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ABSTRACT

The current study investigates the opportunity of using olive mill wastewater (OMWW) as a renewable fuel in internal combustion engines. The biofuel was made from the hydrothermal conversion of OMWW using sub-/supercritical alcohol-water system. OMWW biofuel was blended with pure diesel (designated B0). Three blends namely B10, B20 and B30 containing respectively 10%, 20% and 30% by volume of biofuel were tested. This paper focuses on the experimental study of the performance, combustion and emission characteristics of a diesel engine using OMWW biofuel as a partial replacement of conventional diesel fuel. All experiments were performed with constant engine speed (1500 rpm). The engine load was varied from 20 to 100% of full load. It is found that B10 blend offers the best results, compared to B20 and B30, in terms of performance and pollutant emissions. Using B10 showed lower emission levels for unburned hydrocarbons (12%), particulate matter (12%) and carbon monoxide (26%) under high load conditions, compared to that of B20. Moreover, under medium load conditions, B10 further reduced the concentration of pollutants such as particulate matter (55%) and carbon monoxide (65%) compared to that of B30. In addition, the use of B20 and B30 blends resulted in a lower thermal efficiency than that of diesel fuel. Conversely, the thermal efficiency obtained with B10 was the highest even compared to diesel fuel.

Keywords: Biofuel-diesel blend; Emission; alcohol; olive mill wastewater; diesel engine.

Nomenclature

Q	heat, [J]
γ	Ratio of specific heats, [-]
P	Cylinder pressure, [bar]
P_{out}	Brake power output, [kW]
V	Cylinder volume, [m^3]
BTE	Brake thermal efficiency, [%]
BSFC	Brake specific fuel consumption [g/kWh]
LHV	Lower calorific value, [MJ/kg]
$\dot{m}$	Mass flow rate, [kg/s]
ID	Ignition delay, [deg CA]
θ	Crank angle, [deg CA]
L	Connecting rod length, [m]
V_d	Displacement volume, [m]
C	Stroke, [m]
CA	Crank angle
Cr	Compression ratio
PDP	Physical delay period
CDP	Chemical delay period
ϕ	Equivalence ratio
TDC	Top dead center
CI	Compression ignition
HC	Hydrocarbon
NOx	Nitrogen oxides
CO_2	Carbon dioxide
CO	Carbon monoxide
OMWW	Olive mill waste water
HTL	Hydrothermal liquefaction
HRR	Heat release rate

1. Introduction

Nowadays, the world is facing a fossil fuel depletion crisis and environmental degradation due to the pollution from transportation sector. The reduction in underground carbon resources is due to excessive consumption and the indiscriminate extraction of fossil fuels [1]. Adverse effects on human health such as respiratory, development of cancer, cardiovascular diseases, are caused by exhaust gas emissions of internal combustion (IC) engine [2]. In addition, fossil fuels face other challenges, such as the limited capacity of refineries, depletion of resources and security of supply. The search for alternative renewable fuels has become highly pronounced in the current context, promising a harmonious relation with environmental preservation, management, efficiency, energy conservation, and sustainable development. Therefore, it is important to accelerate the development and the use of new alternative energy sources by developing energy efficiency and economical processes for the production of fuels and chemicals [3]. The alternative fuel must be technically feasible, economically competitive and easily available. [4].

Recently, biofuels produced from renewable resources such as plants or biomass-derived organic waste, have become more attractive due to their various benefits, which are related to economy, environment and energy security. Among the major advantages of biofuels: (i) the availability from various biomass sources, (ii) they have a considerable environmentally friendly potential, and allow reduction in greenhouse gases, (iii) they are biodegradable and contribute to the sustainability [5]. The production of biofuels is evolving from a range of sources. First generation biofuels are produced from foods and agriculture resources [6]. Second generation biofuels, on the other hand, are produced from non-food biomass sources [7]. The structured fraction of biomass feedstock is typically composed from hemicelluloses, cellulose and lignin. In addition, the early focus among second generation liquid biofuel (e.g. bio-oil, biobutanol, bioethanol and biodiesel) researchers and producers was often on the lignocellulosic inedible biomass. Therefore, the use of second generation feedstock, which includes inedible and waste, is considered as potential alternative fuel source.

Many conversion processes are currently available and well established for the production of transport biofuels from different types of biomass, including esterification, fermentation, digestion,

hydrothermal liquefaction (HTL), gasification and thermal cracking (pyrolysis) [8]. Among thermochemical conversion processes, recently HTL has attracted much attention due to the direct use of biomass without the need for a preliminary drying step [9]. HTL process aims to produce low molecular weight liquids from a high organic molecular weight feedstock under subcritical or supercritical conditions. The processing of biomass is carried out at moderate temperature and pressure typically between 250-350°C and 10-20MPa, respectively, in the absence of oxygen [10]. The liquefaction products are generally biofuel fraction, aqueous phase, solid residues and a CO_2-rich gas fraction. The biofuel produced has a high calorific value in the range of 30-43 MJ/kg [11–13], and lower oxygen content compared with biofuel from pyrolysis process. Therefore, the HTL biofuel has suitable properties to be used as engine fuel [14].

