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Integrated Solutions to Improve Combustion of Jatropha Oil in Diesel engine: Analysis of Engine Performance and Cyclic Dispersion

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ABSTRACT

This work focused on an integrated approach of solutions to improve the combustion of Jatropha oil in an indirect injection Diesel engine (Lister type). The improvements were assessed using combustion parameters, especially cyclic dispersion. The Jatropha oil was preheated to 100°C and the Diesel engine has received changes in fuel injection timing and injection pressure variation. The overall engine performance (specific fuel consumption, thermal efficiency and exhaust gas temperatures), cyclic dispersion, ignition delays obtained with Jatropha oil and Diesel fuel were compared. The results show that for the performance, the injection timing of 20 Crank Angle (CA) Before Top Dead Center (BTDC) and the injection pressure of 170 bars are the settings adapted for both fuels to the different engine loads. The results also show that cyclic dispersion could be used as a very good tool for assessment of adequate operating conditions of this type of engine, even at low loads. The cyclic dispersion is low with coefficients of variation of the indicated mean effective pressure (COV_{IMEP}) whose values are less than 10%. The lowest values of the COV_{IMEP} are obtained when the injection settings are 20 CA BTDC and 170 bars.

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1. Introduction

Crude Oil and its derivatives, natural gas and coal account for the largest share of the world most widely used primary energy sources. In 2018, they accounted for more than 80 % of primary energy sources used[1]. However, they are nowadays face to environmental problems (climate change, pollution, etc.), resource depletion and fluctuations in oil price. It is more than necessary to diversify the sources and develop other alternative and renewable sources. In this context, biofuels and more particularly vegetable oils are one of the possible solutions, like solar, wind, hydropower, etc. The use of vegetable oils in a Diesel engine has the advantage of producing electrical current and mechanical force for a variety of applications. In addition, vegetable oils are not subject to the source intermittency observed with renewable energy like solar, wind, etc. The use of vegetable oils is possible in a short cycle for rural populations in developing countries in particular. The combustion of vegetable oils in engines has been the subject of numerous studies [2–9]. Most of these studies highlighted that the direct use of vegetable oils in a Diesel engine, could lead to many problems if certain conditions are not met. The main problems encountered are carbon deposits on certain parts of the engine, power and thermal efficiency decreases, fuel over-consumption, fuel jet pumping and spraying problems, etc[9]. These problems are related to their physical and chemical nature (low volatility, fatty acid composition, high viscosity) and thermal conditions in the combustion chamber[2,4,9]. But in general, indirect injection Diesel engines are more suitablefor the use of vegetable oils than direct injection Diesel engines [2,10]. On this type of engine (indirect injection), the thermal conditions in the combustion chamber are higher compared to the direct injection Diesel engine for the same loads. Vegetable oils can be used in indirect injection Diesel engines without significant modifications for certain load. Although, at low loads, the problems already mentioned related to the use of vegetable oils appear [4,10].

Several technological solutions have been studied to overcome these problems. They include dual fuel systems which allow the engine to operate under favorable thermal conditions, modification of combustion chambers and pistons with refractory materials, etc[2]. These solutions are generally expensive and are not suitable for rural applications in most developing countries. Other less expensive alternatives and easier to implement, particularly for indirect Diesel engine, include fuel preheating (viscosity reduction), fuel injection timing and pressure modifications. These solutions are usually implemented individually with a study of overall performance and exhaust emissions rates [2,6,7,10,11], but rarely in an integrated form with a detailed analysis of combustion cycles.

The objective of this work is to determine the optimal conditions for using Jatropha oil in indirect injection Diesel engine Lister type at low loads with integrated solutions including: Jatropha oil preheating, fuel timing and injection pressure modifications and use cyclic dispersion in order to adequately assess the correct adjustment range for the use of vegetable oils on thistype of Diesel engine. The major contribution of this work is to adopt this integrated approach and use cyclic dispersion analysis as a criterion in addition to overall performance and conventional combustion parameters to determine optimal operating conditions.

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2. Materials and methods

2.1 Engine test bench

The tests were carried out at "Laboratoire de Physique et de Chimie de l'Environnement" of Université Joseph Ki-Zerbo (Burkina Faso). A Lister Diesel engine is installed on a test bench. It is a water-cooled indirect injection Diesel engine with a maximum power output of 7.35 kW. It is coupled, using transmission belts, to a generator that produces the electrical current necessary for the Diesel engine load. The characteristics of the engine and generator are givenin Table 1. The test bench consists of the engine, the current generator, a fuel consumption measurement system, a fuel preheating system, an engine combustion parameter acquisition system and a resistive load bench (consisting of 150W, 500W and 1000W lamps respectively used to apply an electrical load to the generator). The schematic diagram of the experimental set-up is given Fig. 1. For the measurement of the cylinder pressure, a KISTLER piezoelectric sensor type 6125C11 was used. The signals received from the sensor are amplified by a KISTLER 5011B charge amplifier. For the measurement of the injection pressure, a KISTLER piezo resistive sensor type 4067C3000A2 was used. This sensor is connected to a signal conditioner that amplifies the voltage variations measured by the sensor. The two signals from these sensors are collected by a National instruments NI 9215 acquisition module. Three type K thermocouples are used to measure the ambient air temperature, exhaust gas temperature and engine lubricating oil temperature. The thermocouples are connected to National instruments NI 9211 module. A KISTLER angle encoder type 2614C11 is used to clock the signals received from the various sensors with a resolution of 720 points per cycle. LabVIEW is used to record the different signals. The average sample selected is 100 cycles; this number offers a good accuracy for the calculations [12].

2.2 Test procedure and adaptations

Fig. 2 illustrates the procedure used for all tests. The characteristics of Diesel fuel and Jatropha oil are given in Table 2.

The engine operating parameters were modified; the injector setting pressure was varied from 130 to 170 bars in 20 bars steps. This step of variation of the injection pressure allows significant variations in the results[7,9,13,14].

