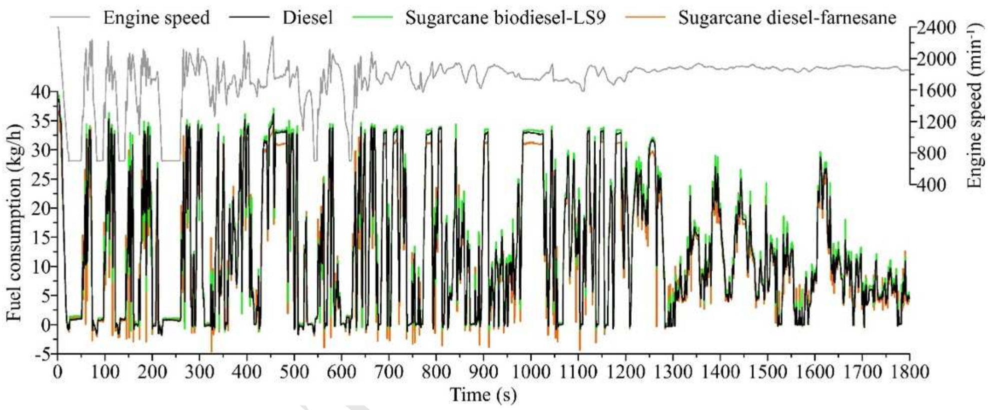

Fig. 23. Instantaneous hourly fuel mass consumption.

The smaller EGR valve opening for sugarcane diesel-farnesane is the cause of the higher intake air mass flow rate measured during the ETC experimental tests for higher engine speed demands, as shown in Fig. 24. It is noticed that the lowest instantaneous air flow rate values are also for the sugarcane diesel-farnesane. This is explained by the effect of the EGR valve when it changes from small to large openings. The sugarcane biodiesel-LS9 leads to a lower air flow rate compared to the reference mineral diesel S50 during the entire ETC experimental tests due to a larger EGR valve opening. The effect of the demand for a higher accelerator pedal position for the sugarcane biodiesel-LS9, since the quantity of fuel injected must be increased to achieve the power demand as seen before (Fig. 23), is related to the injection parameters. Again because of the ECU configuration, the higher accelerator pedal position for the same engine speed causes a higher injection pressure and a significantly earlier fuel injection timing (main injection timing) [43]. The higher injection pressure improves the fuel atomization, compensating for higher fuel viscosities. For sugarcane diesel-farnesane, which is less viscous, the fuel quantity injected is lower during the entire ETC experimental tests (Fig. 23). However, the air flow rate increases at higher brake power demands while it drops for lower brake effective power values and at idling conditions.

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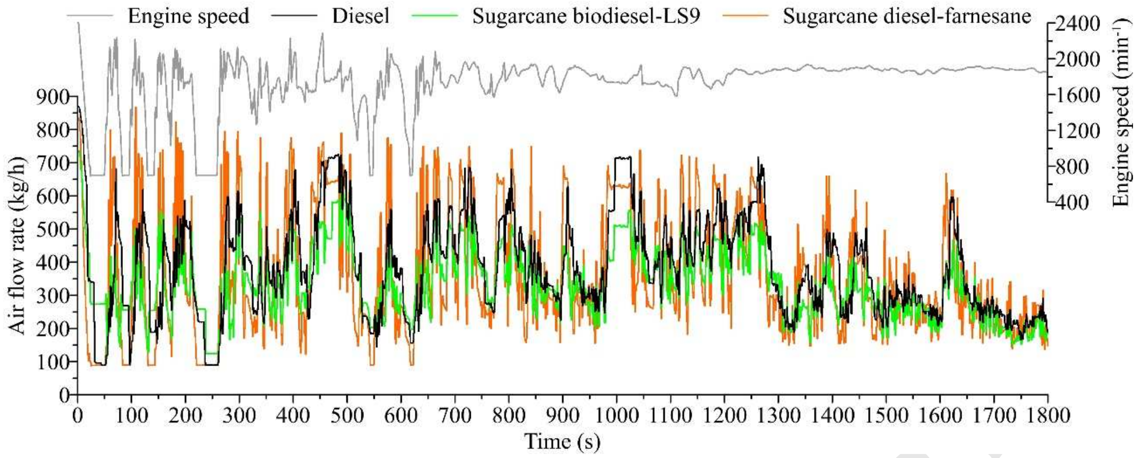

Fig. 24. Instantaneous intake air mass flow rate.