On the other hand, olive oil consumption is rapidly increasing worldwide, due to its high dietetic and nutritional value. The three major olive oil producers worldwide are Italy, Spain and Greece, followed by Turkey, Tunisia, and to a lesser extent Portugal, Morocco, and Algeria [15]. The traditional press extraction method as well as the continuous three-phase decanter process, which is most widely used for the production of olive oil, generate three products: olive oil, olive husk and aqueous waste called olive mill wastewater (OMWW). A serious ecological problem represented by the treatment of OMWW due to its high degree of organic pollution and its pH slightly acid [16,17]. Regarding the environmental impact, it is considered to be a significant polluting waste in all Mediterranean regions [18]. Indeed, in terms of pollution effect, 1 m^3 of OMWW is equivalent to 100-200 m^3 of domestic sewage [15]. Its uncontrolled disposal in water reservoirs leads to severe problems for the whole ecosystem and especially for the natural water bodies (ground water reservoirs, surface aquatic reservoirs, seashores, and sea). It is a turbid liquid, black to dark brown in colour and smells of oil [16,19]. In addition, the dark colour of this waste depends on the age, olive type processed as well as the technology used to extract oil. Its characteristics are very variable and depend on many factors such as the variety and maturity of the olives, the employed extraction technology, the climatic conditions, the cultivation management and the storage time [16,19]. OMWW can be considered as a serious option to produce biofuel since it is inedible and a non-food feedstock.

Few research studies dealing with the strategy adopted to recover these waste through the thermochemical conversion are available. Pyrolysis of OMWW on dried samples has been thoroughly investigated at a heating rate of 5°C/min in order to study the gas formation during this process [20]. Nevertheless, using pyrolysis process is not suitable for this kind of waste due to its high moisture content (more than 80%) [21]. Miranda et al. [22] investigated the combustion of OMWW by studying the thermal behaviour of different wastes coming from olive oil mills by using the thermogravimetric techniques in oxidizing atmosphere. Moreover, Ekin et al [23] examined the gasification of the liquid waste from industrial olive oil in supercritical water. The maximum amount of gas produced, which was 7.7 ml per ml of OMWW, was observed at reaction temperature of 550°C, with a reaction time of 30s. On the other hand, the hydrothermal carbonization of OMWW was studied by Poerschmann et al [24]. They reported that the bio-char yield of 30% (w/w) was associated to the low carbohydrate fraction in the OMWW but no information regarding the bio-oil was published. According to literature, conversion of OMWW into bio-oil, via hydrothermal liquefaction, was considered recently in two studies using sub- and supercritical in water and alcohols [14,21]. They found a bio-oil viscosity of 10.2 and 7.6 mPa.s using ethanol- water and methanol-water co-solvent, respectively. The higher heating value of 43.20 MJ/kg was obtained employing MeOH-water co-solvent which resulted in better physical and chemical properties.

Regarding the use of biofuel in compression ignition (CI) engines, the literature shows a variety of aspects in order to enhance the combustion parameters and engine performance [25–28]. Most of these researches focused on the use of biodiesels and diesel-biodiesel blends. Indeed, biofuels can be blended with neat diesel in any proportion to obtain a biofuel blend, leading to lower hazardous exhaust emissions, resulting in unchanged or improved engine efficiency [29,30]. Table 1 presents some literature studies about the impact of different blends on engine performance from various biofuel sources. Recently algae biodiesel blended in two different proportions (10% and 15%) with diesel, was investigated by Indrareddy et al. [31] on a common rail direct injection diesel engine. The study reported that for both blends, carbon monoxide (CO) exhibits the maximum reduction compared to diesel. The authors concluded that B15 blend can be used as an alternative fuel in the

diesel engine. Moreover, high injection pressures, i.e. 1050 bar, show a considerable improvement in performance and emission characteristics. Miraculas, G. A et al. [32], have studied the performance of biofuel from *Calophyllum inophyllum* oil-based methyl ester and its blends (0, 20, 40, 60 and 100%) with diesel on internal combustion engines. The results were analysed statistically using an experimental design. It was reported that the designed empirical statistical model for optimum performance with lower emission is found to be B30, resulting in a lower break specific fuel consumption at higher loads compared to that of diesel and led to lower emissions. On the other hand, Chauhan, B. S et al. [33] made a review of the effect on performance and emissions of biofuel blends from different renewable sources. The review reports that, based on combustion and performance characteristics, biodiesel-diesel blends containing about 10–20% showed better results than higher blends. However, the use of these blends in diesel engine causes an increase in NOx emission and a decrease in HC, CO and PM emissions compared to diesel.

Table 1. Literature review of biofuels from different sources and the impact of different blends on engine performance.

Feedstock source	Process of biofuel production	Engine loads (biofuel-diesel blends)	Remarks	Ref
Calophyllum inophyllum oil	Esterification	0, 20, 40, 60 and 100%	-Lower emissions were recorded for B30 at a compression ratio (CR) of 19. - The NOx emission increases with CR for higher biofuel blends but reduces the CO and HC emission than that of original CR at maximum load.	[32]
Crude Karanja Oil	Transesterification	10, 20, 30, 40, 50, 80 and 100%	- Brake thermal efficiencies (BTE) of blends have been found to be on increasing trend up to 70% as the load increases. B50 and B100 showed a low brake thermal efficiency.	[27]
Need seeds	Pyrolysis	5% and 10%	- A decrease in BTE till low load and increases in full load along with the reduction in HC, CO, CO_2, and NO_x at full load conditions were noticed.	[34]
Australian pinus radiata wood flour	Hydrothermal liquefaction	5%, 10% and 20%	- With 20% biodiesel, the properties are almost equal to that of diesel but a decrease of engine performance and increase of NOx emissions are noticed. -A maximum of 33% reduction in particulate matter (PM) emissions was observed with 20% blend	[34]
Pistacia lentiscus seed	Liquefaction	30 and 50%	- 50% of blend results in a considerable low NOx emissions level as compared with neat diesel fuel	[35]

