Experimental investigation on the influence of a biodiesel (waste cooking oil) on the performance and exhaust emissions of a compression ignition engine

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ABSTRACT. *In this work, experimental investigations are carried out to evaluate the performance and efficiency of a biodiesel-fuelled engine. The tests were conducted on a direct-injection compression ignition engine, equipped with two acquisition stations, one for measuring high frequency signals (e.g. in-cylinder pressure), the other for measuring low frequency signals (e.g. engine speed). Then, thermodynamic analysis of the data were implemented to calculate the heat release rate and to estimate the losses of heat. The results exhibit that the WCO biodiesel presents a comparable performance to those of standard diesel fuel. The biodiesel combustion is much rapid which leads to a slight reduction in NOx emissions. Furthermore, the presence of oxygen in biodiesel molecule conducts to more complete combustion and reduces the pollutant emissions of the engine.*

RÉSUMÉ. *Dans ce travail, une étude expérimentale est effectuée pour évaluer les performances et l'efficacité d'un moteur alimenté par le biodiesel. Les essais ont été menés sur un moteur à allumage par compression à injection directe, équipé de deux stations acquisitions, l'une pour mesurer les signaux de haute fréquence (e.g. la pression dans le cylindre), l'autre pour mesurer les signaux de basse fréquence (e.g. la vitesse du moteur). Ensuite, une analyse thermodynamique des données a permet le calcul du taux de dégagement de chaleur et d'estimer les pertes de thermiques. Les résultats montrent que le biodiesel extrait des huiles de friture usagées (HFU) présente des performances comparables à celles du carburant diesel standard, cependant la combustion du biodiesel est très rapide, ce qui entraîne une légère réduction des émissions de NOx. De plus, la présence d'oxygène dans les molécules de biodiesel conduit à une combustion plus complète et réduit les émissions polluantes du moteur.*

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KEYWORDS: biodiesel, waste cooking oil, diesel engine, heat release analysis, emissions.

MOTS-CLÉS: biodiesel, huiles de friture usagées, moteur diesel, analyse du dégagement de chaleur, Émissions.

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1. Introduction

During the last decades, the over-exploitation of fossil fuels and their negative impact on the environment require an urgent need for alternative energy sources to meet sustainable energy demand, with a renewable source and a lesser impact on the environment, e.g. solar energy, alternative fuels.

In order to reduce the consumption of diesel fuel and its pollutant emissions, while maintaining the same engine performance, many researchers have tried to develop alternative fuel that are similar to the properties and performance of diesel fuel. To deal with this problem, several solutions are proposed, one of the solutions is to mix petrol-diesel with an alternative diesel (dual fuel). The idea is to inject an alternative fuel as gas (methane, Oxyhydrogen) in the admission collector to blend with the air; the pre-mixed air-gas mixture is sucked thereafter into the cylinder and it is burned by petrol-diesel injected into the combustion chamber (Silvana *et al.*, 2016). The results show that there are reductions in polluting emissions with conservation of the same energy efficiency of engine. Another solution used to reduce petrol fuel consumption is to substitute petrol-diesel with alternative diesel like biodiesel, research has shown that fatty acid esters (biodiesel) resulting from the transesterification of vegetable oils have properties similar to those of petrol-diesel fuel (Enweremadu et Rutto, 2010; Dorado *et al.*, 2004).

Literature, illustrates several advantages of the biodiesel use: it helps to reduce the carbon dioxide emissions to the atmosphere, in addition to its renewability in the nature and safer to handle, furthermore, it has no aromatic compounds, practically no sulphur content (Naima & Liazid, 2013). Besides, Oxygen atoms in the molecule of biodiesel fuel may reduce the emissions of pollutants such as; carbon monoxide (CO), total hydrocarbon (THC) and particulate matter (PM) (Lapuerta *et al.*, 2002; Alamu *et al.*, 2008). In addition, Brake thermal efficiency (BTE) and combustion temperature of biodiesel are lower than those of diesel, resulting in a little decrease in NOx emission of biodiesel compared to diesel is observed at low loads and medium engine speeds. However, it is known that biodiesel have some drawbacks compared to petroleum diesel fuel such as; worse low temperature properties, great emissions of some oxygenated hydrocarbons, high specific fuel consumption, decrease in brake thermal efficiency and important production cost (Dunn *et al.*, 1996; Canakci & Van-Gerpen, 2001).

The problem of production cost has been partially solved by the use of waste vegetable oils or animal fats as raw materials in the transesterification process (Dorado *et al.*, 2003; Awad *et al.*, 2013). However, during frying the vegetable oil various physical and chemical changes are undergoes and many undesirable compounds are formed, including free fatty acid and some polymerized triglycerides which increase the molecular mass and reduce the volatility of the oil (Enweremadu et Mbarawa, 2009). Therefore, fatty acid esters obtained from frying oil affects the fuel properties (e.g. viscosity that reduces the burning characteristics) leading to a high amount of carbon residue (Kulkarni et Dalai, 2006).