The fuel injection timing used are: 15 CA BTDC, 20 CA BTDC and 25 CA BTDC. For the injection timing, the accuracy is 5 CA, which justifies the ranges of variations used. Other settings have been tested in preliminary tests, but they didn't give satisfactory results. The standard engine operating settings by the manufacturer are 20 CABTDC injection timing and 130 bars fuel injection pressure.

For each operating condition, tests were conducted first with Diesel fuel whose results were used as reference. Jatropha oil were preheated at 100°C, this temperature allows a significant reduction in the viscosity of the oil. Various studies have shown that preheating vegetable oils up to 100°C leads to engine performance that are comparable to those obtained with Diesel fuel [15–17].

Diesel fuel was used at ambient temperature. The combination of lamps makes it possible to obtain the different loads applied to Diesel engine. In this study, engine performance was compared for three loads: 26%, 43% and 70%.

The overall performances considered in this study are fuel specific consumption, thermal efficiency and exhaust gas temperature. The combustion parameters are ignition delay, rate of heat release and cyclic dispersion.

For the estimation of the fuel ignition delay, the start of combustion was identified with the injection curve and the ignition start of combustion corresponding of the first positive point of curve of heat release rate after the injection of fuel. The heat release rate was determined from equation (1):

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP}{d\theta} + \frac{dQ_{wall}}{d\theta} \quad (1)$$

With P the gas pressures in the cylinder, V the volume of the combustion chamber as a function of the crank angle, $\gamma = \frac{C_p}{C_v}$ the ratio of the specific heat capacities (C_p and C_v) and θ the angle. $dQwall$ the rate of heat transfer to the walls. This parameter is obtained from equation (2).

$$\frac{dQ_{wall}}{d\theta} = Ah(T - T_{wall}) \quad (2)$$

With A gas-wall contact surface, h heat transfer coefficient (in this work, the coefficient used is the one ofHohenberg), T the instantaneous gas temperature, Twall the temperature of the walls and ω angular speed.

Cyclic dispersion was evaluated using variations in the indicated mean effective pressure for the number of mean samples selected. The coefficient of variation of the indicated mean effective pressure is the most commonly parameter used for a quantitative evaluation of cyclic dispersion [18]. So, this coefficient, given by equation (3), were used in this work [18–21].

$$COV_{IMEP} = \frac{Std_{IMEP}}{\overline{IMEP}} x100 \quad (3)$$

WithStd$_{IMEP}$: the standard deviation of the indicated mean pressure and $\overline{IMEP}$: the average indicated mean effective pressure.

3. Results and discussion

3.1. Results of overall performance: Specific consumption, thermal efficiency and exhaust gas temperature

Fig. 3.a, b and c show the results of the engine specific fuel consumption for 26%, 43% and 70% of the load applied to the engine for the different pressures set and injection timing used.

The increase in fuel injection pressure leads to a reduction in the specific fuel consumption of the engine, regardless of the fuel used, engine load, injection timing and compared to the standard engine setting (Fig. 4).

This can be explained by an improvement in the fuel droplet size reduction and a better air-fuel mixture [4,22]. For this work, the Sauter Mean Diameters of the droplets are given in Table 3. Similar results have been obtained in previous work, and justify this finding by a decrease in droplet size when the fuel injection pressure increases [14,22–24].

There are still some differences between Jatrophaoil and Diesel fuel. There is an over-consumption of vegetable oil compared to Diesel, however, this over-consumption of fuel in the engine is reduced when the fuel droplet size is improved (preheating and increased injection pressure). Also, there is a reduction in Jatropha oil consumption ofthe engine when the injection pressure is increased compared to operation with Jatrophaoil at an injection pressure of 130 bars. These results are similar to those obtained in previous studies, which justify the over-consumption of engines operating with vegetable oils compared to Diesel fuel by their low lower calorific value [2,4,25]. Indeed, to provide the same energy, the engine consumes a greater quantity of vegetable oil. Increasing the injection pressure to 170 bars has a positive effect on the consumption of engines operating with vegetable oils. Variations in the engine injection timing didn't give satisfactory results. The specific fuel consumption of the engine is higher with late injection timing as well as with early injection timing.

Indeed, a late injection timing (15 CA) leads to low pressures in the engine combustion chamber and reduces the work with a cycle that looks like Joule cycle. These low pressures in the cylinder result in low power output with a consequent high mass consumption per unit of power [8,26,27].

In addition, early injection timingleads to high peaks by compression of the charge after ignition of the fuel, resulting in excess fuel consumption [26,27]. Consequently, the standard injection timingof the engine allows the lowest specific fuel consumption to be achieved. Similar results have been found in previous work [26,27].

Fig. 5.a, b and c show the results of the overall thermal efficiency of the engine operating with Diesel fuel and Jatrophaoil at different engine operating conditions.

The analysis of these results shows that the overall thermal efficiency of the engine increases with the load. Depending on the fuel injection timing and pressure set, the 20CA BTDC and 170 bars settings offer the best overall thermal efficiency for both fuels due to the same reasons given for the specific consumption. However, it should be noted that the differences obtained between the two fuels are in the order of magnitude of the uncertainties(about 5%).Indeed, for the same effective power, the energy deficit of Jatrophaoil (lower calorific value) is compensated by its overconsumption, giving overall thermal efficiency comparable between the two fuels[4]. An additional reason is the higher density of jatrophaoil compared to Diesel fuel that reduces friction losses and losses through cracks, thus improving the overall thermal efficiency of the engine [4,15].

Fig. 6.a, b and c show the exhaust temperature results of the engine under standard and modified conditions according to the three loads applied.

The engine exhaust temperatures increase when the load increases. This is due to the fact that the increase in load contributes to an increase in the temperature of the combustion chamber, which leads to an improvement of combustion conditions in the engine. Irrespective of the load applied, exhaust gas temperatures are relatively high with Jatropha oil compared to those obtained with Diesel fuel. At the standard engine setting, the exhaust gas temperatures obtained with Jatrophaoil are about 5% higher than those obtained with Diesel fuel at loads of 26 to 70%. This is due to the slow thermal decomposition of vegetable oils. So, the combustion is mainly diffusive, leading to heat release that continues until the beginning of the exhaust of the combustion gases.