The literature survey reveals that the usage of biodiesels, obtained from vegetable oils and animal fats, in CI engines are widely examined. In fact, biodiesels must meet the international standards and therefore their characteristics are not that different. However, few works investigated the use of alternative fuels, obtained from wastes via HTL, in engine as the biofuel characteristics could be different from that of conventional diesel. This constitute our motivation to make biofuel from HTL of OMWW. The focus of this research is to analyse and evaluate the effects of diesel and biofuel blend on diesel engine performance and emission characteristics under various loads. For that, the physicochemical properties of biofuel and biofuel blends are determined according to the standard methods and compared with those of conventional diesel fuel. Thereafter, an experimental investigation is carried out to study the effect of this biofuel blended with diesel fuel (10, 20 and 30% by v.) on the performance and the exhaust emissions (carbon monoxide CO, unburned hydrocarbon HC, nitrogen oxide NOx and soot) of a diesel engine operating under various engine load conditions. Results are compared with those of conventional diesel operation, taken as a baseline. To the best of our knowledge, there is no study dealing with the use of blended biofuel from HTL of OMWW in a conventional diesel engine.

2. Materials and methods

2.1.Bio-fuel production

The OMWW feedstock used to produce biofuel was collected from a traditional oil mill in Beni Amrane, a city situated in the north of Algeria. The characteristics of the feedstock (i.e. ash content, water content, volatile matter, fixed carbon, fat and the C, H, N contents) were determined using the protocols reported in the previous work [36]. The biofuel was produced using a HTL batch reactor at high temperature (200-320°C) and pressure (5-17 MPa) in the oxygen absence. Nitrogen was used to sweep air from the reactor inside before heating start. Ethanol co-solvent with a ratio of 50/50% was used as alcohol to upgrade the quality of biofuel as reported in our previous work [14]. The schematic of biofuel production from OMWW liquefaction is illustrated in Figure 1.

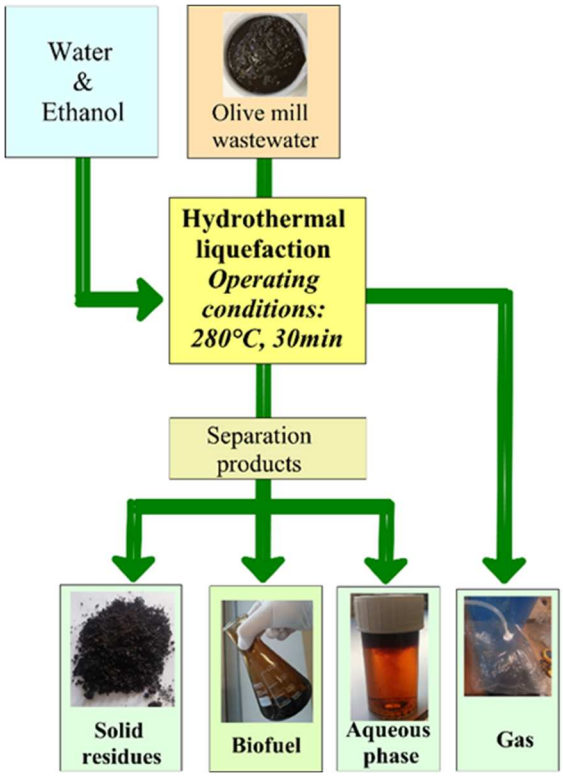


Fig.1. Schematic presentation of biofuel production.

2.2.Biofuel properties

The main physical and chemical properties of biofuel blends and diesel were determined according to ASTM standard and are depicted in Table 2. The biofuel obtained from the liquefaction of OMWW is composed of a large quantity of esters, about 79.31 % which were attributed to the esterification reactions between fatty acids and ethanol [14]. In fact, the concept of blending low viscous pure diesel with the high viscous OMWW biofuel reduces the cost of fuel substantially. The lower cetane value of OMWW biofuel could be balanced by higher cetane value of pure diesel to achieve better combustion.

The biofuel has higher viscosity, higher density, and higher flash point compared to the biofuel blends. The lower calorific value is due to lower hydrogen content in biofuel [37], where the value is about 11.49wt.% [14]. The cetane number of biofuel is slightly lower compared to that of diesel fuel and biofuel blends. However, higher cetane number implies a shorter ignition delay while lower cetane number may produce knock in the engine. Similarly, viscosity and flash point are important properties affecting the volatility, flow of fuel, and spray characteristics. Higher viscosity and flash point leads to lower volatility and consequently poor combustion. Hence it is decided to blend the OMWW biofuel

with pure diesel at different proportions to balance most of the properties and to bring it closer to pure diesel. Experiments were conducted on CI engine using diesel fuel, OMWW biofuel blends with diesel at 10, 20 and 30 percent by volume.

Table 2. Properties of biofuel, diesel fuel and their blends.

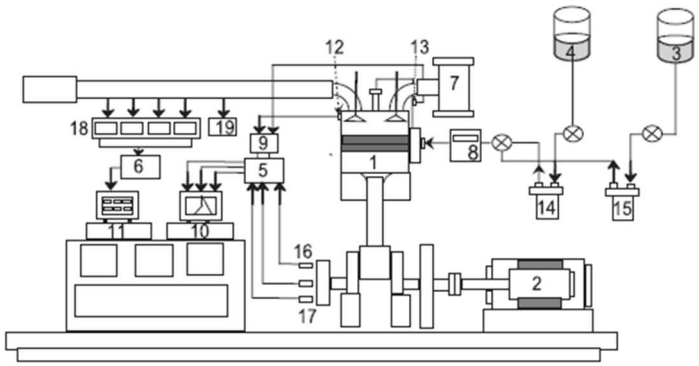
Characteristics	Unit	Diesel	Biofuel	Blends			Test Method
				B30	B20	B10	
Density at 15°C	kg/m^3	852	899.07	846.75	838.75	832.79	ASTM D1298
Lower calorific value	MJ/kg	43.20	39.21	41.87	42.46	42.91	ASTM D240
Viscosity at 40°C	mPa.s	1.71	10.20	2.58	2.22	1.90	ASTM D445
Flash point	°C	67	117	73	74	77	ASTM D93
Cetane number	-	57.56	54.33	56.63	56.12	57.88	ASTM D613