Currently, the high cost of biodiesel fuel is a major barrier to its commercialization compared to the petroleum diesel fuel. It is well known that approximately 70-85% of the total biodiesel production cost comes from the cost of the raw material (Xiangmei *et al.*, 2008). The use of low-cost feedstock such as WCO should help to produce a biodiesel fuel competitive in price with petroleum diesel. Numerous studies have been conducted on biodiesel production and emissions testing in the past two decades. Most of the current challenges are targeted to reduce the biodiesel fuel production cost, since it is still higher than the petroleum diesel. This opens a golden opportunity to use the WCO as a feedstock for production (Cheikh *et al.*, 2016). Everywhere in the world, there is an enormous amount of waste lipids generated from restaurants, food processing industries and fast food shops every-day. In China, with annual consumption of edible oils approaching 22 million tons the country generates more than 4.5 million tons of used oil and grease each year, roughly the half of which could be collected through the establishment of an integrated collection and recycling system (Meng *et al.*, 2008). Those 2 million tons of "ditch oil" alone would guarantee the smooth operation of all current biodiesel production lines. Reusing of these waste greases cannot only reduce the burden of the government in disposing the waste, maintaining public sewers and treating the oily wastewater, but also lower the production cost of biodiesel significantly. Furthermore, biodiesel fuel has been demonstrated to be successfully produced from waste edible oils by an alkali-catalyzed transesterification process, and can be considered as alternative fuels in diesel engines and other utilities (Mittelbach & Gangl, 2001; Al-Widyan & Al-Shyoukh., 2002a; Al-Widyan *et al.,* 2002b).

Obtained results after using waste cooking oil as fuel for diesel engines showed that the resulted fuel has high viscosity and low calorific values that will have a major impact on the spray formation and gives a high Cetane number. The last two parameters affect the initial of the combustion, which decreases the ignition delay of UCO biodiesel. Moreover, it is found that the peak pressure of UCO biodiesel and its blends is higher than the one of petrol-diesel fuel. Increased Oxygen content in the biodiesel fuel improves the combustion compared to petrol-diesel fuel. Furthermore, CO and unburned HC emissions were significantly decrease with biodiesel due to a more complete combustion caused by higher oxygen content.

2. Materials and methods

In the present investigation, a naturally aspirated DI diesel engine developing 4.5 KW at 1500 rpm was used. Table 1 presents the test engine specifications used in the current study. The engine is mounted on a fixed table and coupled with an eddy current dynamometer, that converts the mechanical energy generated by the engine to an output network directly. In order to manage the control and the acquisition of measured signals, two systems were installed. The first system controlling the engine dynamometer and the acquisition of low-frequency measurements (torque, engine speed, pressure and temperature in the collectors). While the second system measuring high-frequency signals, which present the cylinder pressure, fuel injection pressure and the angular position of the crankshaft. The pressure in the cylinder was measured at a frequency of 90 KHz using a piezoelectric pressure sensor with type: AVL QH32D. The injection pressure was measured by a piezoelectric pressure transducer with another type named: AVL QH33D located in between the injection pump and the fuel injector. Where an encoder wit type AVL 364C placed on the flywheel, was used to measure the angular position of the crankshaft.

Table 1. Specification of test engine

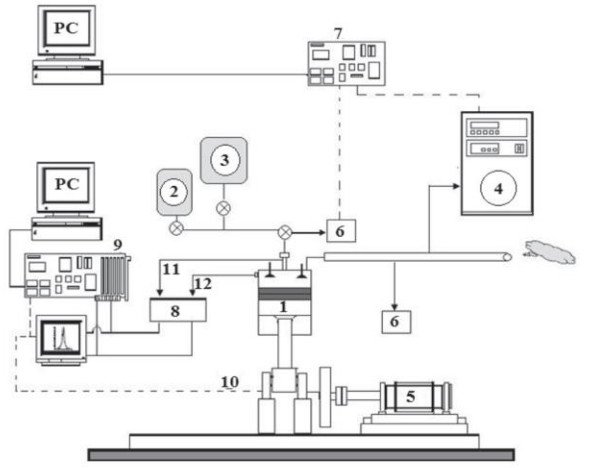
Lister Petter – TS 1 series	
General informations	4 Strokes, natural aspiration
Number of cylinders	1
Cooling system	Air cooled
Stroke	88.94 mm
Bore	95.5 mm
Connecting rod length	165.3 mm
Volumetric ratio	18:1
Maximum power	4.5 kW at 1500 RPM
Swept volume	630 cc
Injection timing	13° CA before TDC
Injection pressure	250 bar

Table 2. Properties of waste cooking oil biodiesel (WCO) and petrol-diesel

Properties	Biodiesel	Diesel
Kinematic viscosity ($mm^2. s^{-1}$ at 343 K)	4.0	2.3
Density (kg/m3)	0.86	0.80
Lower heating value ($kJ. kg^{-1}$)	38.152	43.450
Chemical composition	$C_{17}H_{31}O_2$	$C_{21}H_{44}$
Flash point (K)	433	333
Stoichiometric air/fuel ratio	12.25	15.0

In the present experiment, the intake air-flow was measured by a differential pressure transmitter with a type: LPX 5481. For the temperature measurements, the test engine was equipped with a series of thermocouples type K. Ambient temperature was measured by an active transmitter for humidity and temperature, type HD 2012 TC/150. The fuel flow was calculated with the use of a Coriolis mass flow meter. Figure 1 shows a full schema of the test engine setup.