The improvement in the spray quality of the fuel spray through the increase in fuel injection pressure leads to a decrease in exhaust gas temperatures for both types of fuel, regardless of the load applied to the engine. For Diesel fuel, with the increase in injection pressure to 170 bars, there is a 5% reduction in exhaust gas temperature compared to the standard engine setting. With Jatropha oil, the increase in injection pressure (170 bars) results in exhaust gas temperature comparable to those obtained at the standard setting of the Diesel engine. This is because the smaller the droplet size of the injected droplets, the easier it is for the fuel to vaporize when mixed with air to initiate combustion [9,28,29]. This increases the peaks of premix combustion by reducing the diffusing phase, thus lowering gas temperature during exhaust.

With Diesel fuel and Jatrophaoil, an increase in exhaust gas temperature is observed (10-13% for Diesel fuel and 10-15% for vegetable oil, respectively) when the fuel injection is delayed to 15 CA BTDC compared to the standard engine injection timing (20 CA BTDC). This result is the same for all loads. Indeed, when injection is delayed, a large part of the combustion occurs during the diffusion phase and continues through the exhaust, thus increasing the temperature of the engine exhaust gases [26,30,31]. At 25 CABTDC, it can also be noted that exhaust gas temperatures are relatively higher compared to those obtained with standard engine injection timing. Similar results have been obtained in previous work [26,27]. According to these authors, when the delay in injection timing is important, most of the heat is generated and released (earlier) before the end of the compression phase in the engine, and this heat can be dissipated either through wall losses or cracks and then released again during the last phase of the cycle (the exhaust) [26].

Considering the overall performance of the engine operating with these various modifications, the appropriate settings for injection timing and pressure are 20 CA BTDC and 170 bars respectively. However, the overall performance gives only information on a global scale of the engine operation and their use for appropriate adjustment of the engine is not very accurate. In the next point, combustion parameters are analyzed in order to better understand the combustion process for more accuracy in engine operating parameters adjustment.

3.2 Combustion parameters: ignition delay, heat release rate and cyclic dispersion

Fig. 7.a, b and c show the ignition delays for Jatropha oil and Diesel fuel, depending on the different operating conditions.

These figures show that ignition delays decrease with increasing engine load. This is due to the improved thermal conditions in the engine combustion chamber required for combustion as the load increases [4,26]. The ignition delays appear to be approximately the same for both fuels. However, depending on the modification or engine load, some disparities appear. For relatively low loads (26%), the ignition delay for Jatropha oil is higher than that of Diesel fuel, but for other loads (43% and 70%) this trend seems to be reversed. Some work have shown that for high loads (high cylinder gas temperature), the ignition delays of vegetable oils are shorter or equal to those obtained with Diesel fuel for certain types of engines [4,10].

It can also be noted that the ignition delay of the fuel in the engine decreases with increasing fuel injection pressure. This may be due to the improved quality of the fuel spray through finer droplets (which are more spontaneous in ignition) [4,26,28]. The shortest ignition delays were obtained at an injection pressure of 170 bars.

The ignition delay increases as the injection timing increases (25 CA). This is because the temperature and pressure in the cylinder are lower during compression if injection occurs early in the cycle, resulting in low evaporation and a long delay phase [7,26]. The fuel accumulates to reach its auto-ignition temperature, the air-fuel mixture is large, resulting in a large pressure rise during fuel ignition and the kinetic combustion peak is larger.

When the fuel injection timingis less delayed (compared to standard injection timing), a reduction in the ignition delay is obtained. Indeed, when the ignition is less delayed, the physical conditions are favorable for ignition of the fuel in the combustion chamber, and the fuel spray ignites instantaneously resulting in short delays [32,33]. However, combustion is initiated with a low rate of air-fuel mixing, which results in lower kinetic combustion peaks [8,26], and higher fuel consumption to provide the effective power required from the engine. This result is consistent with those obtained for the specific fuel consumption of the engine at 15 CA BTDC injection timing.

The analysis of fuel ignition delay in this type of engine shows that the standard setting (injection timing) is the most suitable for this type of engine. In addition, increasing the fuel injection pressure (170 bars) improves the ignition delay.

Fig. 8.a, b and c show the results of the combustion phases through the comparative heat release rate curves for a fixed injection pressure and for different fuel injection timing.

Analysis of the heat release rate curves obtained for the various settings shows that:

The heat release rate curves have similar trends for both fuels. After the ignition delay phase, the kinetic and diffusion combustion phases occur. Kinetic combustion peaks are more important for Diesel fuel (52.86 J/CA) compared to Jatropha oil (36.86 J/CA) at the standard engine setting. This is due to the low Diesel droplets size (compared to Jatropha oil) that vaporize easily (lower vaporization temperature) when mixed with air, resulting in a relatively higher rate of air-fuel premixing than in the case of Diesel. Thus, when this mixture ignites, the kinetic combustion peak is greater with Diesel fuel than with Jatropha oil. When the injection timing is late, compared to standard injection timing, the kinetic combustion peak is low (42.2 J/CA at 15 CABTDC for Diesel fuel). However, with early injection timing, the kinetic peak is much higher (61.44J/CA at 25CABTDC for Diesel). This result confirms the data obtained earlier for the ignition delays. The heat release rate curves obtained with Diesel fuel and Jatropha oil are sensitive to changes in fuel injection pressure. The increase in fuel injection pressure leads to an increase in the kinetic peak of combustion, in the case of Diesel fuel, its increases from 52 J/CA to 61.44 J/CA when the injection pressure varies from 130 to 170 bars. With Jatropha oil, this increase is less important, it varies from 37 J/CA to 39.28 J/CA when the injection pressure varies from 130 to 170 bars. In fact, the increase in injection pressure, as mentioned above, improves the spray quality of the fuel, which results in an increase in the quantity of fuel that can self-ignite [13]. However, the slow vaporization of vegetable oil when thermal conditions are not favorable is a limit to the premixing process. In the last phase of combustion (diffusion phase), a late injection timing leads to a more diffusive combustion. For Jatropha oil, the total amount of heat released during this phase is 1100 J (at 15 CA BTDC) compared to 960 J at 20 CA BTDC. This result confirms the high exhaust gas temperatures obtained when the injection timing is late. In conclusion, the increase in fuel injection pressure to 170 bars improves the atomization of fuel spray and subsequently the rise in heat (kinetic peak) in the engine. The early injection timing of fuel allows higher kinetic peaks, but this spontaneous heat obtained may not be beneficial during the cycle. Indeed, this heat release must occur at a specific injection timing to improve the work performed by the engine.