2.3. Experimental setup

The results presented in the present study were carried out in the laboratory of IMT Atlantique. The experimental test bench is based on a single-cylinder direct injection, naturally aspirated, compression ignition engine (manufacturer: LISTER-PETTER). The power output is 4.5 kW at a rotational speed of 1500 rpm. It is equipped with several instruments to perform the engine tests, to analyse the combustion and the measurements of exhaust emissions. The orifice meter connected to a large tank is attached to the engine manifold to measure the air flow. The fuel flow rate is measured with a Coriolis mass flow meter. Chromel-alumel thermocouple in conjunction with a low frequency data acquisition system is used for measuring the exhaust gas temperature. An exhaust analyser (COSMA) is used for measuring hydrocarbons (HC) by a flame ionization detector, while carbon monoxide (CO) emissions are measured using an infrared device. Nitrogen oxides (NOx) in the exhaust is measured by using a BECKMAN chemiluminescence NO/NOx analyser. Particulate emissions are measured using a Pegasor Particle Sensor (PPS-M). PPS-M is based on a novel measurement technique enabling real-time, continuous and high sensitivity measurements of raw exhausts PM emissions without diluting the exhaust gas. A high frequency data acquisition system in connection with two AVL piezoelectric transducers is used for measuring the cylinder pressure and fuel line pressure histories. An optical shaft position encoder is used to give signals at TDC. The data of cylinder pressure was recorded every 100 consecutives cycles with a sampling interval of about 0.2 crank angle (CA). Before each experiment, the engine is calibrated according to the manufacturer

catalogue values. All data are collected after the engine has stabilised. During the entire investigations, the working parameters of the test engine are fixed as injection timing of 13◦CA before TDC for, engine speed of 1500 rpm, and compression ratio of 18. More details about the acquisition system; sensors precision and uncertainty calculation can be found in our previous works [38,39]. Figure 3 shows the engine experimental setup. A schematic diagram of engine setup is depicted in Figure 4. Table 3 shows the engine testing procedure and the engine specifications.

Fig.3. Experimental setup.



1. Test engine
2. Dynamometer
3. Biofuel tank
4. Diesel fuel tank
5. A/D card for pressure
6. A/D card for Analyser
7. Air tank
8. Fuel flow meter
9. Charge Amplifier
10. Fast data acquisition system
11. Slow data acquisition system
13. Injection pressure sensor
14. Diesel fuel filter
15. Biofuel filter
16. TDC encoder
17. Speed sensor
18. Exhaust gas analyser
19. Particulate meter

Fig.4. Schematic diagram of engine setup.

Table 3. Lister-Petter engine specifications.

General details	Single cylinder, naturally aspirated, 4-Stroke
Cooling system	Air-cooled
Injection system	Compression Ignition, direct injection
Bore×stroke	95.3 mm×88.9 mm
Connecting rod length	165.3 mm
Compression ratio	18:1
displacement volume	630 cm^3
Fuel injection timing	13° BTDC
Fuel injection pressure	240 bar
Rated power output	4.5 kW at1500 rpm
Orifices×diameter	4×0.25 mm
Piston type	Cylindrical bowl (diameter : 45mm and depth : 15mm)
IVO	36° CA before TDC
IVC	69° CA after BDC
EVO	76° CA before BDC
EVC	32° CA after TDC

Before starting the serial tests, the engine test bench was calibrated according to the instructions given in the manufacturer catalogue. The pressure cylinder, brake torque, flow rates of air, diesel and blend fuels are registered to determine the engine performance. The injection timing of the pilot fuel is set at 13° CA before TDC for all experiments. Experiments are carried out at a constant speed engine (1500 rpm) and different loads (25%, 50%, 75% and 100% full load).

2.4.Combustion analysis model

Processing pressure data in the form of smoothing is essential, depending on the noisy trend of the pressure signal between successive values. For our case, the smoothing was established using the smoothing equation of the instantaneous pressure data used by several researchers, as reported in the literature [40]. The combustion process was examined by the determination of the heat release rate (HRR). The HRR was calculated analytically by applying the first law of thermodynamics and the ideal gas equation. As shown in the equation below, to obtain the HRR, the variation of the cylinder pressure and volume is used.

$$\frac{dQ_{net}}{d\theta} = \frac{dQ_c}{d\theta} - \frac{dQ_W}{d\theta} = \frac{\gamma}{\gamma-1} P[\frac{dV}{d\theta}] + \frac{1}{\gamma-1} V[\frac{dP}{d\theta}] \quad (1)$$

Where $\frac{dQ_{net}}{d\theta}$ is the net HRR, $\frac{dQ_C}{d\theta}$ is the rate of heat released by the fuel combustion and $\frac{dQ_W}{d\theta}$ is the heat transfer rate through the cylinder wall obtained from the Woschni's correlation [39]. *P* is the cylinder pressure and γ is the ratio of specific heats. From the literature, the γ is generally fixed as 1.35. *V* represents the combustion chamber volume which depends on the crank angle (θ) and the geometric parameters of the engine. The cylinder volume *V* i*s* obtained as follow:

$$V(\theta) = V_d \left[\frac{Cr}{Cr-1} - \frac{1-\cos\theta}{2} + \frac{1}{2}\sqrt{\left(2\frac{L}{C}\right)^2 - \sin^2\theta} \right] \quad (2)$$

Where V_d, C_r, L and C are respectively the displacement volume, the compression ratio, the connecting rod length and the stroke.