The sugarcane biodiesel-LS9 presented the lowest PM emissions, not only because it is an oxygenated fuel without aromatics, since the absence of aromatic compounds favors the reduction of particulates and NO_x formation as a result of high CN associated with the paraffinic structure [44], but also because of the earlier injection timing caused by the higher accelerator pedal position. This leads to higher in-cylinder temperatures, which help the oxidation of soot [36]. Without any drawbacks to this effect, the EGR valve opening increases to avoid the formation of NO, i.e., it opens as much as needed to keep the NO_x emission levels and lower soot emission values compared to the reference mineral diesel fuel. For the sugarcane diesel-farnesane, the lower fuel quantity injected leads to reduced PM emissions indeed, compared to the reference diesel fuel. In addition, it causes higher intake air flow rates which induces leaner air-fuel mixtures (since the EGR valve opens less due to the lower accelerator pedal position). This causes the NO_x emissions not to decrease compared to the reference mineral diesel fuel (the NO_x concentration values for sugarcane diesel-farnesane and reference mineral diesel S50 are similar).

3. Specific emissions during the ETC experimental tests

Specific emissions and PM over the ETC were obtained to verify the accomplishment of Brazilian standard ABNT NBR 15634. This standard follows the EURO V, so that exhaust emissions are measured downstream of the DOC and PM is collected before the DOC, as Fig. 1 illustrates. Table 5 shows the results obtained and the limit specified by the standard for each emission.

Table 5. Specific exhaust emissions, PM and standard limits.

	EURO V Limits (g/kWh)	diesel S50 (g/kWh)	sugarcane biodiesel-LS9 (g/kWh)	sugarcane diesel-farnesane (g/kWh)
NO_x	2.00	2.136	1.770	1.82
THC	0.55	0.007	0.011	0.01
CO	4.00	0.143	0.224	0.16
PM	0.030	0.044	0.010	0.01

Fig. 25 shows the average values of CO, THC, NO_x and PM specific emissions measured during the ETC experimental tests. Unexpectedly, the NO_x specific emissions for the reference mineral diesel S50 slightly exceed the regulation limits, while the PM specific emissions significantly exceed these limits. This result can be explained by the fact that the ECU was not calibrated for the Brazilian mineral diesel S50. The specific emissions for both biofuels behave quite similarly, except for CO. The sugarcane biodiesel-LS9 leads to higher CO specific emissions compared to the reference mineral diesel fuel, perhaps due to their different phase change characteristics. This may also have caused higher THC specific emissions [18], as a result of differences in volume losses as a function of temperature. Moreover, as the EGR increases, which is the case of the sugarcane biodiesel-LS9, the results are: (i) the combustion temperature decreases and the cooling area increases [45], leading to decrease THC oxidation

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in the expansion and exhaust processes, (ii) the amount of unburned air/fuel mixture stored at the fire piston ring could be grater due to the increment in the amount of EGR, this is because the recirculated gases push the air/fuel mixture to this space of the fire piston ring where the mixture does not burnt and then, during the exhaust stroke this mixture burns only partially, since the temperature and oxygen are not high enough to produce a complete combustion. Summarizing, the EGR technique decreases NO_x emissions but increases CO and THC emissions.

The THC specific emissions for the sugarcane diesel-farnesane are also higher than for the reference mineral diesel fuel. Different boiling characteristics of the sugarcane diesel-farnesane as a single compound (2,6,10-trimethyldodecane) in comparison to the mineral diesel fuel (a mixture of several hydrocarbons) may have caused higher CO and THC specific emissions [18].

Regarding NOx specific emissions, the main reason which contributes to its decrease lays on the lower combustion temperature reached with the use of the EGR system. This system manages to drastically reduce NO formation. The second reason is related to the reduction of O_2 concentration at the intake manifold [46]. Consequently, NO_x emissions are lower for the cases with higher EGR ratios.

An analysis should be highlighted concerning the PM specific emissions. Due to the lag in the response of the turbocharger under transient conditions (ETC), the real air-fuel ratio decreases, which causes local lack of oxygen in the cylinders and increases the PM emissions [47]. Thus, the oxygenated fuels lead to lower PM specific emissions, as seen in the results from Fig. 25, because their oxygen concentrations offset this effect. The PM specific emissions for the sugarcane diesel-farnesane are significantly lower than those of diesel S50 and, due to the lack of oxygen of the sugarcane diesel-farnesane, slightly higher than those of the sugarcane biodiesel-LS9.