Fig. 9.a, b and c show the results of the coefficient of variation of the indicated mean effective pressure for all engine settings that were tested with the two types of fuels at different loads.

For all the tests carried out, the combustion cycles are regular because the values of the coefficient of variation of the indicated mean effective pressure do not exceed 10% for all the loads tested and regardless the fuel used (this limit is set by the literature) [18]. Previous work have shown that combustion in the type of engine used is more stable compared to the direct injection engine [34].

However, depending on the modifications made to the engine, the type of fuel, the load, the differences in cyclic dispersion become more or less important.

The cyclic dispersion decreases with the increase in the load applied to the engine; this is observed for both fuels used. At low loads (low combustion chamber temperature and long ignition delay), the fuel ignition processes are less controllable and therefore less repeatable, resulting in higher cyclic dispersion [35,36]. As the load increases, combustion chamber temperatures rise and fuel ignition delays are shorter, both of which lead to a decrease in the values of the coefficient of variation of the indicated mean effective pressure. Similar results have been obtained in previous work [26,37,38]. It should be noted that variations in the fuel injection timing compared to the standard value do not give satisfactory results regarding cyclic dispersion. As noted above, when the injection timing is early (25 CA), the fuel ignition delays are relatively long and the kinetic combustion peak is important. This is because the phenomena that occur during the premixing phase are random and less repeatable when the ignition delay is long [36]. Reversely, for late injection timing, the delays are relatively lowered (compared to other injection timing). Nevertheless, the coefficients of variation of the mean indicated effective pressure are high compared to the standard injection timing and overall engine performance is reduced. This results in a greater cyclic dispersion for this injection timing.

At all loads, the injection settings of 170 bars and 20°CA give the lowest values of the coefficients of variation of indicated mean effective pressure. These settings give relatively stable combustion, as cyclic dispersion is low, which is consistent with overall performance results. These results are in accordance with the literature [39]. As it can be noted, the standard injection timing (20 CA BTDC) gives the best results, this mean it has been optimized in this type of engine.

So, at 20CA BTDC, increasing injection pressure, which improves atomization and decreases droplets size, can lead to better combustion stability as observed in this work.

This reflects the importance of this parameter such as the coefficient of variation of the indicated mean effective pressure to improve the understanding of the phenomena that occur in the engine with macroscopic parameters such as overall performance.

4. Conclusion

The aim of this work was to determine the optimal conditions for using Jatropha oil in indirect injection Diesel engine Lister type at low loads with integrated solutions including: Jatropha oil preheating, fuel injection timing and pressure modifications and used cyclic dispersion in order to assess the adequate adjustment range for the use of vegetable oils on certain types of Diesel engines. Analysis of engine performance, combustion parameters and particularly cyclic dispersion shows that:

- Changing the fuel injection timing from the standard setting (20 CA) doesn't improve the engine overall performance (specific consumption, thermal efficiency and exhaust temperatures). The best overall engine performance was obtained for the standard injection timing (20 CA BTDC), with an increase in fuel injection pressure. These results highlighted the improvement in the overall performance of this type of engine when the fuel atomization conditions are improved (increase in injection pressure), particularly for vegetable oils.
- The fuel injection pressure has a small influence on ignition delays compared to the injection timing. For late injection timing, the ignition delays are shorter and for early fuel injection timing, the ignition delays are longer
- Heat release rate curves shown that the kinetic combustion peak varies with the fuel injection timing settings. Early injection timing increases the premixing rate, resulting in an increase in kinetic phase.

- The combustion of Jatropha oil on this type of engine is stable with values of the coefficient of variation of the indicated mean effective pressure not exceeding 10%. At 20 CA BTDC fuel injection timing, with an injection pressure of 170 bars, the cyclic dispersion is the lowest. Analysis of the cyclic dispersion makes it possible to consolidate the results obtained for the overall performance of the engine, i.e. the settings suitable for Jatropha oil combustion in this type of engine.

- The coefficient of variation of the indicated mean effective pressure is a good tool for assessing the combustion of vegetable oils in Diesel engines.

In view of the analysis of the engine performance and combustion parameters, it appears that the settings allowing a fairly good use of vegetable oils on this type of engine are 170 bars for the injection pressure and 20 CABTDC for the fuel injection timing.

Table 1. Engine and generator characteristics

	Characteristics	**Specification**
Engine	Type	Lister, water cooled, indirect injection, single cylinder
	Bore × stroke (mm)	120 ×139.7
	Standard fuel injection pressure	130 bars
	Standard fuel injection timing	20 CA BTDC
	Rated power (kW)	7.35 kW
	Rated speed (rpm)	1000
	Compression ratio	17 :1
	Conrod (mm)	307
	Injection pump	BOSCH
	Type injection	Indirect
Generator	Type	STC
	Rated power (kW)	8 kW à 1500 rpm
	Cos φ	0,8
	Rated speed (rpm)	1500

Table 2. Characteristics of Diesel fuel and Jatropha oil

Characteristics	**Methods**	**Diesel fuel**	**Jatrophaoil**
Kinematic viscosity at 40°C (cSt)	ASTM D445	2.44	35.98
Density at 20 °C (kg/m^3)	ASTM D 1298	840	917
Conradsonresidue (%) [25]	ASTM D 189	0.1	0.8
Lower Calorific Value (kJ/kg)	ASTM D 240	43700	39071
Flash point (°C) [25]	ASTM D 97	71	229
Cloud point (°C)	ASTM D 97	-6	4

Table 3. Values of SauterMean Diameter (SMD) at different injection pressure for Diesel fuel and Jatropha oil