3. Results and discussion

3.1. Combustion characteristics

In this part, to diagnose the combustion process at different loads, the results of the cylinder pressure data, HRR, ignition delay and combustion duration are examined. All the results were compared with those of the conventional diesel fuel.

3.1.1. Heat release rate

From the literature, the heat release rate is divided in three steps, ignition delay, premixed combustion and diffusion combustion. Fig. 5 shows the evolution of the heat release rate (HRR) versus the crank angle (CA) for the tested fuels. The higher peak of HRR for premixed phase is obtained for B0 at medium loads and for B20 and B30 at full loads. This is due to the longer ignition delay of B30 and B20 in comparison with that of diesel and B10 at high loads, as shown in Fig. 5b. Indeed, we can see that the combustion starts earlier for the diesel fuel and for the B10 due to higher density of diesel and higher bulk modulus as well as shorter ignition delay [41]. In fact, as reported in the literature, a longer ignition delay leads to the accumulation of fuel which burns at higher rate during premixed combustion phase leading to higher HRR peak values [39]. However, because of this prolongation of the ID, the premixed phase ends later (in the expansion phase) as mentioned previously, at medium loads.

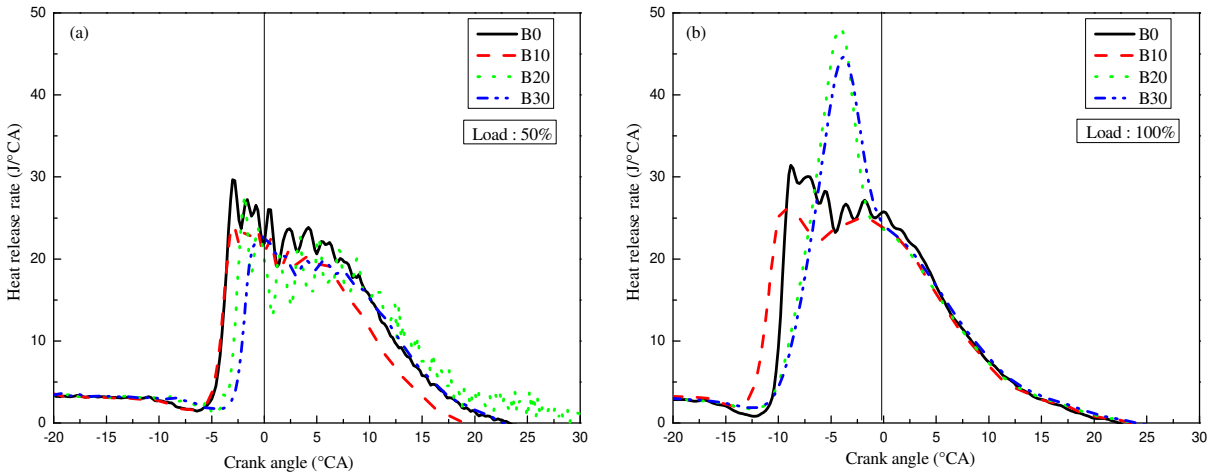


Fig.5. Heat release rate (HRR) with crank angle for diesel and diesel-biofuel blend fuels.

Moreover, it can be seen from Figure 5 that HRR of diesel is higher than that observed for B10 for all loads studied. According to this Figure 5, the HRR of B10 tends to become similar to that of conventional mode at medium loads, during the ID and the premixed phases. In addition, a separation can be seen in the third phase of HRR. The HRR of diesel remains higher than that of the B10 at premixed and diffusion phases, at the same loads. Furthermore, at full load, the curves are detached at the first and second phases of the HRR and tend to overlap in the diffusion phase. This confirms the results related to ID presented in the next section, where the ID was shorter for the B10 compared to the diesel at full load as shown in Figure 5.

3.1.2. Cylinder pressure

Figure 6 presents the evolution of the cylinder pressure at medium and full loads versus crank angle for diesel and biofuel-diesel blend fuels. As shown in Fig 6a, the highest peak of cylinder pressure is observed for B0 (diesel fuel) than that of biofuels tested at medimum loads. However, we can explain these behaviors by the difference in energy content of mixed fuels, as mentioned in Table 2. The position of the cylinder pressure peak indicates the speed to release this energy. It is important to mention that the heat release rate depends on the concentration of oxygen contained in fuel (which accelerates combustion), viscosity (which affects atomization and vaporization of fuel) and latent heat (which affects directly the ignition delay and combustion cooling) [42].

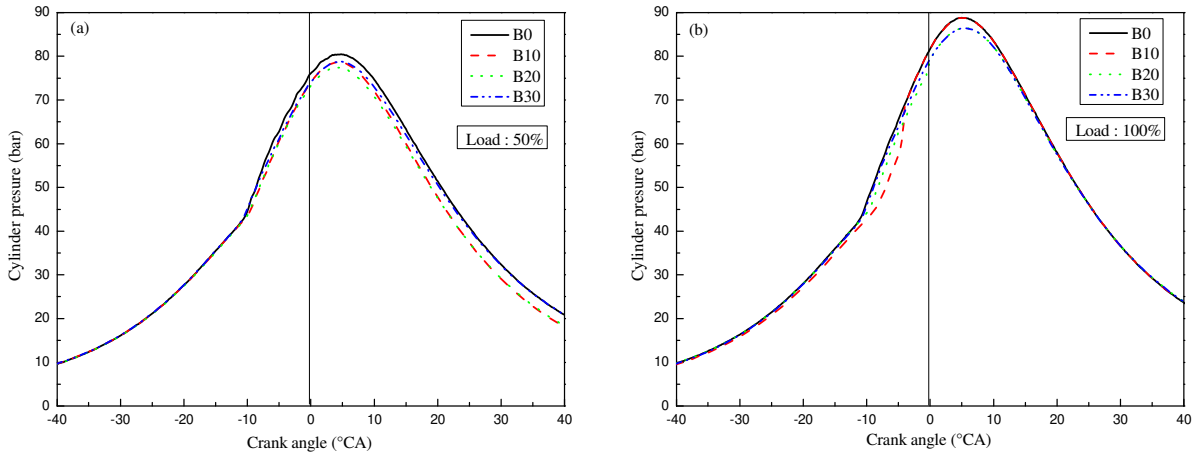


Fig. 6. Cylinder pressure variation with crank angle for different biodiesel.