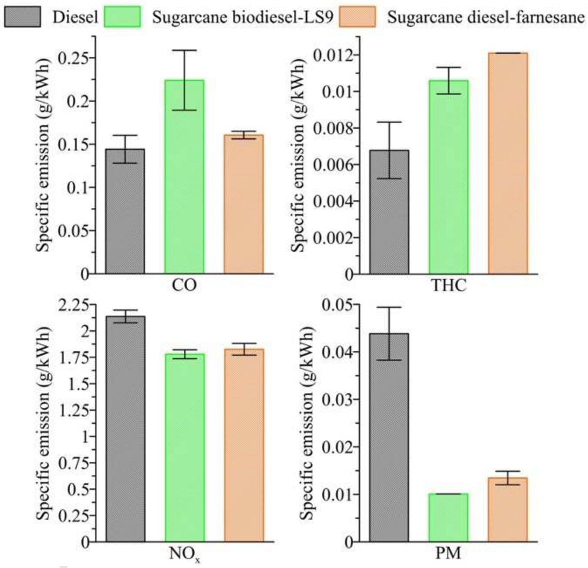

Fig. 25. Average values of CO, THC, NOx and PM specific emissions during the ETC experimental tests.

4. Conclusions

In the experimental tests at full load and variable engine speed, and without ECU modifications, it was corroborated that:

- Engine performance varies due to the variation in LHV, despite of the ECU management performed for each fuel tested.
- The sugarcane biodiesel-LS9 provides the highest value of the mechanical efficiency.
- No significant variations are observed in the $LHV/(A/F)_{st}$ and η_b/λ ratios when the three fuels under study are tested, which means that they don't influence the engine effective power.

In the experimental tests under the same values of performance, regarding the effect of the tested fuels on emissions:

- The sugarcane biodiesel-LS9 shows the lowest THC emissions, which is attributed to the fuel oxygen content. For the sugarcane diesel-farnesane, it is considered that the absence of aromatics compounds

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causes the reduction in THC emissions compared to the diesel S50.

- The oxygen content of the sugarcane biodiesel-LS9 leads to slightly higher NO_x emissions than those of the sugarcane diesel-farnesane. The higher CN and exhaust gas recirculation mass flow of the sugarcane biodiesel-LS9 and the higher H/C ratio of the sugarcane diesel-farnesane compared to the diesel S50 provide a NO_x reduction.
- Despite of the increment in *bsfc* and in %EGR for the sugarcane biodiesel-LS9 compared to diesel S50, the limits of the air-fuel ratio for a complete combustion are not exceeded. This is the reason why CO emissions are lower for the biodiesel. The lower combustion temperature of the sugarcane diesel-farnesane due to the variation of %EGR, which decreases O_2 in the mixture, may cause an increase of CO emissions during the first stage of the combustion process in comparison to the biodiesel-LS9.

In the ETC experimental tests:

- The sugarcane biodiesel-LS9 leads to a decrease in CO emissions during the urban and rural cycles in relation to the other fuels, which is a consequence of the highly transient conditions where the %EGR slightly varies due to the turbo-lag. During the freeway cycle, where the engine speed stabilizes, the incomplete combustion is promoted due to an increase in the %EGR. As for the sugarcane diesel-farnesane, CO emissions values are similar to those of the diesel S50.
- As for the THC emissions, the sugarcane biodiesel-LS9 leads to a notable reduction during the entire test. When the regimes are highly transient, the sugarcane diesel-farnesane shows increased THC emissions compared to the diesel S50, since the rage of distillation from 10% to 50% shows significantly lower temperatures for the diesel fuel and, consequently, higher exhaust gas temperature.
- Regarding the NOx emissions, the sugarcane biodiesel-LS9 causes an increase during the urban and rural cycles compared to the diesel fuel. During the freeway cycle, these emissions decreases, which demonstrates that the EGR acts to favor this reduction. As for the sugarcane diesel-farnesane, a slight drop in the NOx emissions is noticed during the entire test, in comparison to the diesel S50.

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Highlights

Two sugarcane biofuels are tested in a medium-duty diesel engine

Tests at stationary regimes and under the European Transient Cycle are carried out

Both biofuels produce less harmful emissions at idle conditions than diesel S50

Farnesane could improve performance and emissions of diesel S50 for recalibrated ECUs

Sugarcane biodiesel-LS9 can replace diesel S50 if ECU is recalibrated