The biodiesel was prepared from WCO by an alkali-catalyzed process using caustic soda (NaOH) as a catalyst and methanol as an alcohol. Properties of waste cooking oil biodiesel and petrol-diesel used in this work are listed in Table 2.



(1) Test Engine	(7) Low Frequency Data Acquisition System
(2) Biodiesel Tank	(8) Charge Amplifier
(3) Petrol-Diesel Tank	(9) High Frequency Data Acquisition System
(4) Exhaust Gas Analyzer	(10) Crank Angle Encoder/ Speed Sensor
(5) Eddy Current Dynamometer	(11) Injection Pressure Signal
(6) Particulate Matter Analyzer	(12) Cylinder Pressure Signal

Figure 1. Schematic diagram of the test engine setup

In this work, the Savitzky-Golay (SG) method is used to filter the cylinder pressure signal given by the test bench. SG smoothing filter method (based on local least-squares polynomial approximation) is typically used to smooth out noisy signals that have peaks, and it is much better performed than the other standard averaging filters since they minimize the least squares error when fitting a polynomial to each noisy data frame. From another hand, Savitzky-Golay filters are more effective in preserving the relevant high-frequency components of the signal. Input parameters, which gives high performance are: 3 for the polynomial degree and 21 for the frame size.

The determination procedure of heat release rate is based on the processing of the in-cylinder pressure (indicator) diagram presented in Figure 2. The properties of the gas content are considered constant inside each time step (Heywood, 1988; Colin *et al.*, 2016). By applying the first law of thermodynamics on the closed part of the cycle, we can write:

$$m.C_v.dT = -P.dV - \sum dQ_{wall} + dQ_{comb} \tag{1}$$

Considering the working medium as a perfect gas the corresponding gross heat release rate, which is the energy released from the combustion of fuel, is given by equation (2).

$$\frac{dQ_{comb}}{d\theta} = \frac{C_v}{r} V.\frac{dP}{d\theta} + (\frac{C_v}{r} + 1) P.\frac{dV}{d\theta} + \frac{dQ_{wall}}{d\theta} \tag{2}$$

Where P is the cylinder pressure, r is the gas mixture constant and C_v is the specific heat of gas mixture at constant volume.

The cylinder volume V was calculated at any crank angle position, by equation (3).

$$V(\theta) = V_C + \frac{V_S}{2}.(1 + \lambda - \cos\theta - \sqrt{\lambda^2 - \sin^2\theta}) \tag{3}$$

Where V_S is the swept volume, V_C is the clearance volume and λ is the ratio of radius of the crank over the connecting rod length.

The specific heat of gas mixture at constant volume is calculated by equation (4).

$$\begin{cases} C_v = \dfrac{1}{m} \sum_i m_i C_v^i \\ C_v^i = a_0 + a_1 T + a_2 T^2 + a_3 T^3 + a_4 T^4 \end{cases} \tag{4}$$

Where T and m are the temperature of the combustion chamber and the mass of the gas mixture respectively. a_i is the coefficients of the polynomial correlation (Awad *et al.*, 2014). The subscript i numbering each chemical component of the gas mixture.

The heat losses through the walls of cylinder are calculated from equation (5).

$$\begin{cases} \sum \dfrac{dQ_{wall}}{d\theta} = \sum \dfrac{dQ_{wall}}{dt} \dfrac{dt}{d\theta} \\ \sum \dfrac{dQ_{wall}}{dt} = h_c.A.(T - T_{wall}) \\ \dfrac{dt}{d\theta} = \dfrac{1}{6N} \end{cases} \tag{5}$$

Where A is the surface area of the cylinder walls, T is the in-cylinder temperature, T_{wall} is the mean cylinder wall temperature, N is the engine speed. h_c is the convectional heat transfer coefficient.

Using Woschni's correlation h_c is calculated by:

$$\begin{cases} h_c = 3.26 P^{0.80} U^{0.80} B^{-020} T^{-0.55} \\ U = C_1 S_p + C_2 T_r \dfrac{V}{V_r} \dfrac{(P - P_m)}{P_r} \end{cases} \tag{6}$$

Where U and B are the average cylinder gas velocity and bore diameter of the cylinder. S_p is the mean piston speed. C_1 and C_2 are constants that depend on the stroke of cycle. P_r, T_r and V_r are the gas mixture pressure, temperature and volume at some reference state (say inlet valve closing or start of combustion), and P_m is the motored cylinder pressure at the same crank angle as P.

3. Results and discussions

The experimental data covers four ranges 25%, 50% 75% and 100% of full load, at engine speed of 1500 rpm, during the whole cycle for both cases of engine fuelling with WCO biodiesel and petrol-diesel. Cylinder pressure and fuel injection pressure were recorded with an acquisition step of 0.2 °CA. Each cycle presented in this study represents the mean value of the mean of 100 consecutive recorded cycles.