The combination of these factors may explain the behavior of the cylinder pressure mentioned in Figure 3. In addition, the increase in the concentration of biofuel in the mixture results in a longer ignition delay [42]. The combination of the low self-ignition temperature and the relatively high cetane number reduce the ignition delay [42]. This can be visualized in Table 2 where we find these two parameters for the different mixes studied . Moreover, it can be also seen from this Fig. 6b a separation between the curves of cylinder pressure of all the cases studied. This is the consequence of a lower heat release rate for B10 and diesel in the premixed combustion (before TDC), as mentioned previously, due to the specific heat capacity of B10 and diesel which is higher than that of B20 and B30. Furthermore, in the expansion, the curves have the same appearance for all the cases studied and overlap perfectly at full load.

3.1.3. Ignition delay

The ignition delay (ID) is by definition the interval between the start of injection and the beginning of combustion. This parameter is important to analyse correctly the process of combustion and to give additional explanations of the behaviour of both the in-cylinder pressure and the rate of heat release. From the literature, the ID is composed of two periods. The first one is the physical delay period (PDP) and the second one is chemical delay period (CDP). The PDP period is related to the air mixing with the fuel atomization, vaporization and decomposition, as reported by Heywood. Regarding the CDP, it represents the time required for reaction combustion start.

Figure 7 shows the variation of the ignition delay with power output for the fuels tested. The ID decreases with increasing the load for all the fuels studied. This can be explained by the elevation of the cylinder temperature and pressure as well as the equivalence ratio of the mixture. In addition, it can be observed that, for all the cases studied, the ignition delay of B20 and B30 is the highest. This result can be explained by the increase in the temperature of the gases in the cylinder which promotes self-ignition which reduces the ID of B10 and diesel fuel than that of B20 and B30. Moreover, one can observe that at medium and high loads, the ID for B10 tends to be shorter than that of diesel, unlike at low loads.

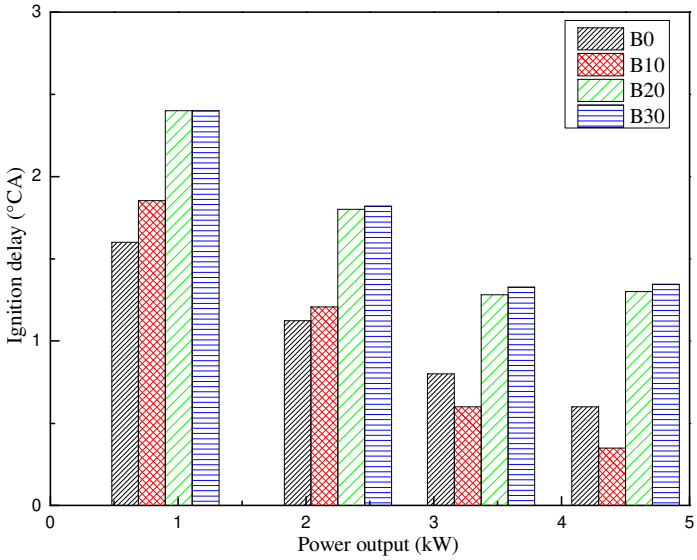


Fig.7. Ignition delay variation with power output.

The long ignition delay for B20 and B30 enhances the fuel-air mixture and therefore results in an improvement of the premixed combustion phase as discussed previously as illustrated in Figure 5. Increasing the concentration of the biofuel in the fuel mixture with diesel accelerates and improves the vaporization of the fuel studied. The presence of a significant amount of biofuel also induces a higher in cylinder temperature and pressure as shown in Figure 5 and 6.

3.2. Engine performance

To analyze the performance of engine, the brake thermal efficiency (BTE) is examined. It is calculated as follow:

$$BTE = \frac{P_{out}}{\dot{m}_f \ LHV_f} \quad (3)$$

Where P_{out} (kW) is the brake power output. $\dot{m}_f$ is the mass flow rate of fuel (kg/s).

It can be seen from Fig. 8a that BTE curves have similar trends for all the fuels over the load range tested. The thermal efficiency increases as the load increases. This is due to the increase in heat produced from the combustion inside the cylinder. Moreover, the increase of the equivalence ratio of the mixture in the cylinder leads to improvement of combustion efficiency. All these factors are beneficial for increasing BTE. In addition, as shown in Fig. 8a, the BTE increases with a uniform gap between the curves related to fuels tested.