	Sauter MeanDiameter (µm)	
Injection pressure (bar)	Diesel fuel	Jatrophaoil
130	41	82
150	36	75
170	27	70

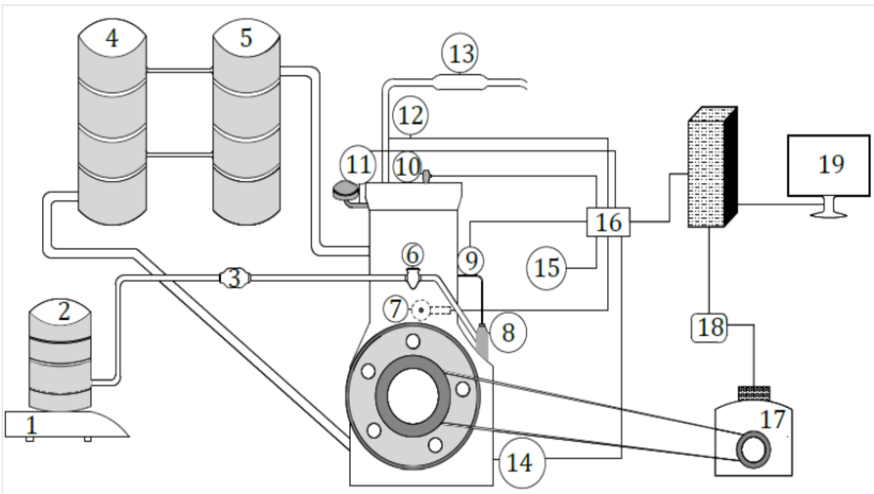

Fig 1. Schematic diagram of the experimental set-up

(1) balance; (2) tank; (3) fuel filter; (4) hot water barrel; (5) cold water barrel; (6) fuel filter; (7) angle encoder; (8) fuel injection pump; (9) injection pressure sensor; (10) cylinder pressure sensor; (11) Intake pressure sensor; (12) Exhaust thermocouple; (13) Exhaust system; (14) Lubricating oil thermocouple; (15) Thermocouple for ambient temperature; (16) Acquisition rack; (17) Generator; (18) Power analyzer; (19) Computer