3.1. Pressure diagram analysis

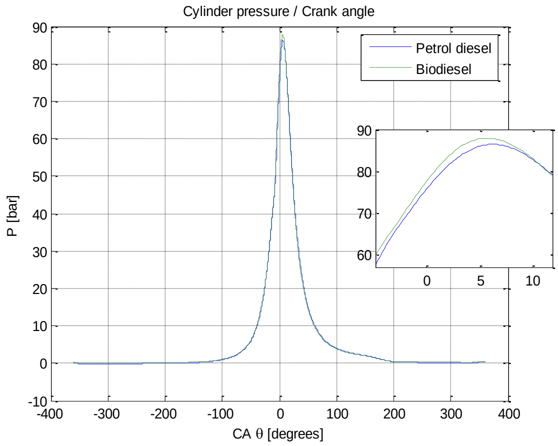


Figure 2. Experimental pressure indicator diagrams at 75% of engine load for Biodiesel and petrol-diesel

Figure 2 shows the full experimental pressure traces for the biodiesel and the petrol-diesel, at 75% of engine load when the engine recorded its best performances. As can be observed the pressure lines for both cases are almost identical, however it is noteworthy that a slight increase in peak of pressure for biodiesel fuel can be perceived.

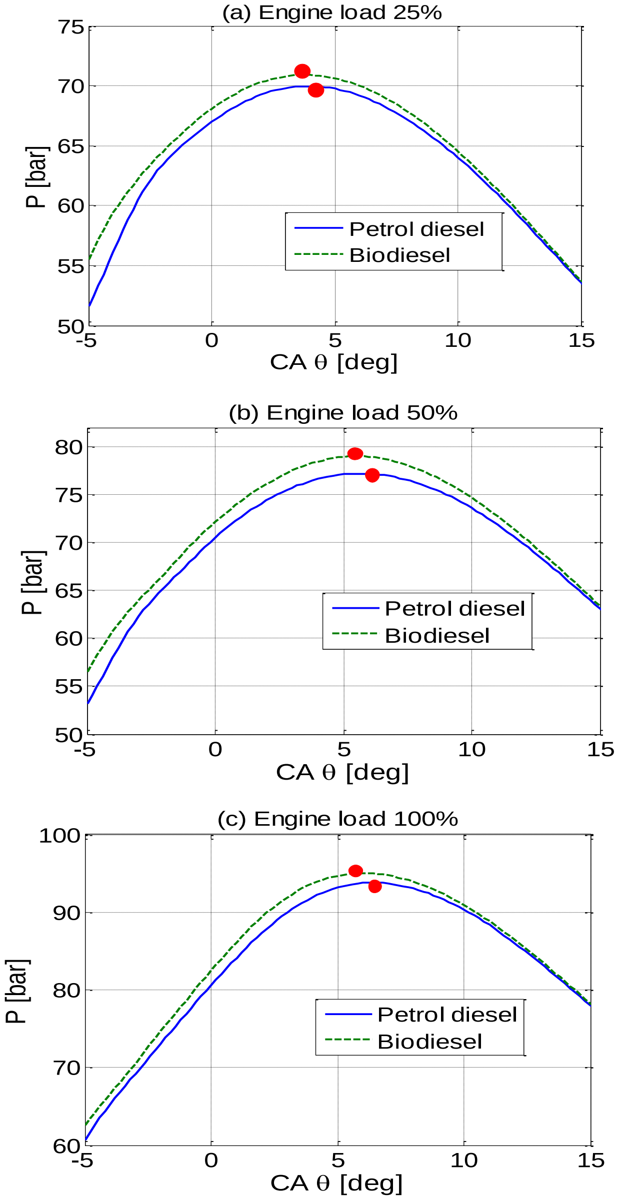


Figure 3. Peaks of cylinder pressure

At other loads of engine, the pressure traces for both fuels are typically similar except at pressure peaks where they are always slightly higher for biodiesel than diesel fuel as shown in Figure 3.

For biodiesel, the low volatility leads to poor atomization and mixing with air that implies a lateness on the start of combustion, however this is counteracted by high Cetane number, which reduces considerably ignition delay. However, a slight advance of peaks pressure toward TDC occurs with biodiesel at all loads. Very early peaking can produce engine knock problem, which affect the durability of the engine. Fortunately, for the WCO biodiesel the extreme case when the pressure peaks approaches the TDC is at low load of engine, in this case the peak remain at 4 CA° far from TDC.

The above parameters lead to drive position of peak cylinder pressure towards TDC and increase cylinder pressure for biodiesel than petrol-diesel.

3.2. Heat release analysis

The heat release pattern is mainly dependent on fuel properties such as Cetane number, LHV and the ability of the fuel to mix well with air. The premixed phase is generated by the combustion of mixture prepared during the ignition delay period. Peak of released energy depends on the rate of combustion in initial stage, which in turn influenced by the amount of fuel burned in premixed phase.

The determination of the HRR curve is very important to analyse the combustion operation, in which we can extract (the ignition delay, the combustion during, etc.). To analyse the various phases of the combustion operation easily, the pressure signal smoothing-filter is necessary, in which the calculation of the HRR depends. Figure 4 and Figure 5 show the heat release rates for Biodiesel and Petrol-diesel without and with filtering respectively. By applying the smoothing filter method of SG it can be seen that the analysis operation becomes clear and easy.

