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Experimental examination of performance, exhaust emission and combustion behaviours of a CI engine fuelled with biodiesel/diesel fuel blends

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ABSTRACT

The main objective of this study is to evaluate the effect on diesel engine performance, exhaust emission and combustion characteristics of canola oil biodiesel. The biodiesel was blended with petroleum-based diesel fuel (DF) by 10% (B10), 20% (B20) and 50% (B50) volumetrically. Engine tests were carried out on a direct injection, single-cylinder, compression ignition engine at different engine loads and the results were compared with the reference DF. The results showed that the maximum cylinder pressure and heat release rate values of DF were higher than those of the biodiesel blends at all engine loads. The addition of biodiesel to DF increased the brake specific fuel consumption (BSFC) and decreased the brake thermal efficiency (BTE) at all engine loads. In addition, biodiesel blends caused a slight increase in nitrogen oxide (NO_x) emissions while causing a decrease in smoke opacity and carbon monoxide (CO) emissions.

Keywords: Biodiesel, canola oil, diesel engine, combustion, exhaust emissions

1. INTRODUCTION

Recently, interest in alternative fuels has increased to meet growing energy needs and to prevent air pollution caused by petroleum-based fuels. Among these, biodiesel is renewable, clean and promising fuel and has been extensively investigated for use as an alternative fuel for compression ignition engines. Biodiesel is a renewable, biodegradable, nontoxic fuel and has similar combustion behavior with petroleum-based diesel fuel. For this reason, it can be used purely or by blending with DF in a diesel engine without any modification.

Biodiesel has some advantages over mineral diesel fuels. It contains 10–12% oxygen by weight and no aromatics and almost no sulphur. Furthermore, since biodiesel is an oxygenated fuel, it is completely burned and releases less CO, carbon dioxide (CO_2), sulphur oxides, particulate matter, volatile organic compounds and unburned hydrocarbon (HC) emissions [1]. Biodiesel fuels also are stored more safely because of the high flash point temperature. However, the most important disadvantages of biodiesel fuels are the higher viscosity and density as well as higher clouding and cold filter plugging point. Also, the heating value of biodiesel is lower than that of petroleum-based diesel fuels.

The transesterification process is widely used in the biodiesel production from various vegetable oils or animal fats. Biodiesel is a fatty acid methyl ester which is obtained as a result of reaction with an alcohol in the presence of a catalyst from vegetable seed oils, waste frying oils, animal fats and all kinds of biological oils and used as a fuel. It is a fact that low-cost feedstocks such as waste cooking oils [2], non-edible oils [3], and animal fats [4] are more advantageous than high-priced feedstocks such as edible vegetable oils. However, due to the increase in the number of processes,

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their esterification is more difficult and ester specification standards may not be met [5]. For this reason, biodiesel production from high-priced edible vegetable oils such as rapeseed (low-erucic variety; canola being similar to low-erucic rapeseed) [6], palm [7], soybean [8], corn [9], canola [10], sunflower [11] etc. are still widely preferred in order to produce a high-quality biodiesel that meets the standards. In the literature, there are many studies using edible vegetable oil biodiesel in compression ignition (CI) engines.

Ozsezen et al. [12] examined the combustion behaviours and performance of a direct injection (DI) diesel engine fueled with frying palm oil methyl ester and canola oil methyl ester. They reported that the combustion behaviours slightly changed and engine performance slightly worsened compared to DF. Also, they noted that the biodiesels led to a decrease in CO, unburned HC and smoke emissions but an increase in NO_x emissions. Öztürk [10] studied the mixture of canola and hazelnut oil with DF on a DI, single cylinder diesel engine. He noted that the ignition delay (ID) and combustion duration (CD) increased and the maximum heat release rate (HRR_{max}) declined with the addition of the biodiesel. He also reported that the HC, CO, and smoke emissions decreased while NO_x emissions increased. Labeckas et al. [7] studied that the effect of rapeseed oil biodiesel on engine performance and exhaust emissions of a DI diesel engine. The results showed that BSFC of values rapeseed oil biodiesel were higher than that of DF. Çelikten et al. [13] studied engine performance and exhaust emissions of rapeseed oil and soybean oil methyl ester injected at different pressures. The results showed that the engine performance and exhaust emission characteristics of rapeseed oil and soybean oil methyl esters were almost the same with those of DF.

A number of studies were carried out on diesel engines to examine the exhaust emissions and engine performance of various edible vegetable oil biodiesel-diesel blends, as mentioned above. However, studies on combustion behavior in the literature appear to be more limited when compared to studies of engine performance and exhaust emissions. The main objective of this experimental study is to simultaneously and systematically determine and compare the effects of canola oil biodiesel-diesel blends on combustion, performance and exhaust emissions.