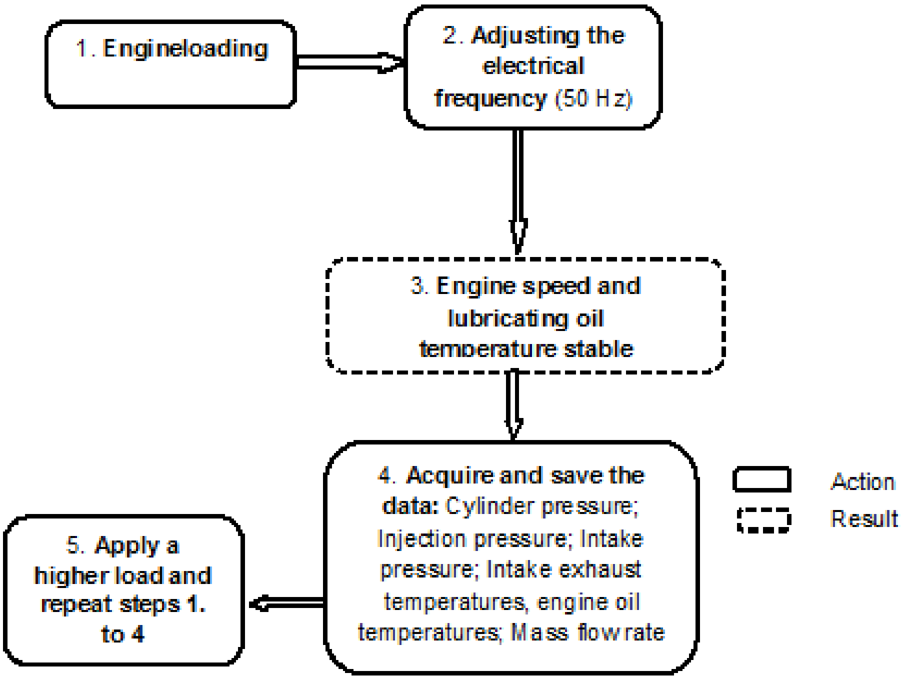

Fig 2.Test procedure

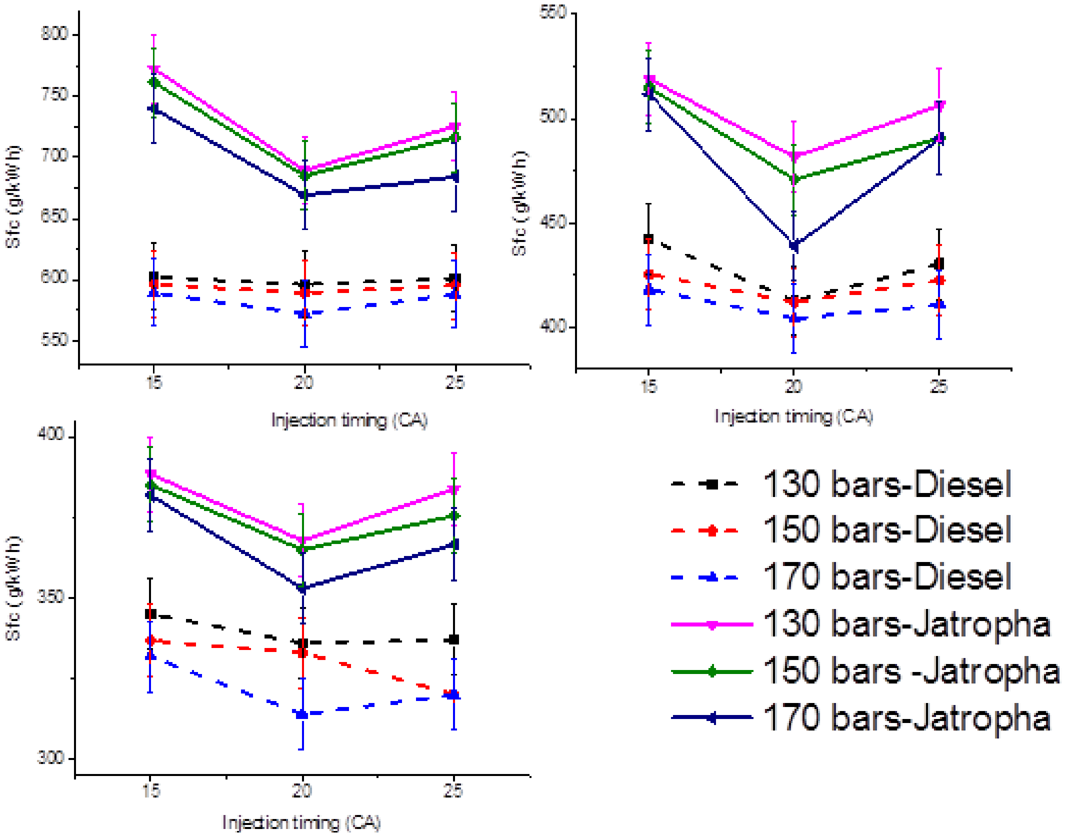

Fig 3. Specific fuel consumption of the engine as a function of the injection timing for different injection pressures for the 3 loads: (a) 26%, (b) 43% and (c) 70% of load

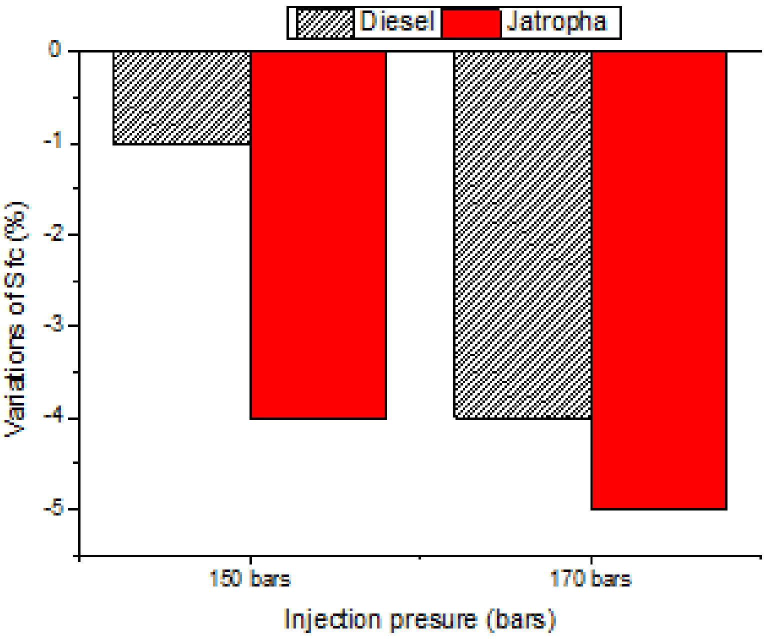

Fig 4. Variation of engine Sfc as function of injection pressure compared to standard pressure

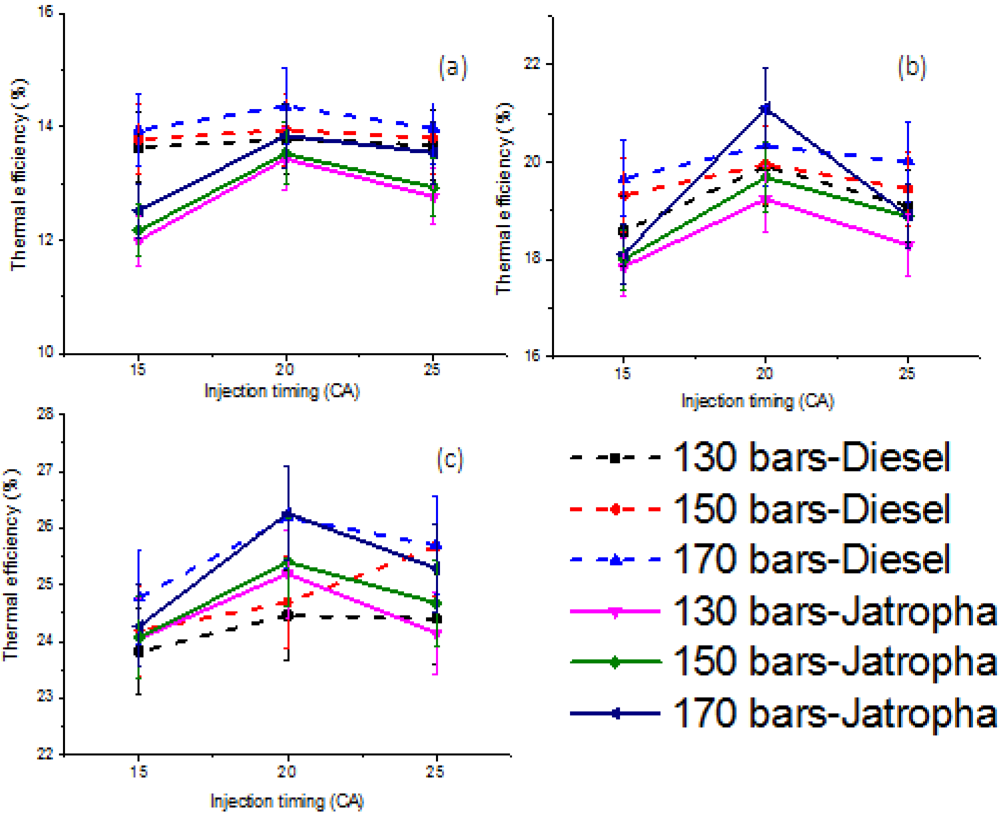

Fig 5 .Engine thermal efficiency as a function of the injection timing for different injection pressures for the 3 loads: (a) 26%; (b) 43%; (c) 70%