Figure 5 represents the corresponding heat release rate at 1500 rpm and 75% of engine full load. A quantity of heat is necessary for the vaporization of the injected fuel, therefore we can perceive that the HRR slope down during the delay period. The biodiesel, due to its higher Cetane number, gives a shorter Ignition delays compared to petrol-diesel. After the ignition delay, it can be se en that the biodiesel starts combustion earlier unlike petrol-diesel, which takes more time to ignite, this leads to more injected fuel before start of combustion, that leads to a larger amount of fuel accumulation in the combustion chamber at the time of the premixed combustion stage. This leading to explain the premixed combustion heat release is higher for diesel owing to greater volatility and better mixing of diesel with air in this stage. The premixed fuel-air burns rapidly followed by diffusion combustion, where the burn rate is controlled by fuel-injection. After the peak point, the heat release rate decreases faster for petrol-diesel than for biodiesel, this is due to the existence of oxygen in the biodiesel molecules, which improves its diffusive combustion phase. The same observation still holds true for the combustion characteristics at all the other engine loads.

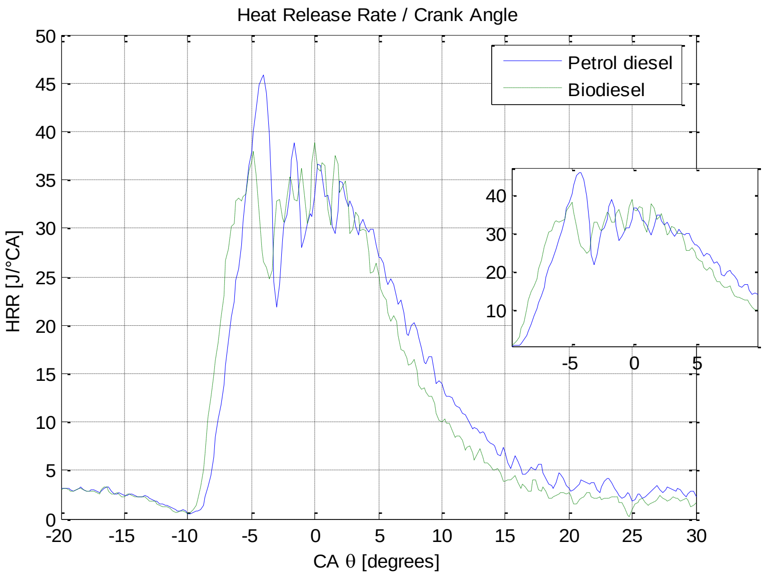


Figure 4. Heat release rates calculated with smoothing for Biodiesel and Petrol-diesel

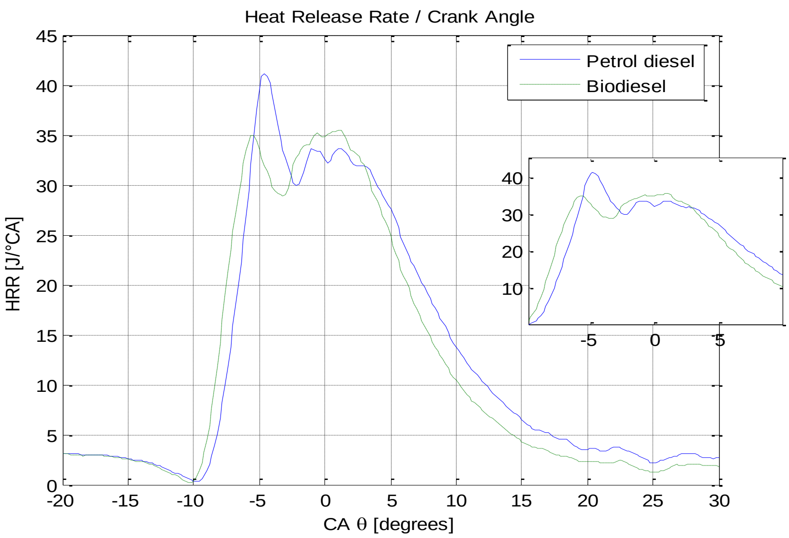


Figure 5. Heat release rates calculated without smoothing for Biodiesel and Petrol-diesel

Figure 6 illustrates the evolution of burned fuel fraction depending on the crank angle for 1500 rpm and at 75% engine load. Burned fuel fraction is obtained by the ratio HRR/ LHV. It is observed that the beginning of the burned fuel fraction curve of the biodiesel is faster compared to petrol-diesel. Knowing that, the injected mass of biodiesel is greater than that of petrol-diesel, (0.02405387 g/cycle for biodiesel and 0.02205094 g/cycle for petrol-diesel). The amount of biodiesel burned fuel mass is important; this is due to that the low heating value of petrol-diesel (43.450) is higher than that of biodiesel (38.152), which generates a higher consumption of biodiesel to maintain the same effective power as petrol-diesel. As shown in Figure 6, the burned mass can reach 92.89% for the injected amount of biodiesel and 89.5% for the injected amount of petrol-diesel.