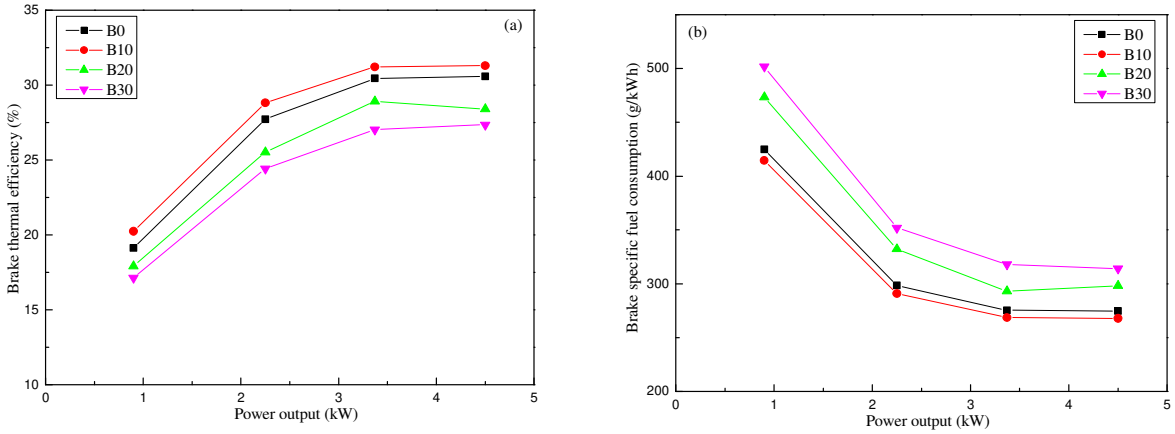


Fig.8. Evolution of the BTE (a) and BSFC (b) versus loads.

However, according to the Fig. 8a, it is noticed that despite the low calorific value of B10, compared to that of diesel and the two other blends tested, the performance of B10 remains slightly higher or similar to that of diesel. On the other hand, the BTE of B10 is the closest to that of Diesel in comparison to that of B20 and B30. In fact, if biofuel is added in small amount (10%), the biofuel's high oxygen content induces a good combustion efficiency which results in an improvement in the brake thermal efficiency (B10). Moreover, this behaviour can be explained by the high cetane number and flash point as well as a low viscosity than those of B20 and B30 as shown in Table 2. This reflects the combustion process deterioration of B20 and B30 tested, leading to a reduction of the combustion efficiency resulting from the low atomization of the fuel tested and of the high viscosity, as reported by Nour et al. [42]. Consequently, the B30 presents a lower BTE with a value of 28% at full load. However, for diesel and B20, the BTE is about 31% and 29%, respectively at the same load. The same

observation was reported by Mccarthy et al [43]. Their study was focused on the analysis and comparison of performance and emissions of an internal combustion engine fuelled with petroleum diesel and different biodiesels. They found that the performance of biodiesel fuels reduces with increasing blend ratio, and the fuel consumption increase was in the range of 7–10%.

The Fig. 8b shows the variation of BSFC with respect to load for diesel fuel and diesel-biofuel blends. Under all load conditions, the BSFC of B20 and B30 is higher of about 14% and 17%, respectively, compared to that of diesel and B10. This can be explained by the higher viscosity and lower energy content of B20 and B30 in comparison with that of diesel fuel and B10. This led to injecting higher quantity of fuel to meet out the same power output as that of pure diesel and B10 operation which agree with the same observation reported in literature [43,44].

3.3. Exhaust emissions

3.3.1. HC and CO emissions

The emissions of unburned hydrocarbons (HC) are by definition the result of incomplete fuel combustion. Fig. 9a shows the variation of the HC emissions versus loads. It can be noticed that the curves have the same trend. At low loads, the concentration of HC emissions tends to be the same. At high loads, the B10 emits less HC than the other cases studied, with a maximum observed for the case of B20. These observations are attributed to the prolonged ID and the delayed combustion into the expansion stroke end. All these parameters will induce a less time of high temperature to give complete conversion, consequently, a high concentration of HC emissions as reported by Nour et al. [42].

The variation of the concentration of carbon monoxide (CO) emissions recorded in the exhaust gases for the different fuels tested is shown in Fig.9b. It can be seen that the curves describe the same trend regardless of the fuel studied. Furthermore, the concentration of CO emissions increases mainly at high load due to high amount of fuel injected and less available oxygen for complete combustion of the fuel. The same observation was reported in literature [42,45].

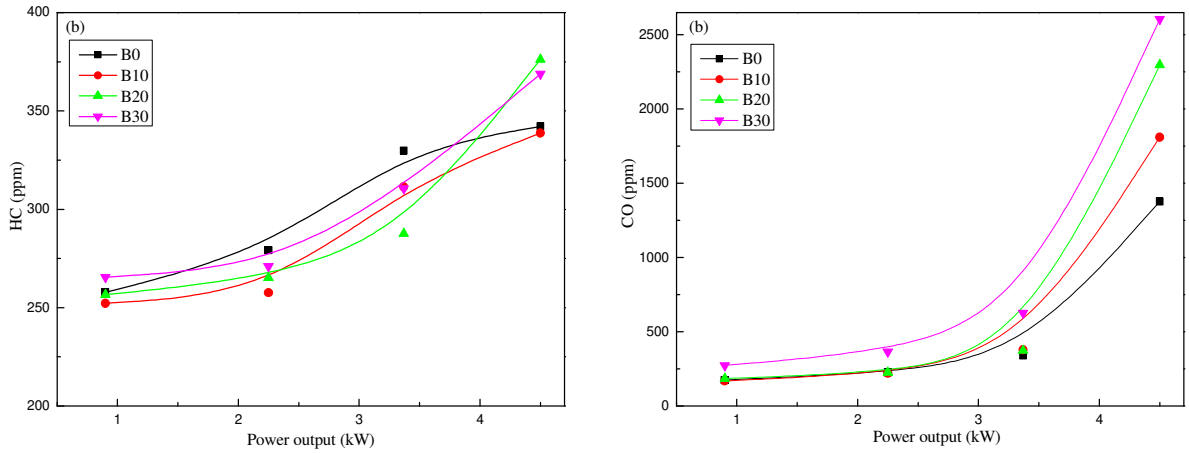


Fig.9. Variation of HC emissions (a) and CO emissions (b) versus loads.