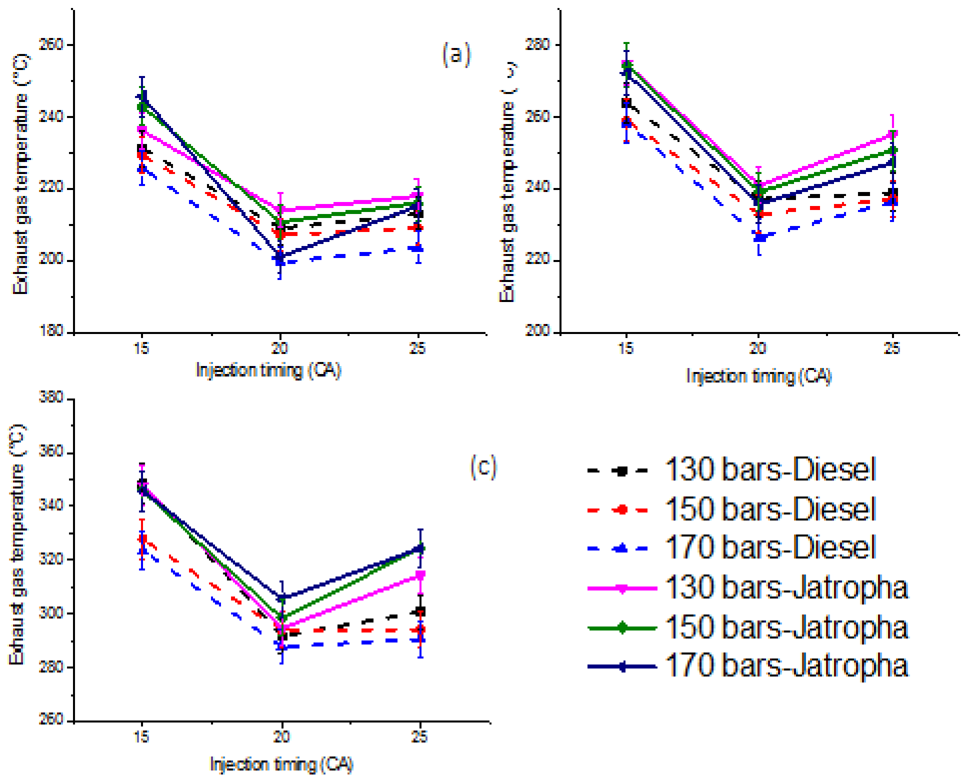

Fig 6. Exhaust gas temperatures as a function of the injection timing for different injection pressures with the 3 loads: (a) 26%; (b) 43%; (c) 70%

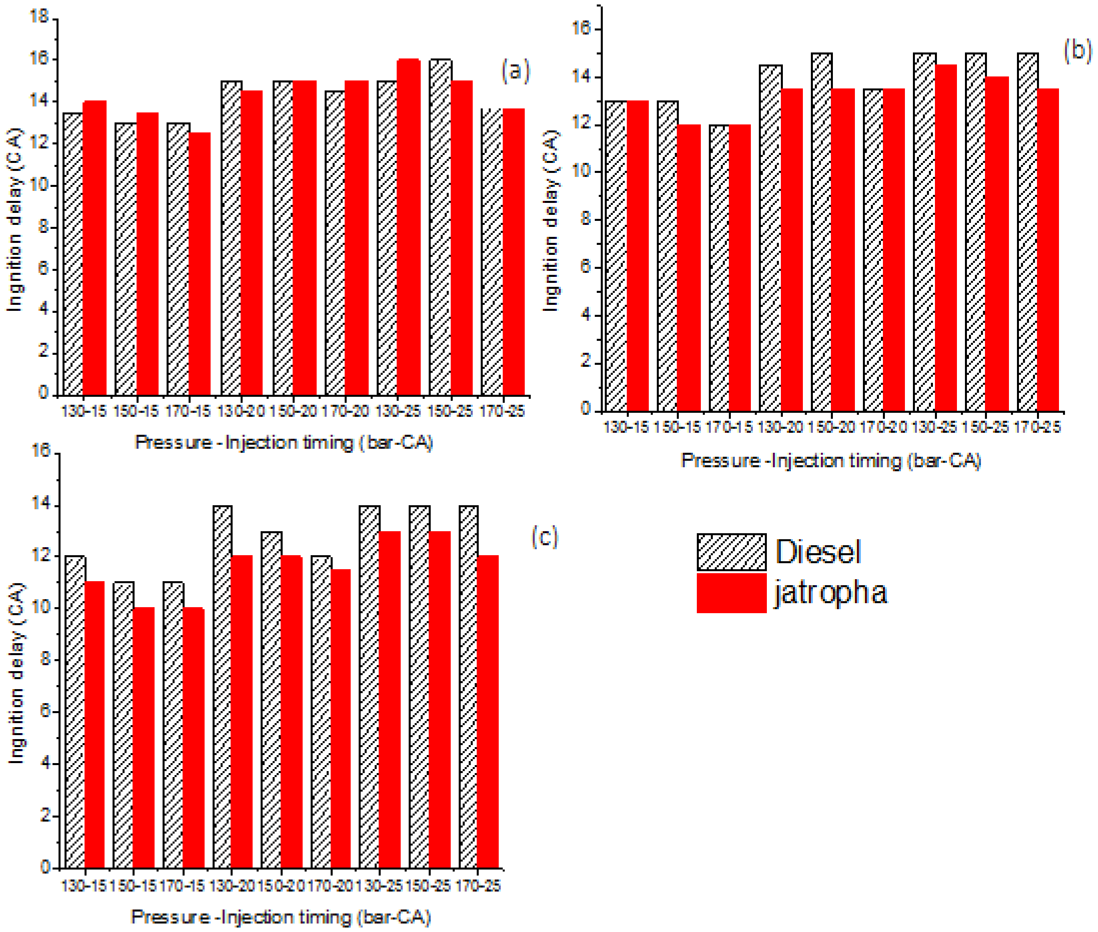

Fig 7. Ignition delays as a function of the different settings (pressure and injection timing) with the 3 loads: (a) 26%; (b) 43%; (c) 70%