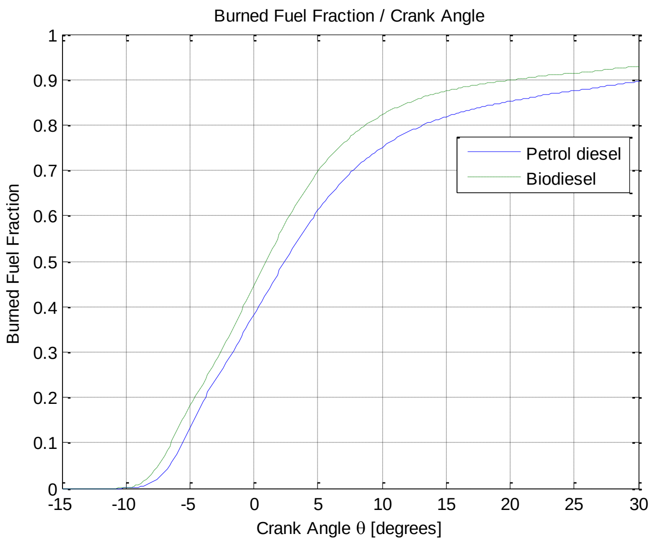


Figure 6. Burned fuel fraction for Biodiesel and Petrol-diesel

3.3. Engine performances

Brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) were given as engine performance in Figures 7 and 8 for biodiesel and petrol-diesel fuel at different loads. As shown the BSFC of biodiesel is higher than that of petrol-diesel fuel for all conditions owing to the lower LHV and higher viscosity of biodiesel. Therefore, the amount of fuel introduced to the cylinder for a desired energy input has to be greater with the biodiesel.

The output power imposed by the engine brake, grant a nearly identical shape of pressure traces at all the engine loads. Biodiesel fuel presents comparable performance output to petrol-diesel, with a slight increase in fuel consumption. It is to be noted that the brake thermal efficiency is simply the inverse of the product of the specific fuel consumption and the lower calorific value of the fuel; this leads to increase of brake specific fuel consumption for the biodiesel in order to compensate the corresponding lower heating value. Otherwise, the engine marked the higher brake thermal efficiency at the vicinity of three-quarters of full load with biodiesel fuel. Moreover, mechanical and indicated thermal efficiency keep identical trends either using WCO biodiesel or petrol-diesel for all loads.

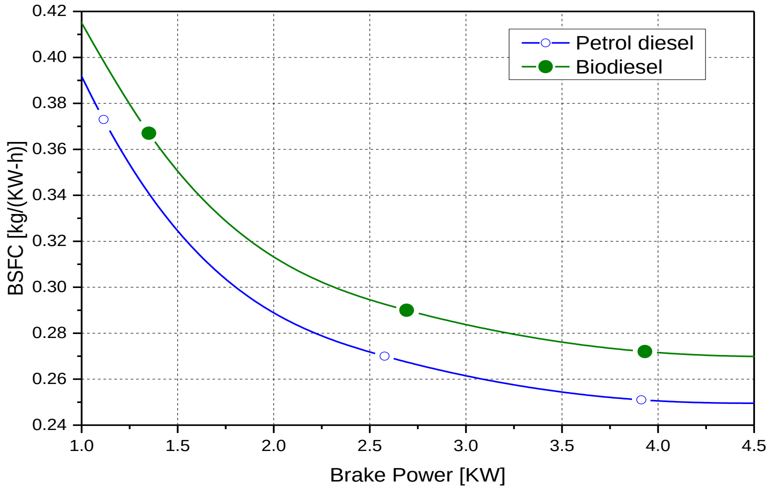


Figure 7. Brake specific fuel consumption

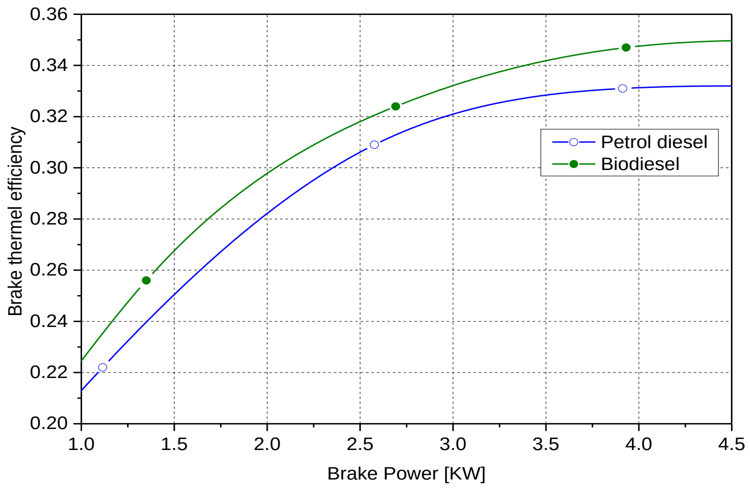


Figure 8. Brake thermal efficiency

3.4. Engine emissions

Of the major exhaust pollutants, both unburned hydrocarbons and nitrogen oxides are ozone or smog forming precursors. The use of biodiesel results in substantial reduction of unburned hydrocarbons. Emissions of nitric oxides are slightly reduced or slightly increased depending on the engine load for biodiesel compared to petrol-diesel. Even the presence of oxygen in biodiesel, which form a suitable environment to NOx formation, the beginning of the combustion process is delayed (from the injection of biodiesel or petrol-diesel to the point of auto-ignition). This delay results in low NOx formation since the products of combustion are exposed to high temperature for a short period. So, the reasons for no significant NOx emissions increase with biodiesel compare to petrol-diesel fuel (see Figure 9).