At low and medium loads, the concentration of CO emissions remains mostly the same for diesel, B10 and B20. In addition, at high load, the diesel was the less pollutant fuel in terms of CO emissions. Regarding the case of B30 blend, it had a maximum of CO emissions regardless the load studied. This can be explained by the formation of large droplet size, worsening fuel atomization and the highest viscosity of fuel.

3.3.2. NOx and particulate emissions

For many years, it has been known that the major problem of the conventional diesel engine is the high concentration of particulate and NOx emissions [38]. In this part, the concentration of this emission type was analysed. The results obtained from the combustion of the different fuels (diesel and diesel-biofuel blends) are depicted in Figure 10.

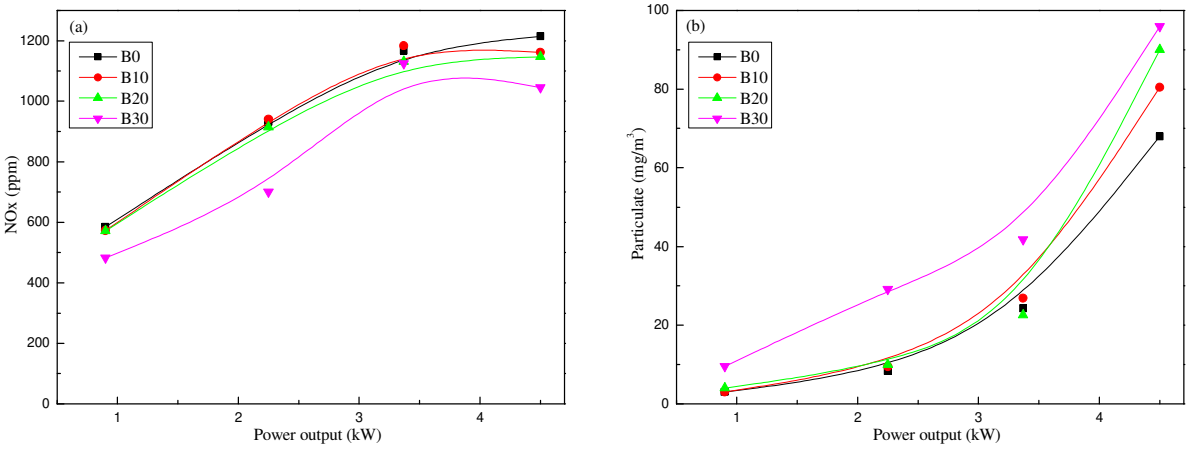


Fig.10. Variation of NOx emissions (a) and particulates emissions (b) versus loads.

The analysis of NOx emissions in exhaust gases at different loads is shown in Fig. 10a. It can be perceived that the curves describe the same trend regardless the fuel studied. Moreover, the concentration of NOx emissions remains similar to that of diesel fuel for all loads except for the B30 fuel. This confirms the results previously presented in Fig. 10b, where CO emissions are inversely proportional to NOx emissions. This behaviour was also reported by Heywood [46]. This is the consequence of the decrease of the average gas temperature in the cylinder.

According to the literature, the early formation of soot particles at the diffusion combustion process level is due to the dissociation of the fuel at high temperature [46]. One of the main factors in the formation of these emissions is the biofuel tested composition. In fact, the bond C-C strongly favouring the production of these emissions [46].

From Fig. 10b, it was found that the concentrations of soot emissions were similar at low and medium loads except for B30, where it has the highest concentration of soot emissions whatever the load studied. Moreover, regarding the full load, B10 and B20 remain more pollutant than diesel in terms of soot emissions. Alagu et al [34] had reported the same results. They found that at full load condition, the concentration of soot emissions of biodiesel (B10 blend) was higher (52.8%) than fossil diesel operation. This is due to the highest viscosity of fuel leading to worsening fuel atomization resulting in the formation of large droplet size. Moreover, a lower cetane number produces a longer ID which induces to short time for the formed soot to be oxidized, consequently an increase of soot emissions [42].

4. Conclusion

This study focused on the impact of OMWW biofuel blended with diesel fuel on performance, combustion and pollutant emissions of a compression ignition engine. Engine tests were performed with different proportions of biofuel.

From the present results, the blend of B10 presents a reduction of 26% of carbon monoxide, 12% of unburned hydrocarbons and 12% of particle emissions with an improvement in brake thermal efficiency of 10% compared to those of B20. At medium loads, B30 had a higher carbon monoxide concentration and particulate emissions of about 65% and 55%, respectively, with a deterioration of the brake thermal efficiency of 13% compared to that of B10. On the other hand, at high loads, B10

has less polluting effects by around 43% of carbon monoxide, 10% of unburned hydrocarbons and 20% of particles compared to that of B30. Consequently, B10 is the cleaner and more efficient fuel than B20 and B30. Furthermore, it should be noted that despite the degradation of the performance of B20 and B30, these biofuels make it possible to obtain acceptable results while being partly composed of biofuel from waste. This will therefore allow to replace fossil diesel fuel and help to reduce greenhouse gas emissions.

On the other hand, to reduce the economic impact caused by the fossil fuel crisis and avoid relying on existing biofuels, it is important to seek locally available and renewable biofuel. Wherefore, OMWW biofuel has a high potential to be used as engine fuel for reducing greenhouse gas emissions. A future work will be devoted to the production of biofuel from OMWW using methanol as solvent. Engine tests will be carried out using different mixtures and results will be compared to the present work.

5. Acknowledgments

The authors are very grateful to the technicians of the Energy Systems and Environment Department of IMT Atlantique for their advice and support.

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Graphical abstract