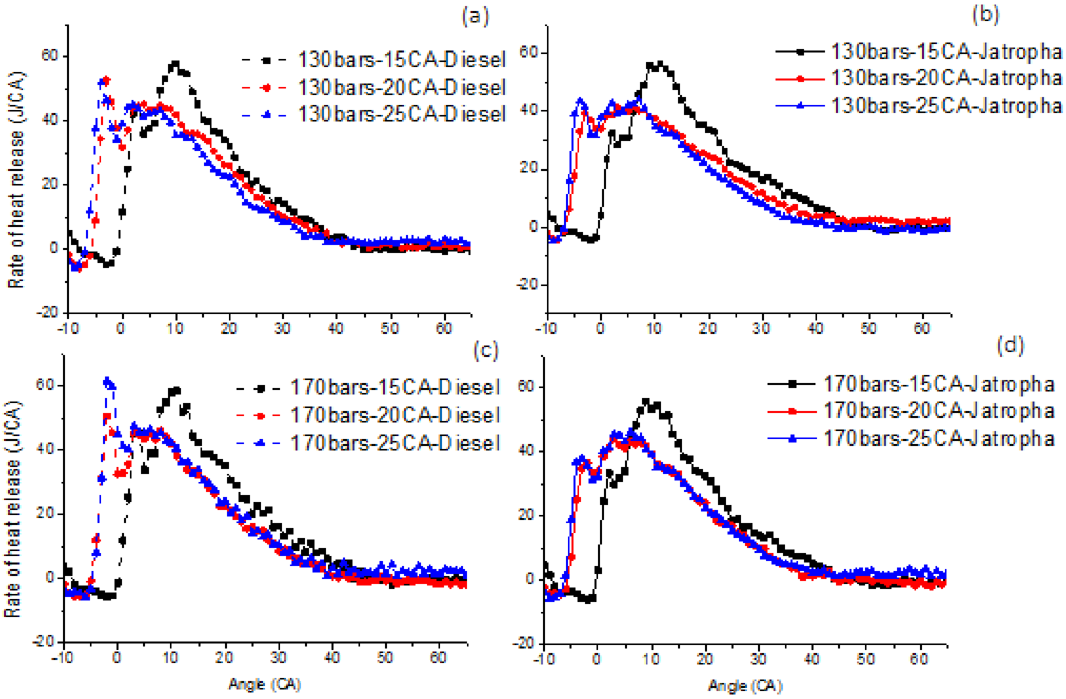

Fig 8. Heat release rate of engine as function of crank angle at different settings: (a) Diesel at standard injection pressure; (b) Jatropha oil at standard injection pressure; (c) Diesel at 170 bars injection pressure; (d) Jatropha oil at 170 bars injection pressure

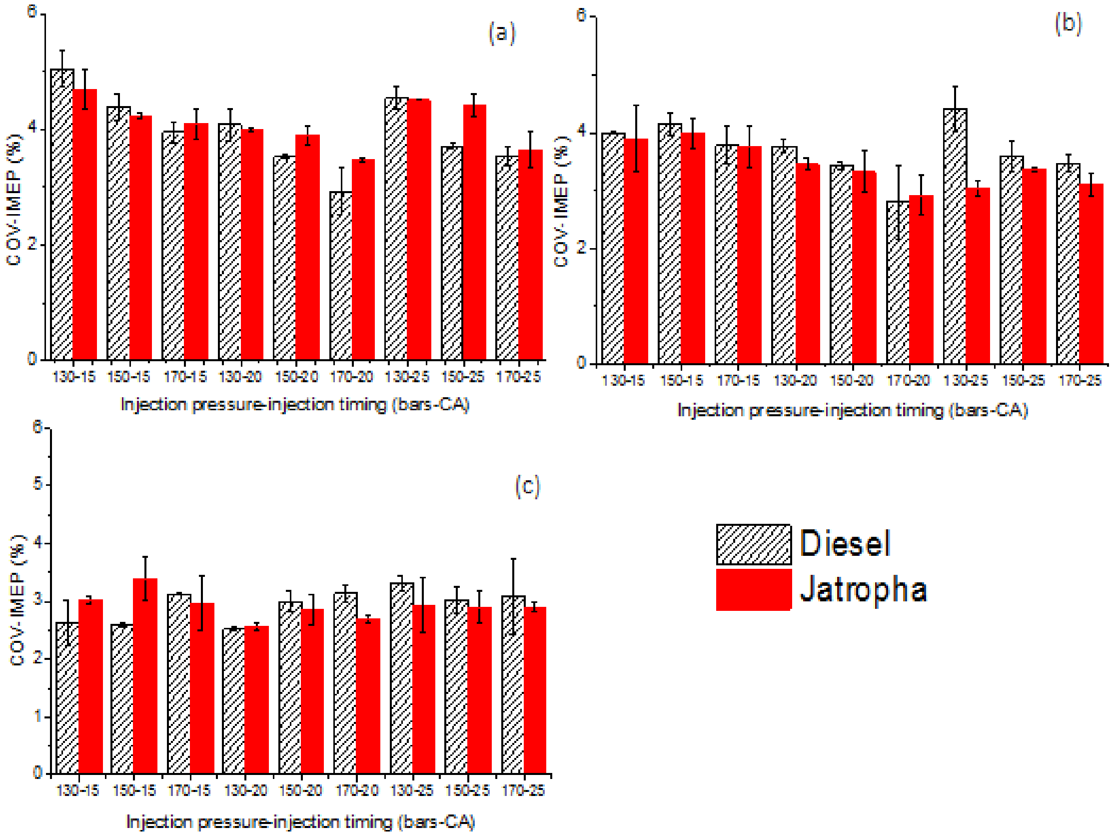

Fig 9. COV$_{IMEP}$ as a function of the different settings (pressure and injection timing) with the 3 loads: (a) 26%; (b) 43%; (c) 70%

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Abbreviations

BTDC	**Before Top Dead Center**
CA	**Crank Angle**
COV	**Coefficient Of Variation**
D	**Diesel**
IMEP	**Indicate Mean Effective Pressure**
J	**Jatropha oil**
Sfc	**Specific fuel consumption**