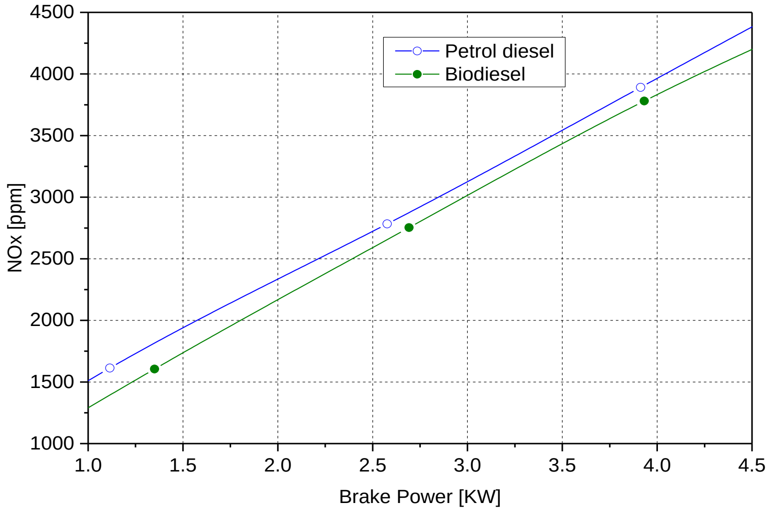


Figure 9. NOx emissions

Figure 10 demonstrates the emitted total unburned hydrocarbons, where one can observe that the HC emitted by the petrol-diesel are higher than the corresponding ones of biodiesel. As known, the formation of unburned hydrocarbons originates from various sources in the engine cylinder. Almost the total HC emitted is already mixed beyond the lean flammability limit during the ignition delay period and thus will not be able to auto-ignite or sustain a fast reaction front. Experiment shows that exhaust emissions of unburned hydrocarbons are on average 67 % lower for biodiesel than petrol-diesel, fact attributed to short ignition delay for biodiesel.

CO emissions with brake power at engine speed of 1500 rpm are presented in Figure 11. CO increases as load increases and generated by quantity of fuel injected. It's well apparent that CO for biodiesel fuel is lower than diesel fuel due to the existence of O_2 in their composition, which increases the rate of different elementary reactions. As the load increases as the effect of oxygen content in biodiesel molecules can be distinguished. We recorded a reduction of 25% of CO with biodiesel compared to petrol-diesel.

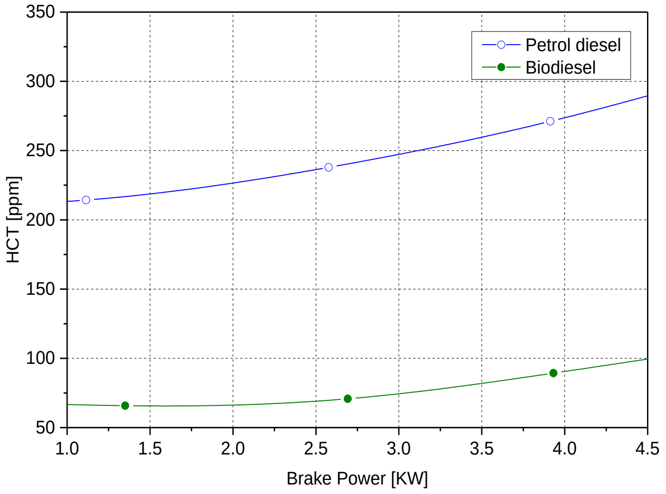


Figure 10. Unburned hydrocarbons

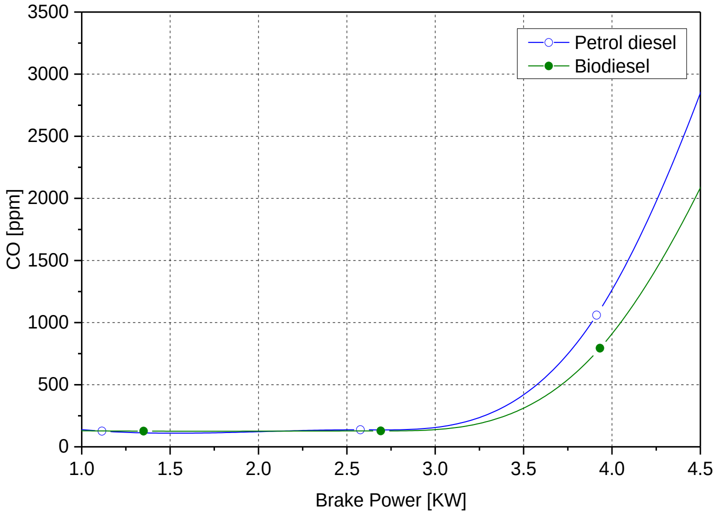


Figure 11. CO emissions

Figure 12 illustrates the particulate matter emissions for both fuels at various brake power. One can observe that the soot emitted by the WCO biodiesel fuel is significantly lower than that for the corresponding petrol-diesel fuel case, with the reduction being higher at high load. Experiments show a reduction of 56% by biodiesel compared to diesel fuel at three quarters of engine load.

At low loads, the soot emissions are nearly similar for both fuels, this may be attributed to the engine running effectively overall 'leaner'. Since the aspirated air mass remains the same contrariwise at high loads when the combustion was being assisted by the presence of the fuel-bound oxygen of the biodiesel even in locally rich zones, which seems to have a dominant influence. This reduction is attributed also to lower stoichiometric air-fuel ratio and reduced aromatic content reducing carbonaceous soot (Su *et al.*, 2013).

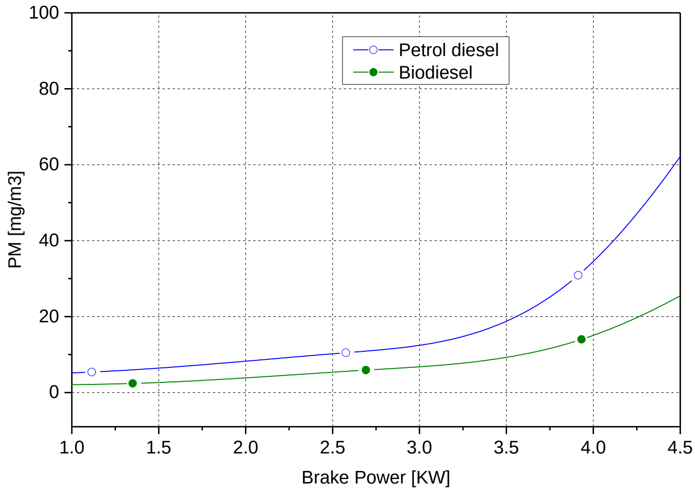


Figure 12. Particulates emissions

4. Conclusions

In this study, an experimental investigation of combustion and heat release of a DI-CI engine fueled with WCO biodiesel and diesel fuel was carried out. Based on the experimental study, the pressure signal given by the test bench requires pretreatment 'filtering' to be used. Several standard averaging filters were used in the previous studies. In this work a smoothing filters method (Savitzky-Golay) is used. The main results of combustion and heat release rate analysis are summarized as follows:

Analysis of the combustion indicated that biodiesel exhibited when the pressure peak cylinder of the engine running with biodiesel is slightly higher than the engine running with petrol-diesel. The main reason for a the high in-cylinder pressure in the CI engine running with biodiesel could be due to the advanced combustion process being initiated by the high Cetane number of biodiesel. In addition to other relevant physical properties such as viscosity, and density.

When the engine running with petrol-diesel or biodiesel fuels, results show the same combustion stages at all load conditions, except for a slight variation in peak heat release rate and ignition delay. By the use of biodiesel, the ignition is seen to be advanced compared to petrol-diesel fuel.

From the obtained experimental results, the emission exhaust such as CO, HC and PM were lower for biodiesel fuel in comparison with the petrol-diesel.

The use of biodiesel has been seen to increase the brake specific fuel consumption with less than 10 % of the CI engine due to its; low heating value, high density and viscosity.

The results exhibit that the WCO biodiesel presents a comparable performance to those of petrol-diesel fuel. Consequently it can be concluded that biodiesel fuel can be used in the CI engine without any modification, in addition to that the undesirable combustion features such as; unacceptable high cylinder gas pressure rises are not observed.

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Nomenclature

Abbreviations

ATDC	After Top Dead Centre
BSFC	Break Specific Fuel Consumption

BTDC	Before Top Dead Centre
CA	Crank Angle
DI	Direct Injection
HC	Hydrocarbons
HHR	Heat Release Rate
LHV	Lower Heating Value
NOx	Oxides of Nitrogen
PM	Particulate Matter
Rpm	Revolution Per Minute
SG	Savitzky-Golay
TDC	Top Dead Centre
UCO	Used Cooking Oil
WCO	Waste Cooking Oil
Variables	
A	surface area of the cylinder walls, m^2
a_i	coefficients of the polynomial correlation
Cv	specific heat at constant volume, J. g^{-1}. K^{-1}
h_c	convection heat transfer coefficient, W. m^{-2}. K^{-1}
m	fuel-air mass in-cylinder, g
N	engine speed, Rpm
P	in-cylinder Pressure, Pa
R	gas mixture constant, J. mol^{-1}. K^{-1}
S_p	mean piston speed, m. s^{-1}
T	temperature, K
U	mean gas velocity in the cylinder, m. s^{-1}
V	volume of the combustion chamber, m^3

Greek symbols

λ	ratio of radius of the crank over the connecting rod length
Q	heat release rate, J. °CA^{-1}
θ	crank angle position, deg

Subscripts

comb	combustion
m	motored cycle
r	reference conditions
wall	walls of cylinder