2. MATERIALS AND METHODS

In this study, the effects on engine performance, exhaust emission and combustion behaviours of canola oil biodiesel addition to DF. The canola oil biodiesel was blended with DF by 10% (B10), 20% (B20) and 50% (B50) volumetrically. Petroleum-based commercial diesel fuel was used to obtain reference data. The basic properties of the test fuels are seen in Table 1.

Table 1. The basic properties of the test fuels.

Properties	DF	B10	B20	B50
Density (kg/m^3)	831.5	836.7	841.9	857.5
Kinematics Viscosity (mm^2/s)	2.40	2.60	2.79	3.36
Cetan Number	58.8	58.3	57.9	56.5
Lower heating value (MJ/kg)	43.20	42.73	42.26	40.85
Hydrogen (wt%)	13.4	85.7	84.8	82.2
Carbon (wt%)	86.6	13.1	12.8	12.0
Oxygen (wt%)	0	1.2	2.4	5.8

A schematic diagram of the test setup was seen in Figure 1. The tests were performed under the same operating conditions on a naturally aspirated, air-cooled, direct-injection and single cylinder diesel engine. The main properties of the test engine was seen in Table 2. To load the engine, a Kemsan brand DC dynamometer was used that produced 15 kW of power at 3000 rpm. The tests were performed under variable engine brake mean effective pressure (BMEP) of 0.9 bar, 1.8 bar, 2.7 bar and 3.6 bar at a constant engine speed of 1500 rpm.

A Kistler brand 4550A model torque meter was used for engine torque measurement. A Kistler 2614B model encoder was used to measure crank angle, engine speed and top dead center (TDC). An A3 Kistler 6052C piezoelectric pressure sensor was used for measuring the cylinder pressure. Fuel line pressure was measured using a Kistler brand 4065B piezoresistive sensor was mounted on the fuel line. The values of the cylinder and the fuel line pressure were obtained at a crankshaft angle of 0.1 degree. A Kistler KiBox data collection system was used to record and process all experimental data. A Mobydick 5000 gas analyzer was used for measuring emission values. The main properties of the exhaust gas analyzer and opacimeter are shown in Table 3.

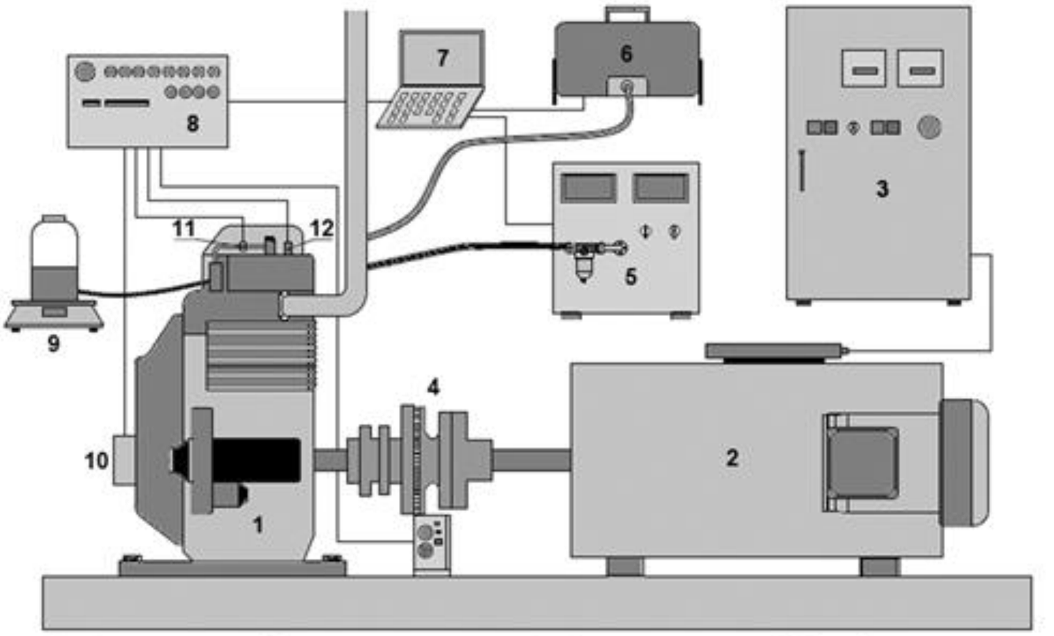

Figure 1. Schematic diagram of the test setup.

Table 2. Main characteristics of the test engine

Items	Specifications
Model	Lombardini 15LD350
Maximum torque	16.6 Nm/2400 rpm
Maximum power	5.5 kW/3600 rpm
Compression ratio	20.3/1
Displacement	349 cm^3
Bore × stroke	82 mm × 66 mm
Injection pump type	QLC type
Nozzle opening pressure	207 bar
Injection nozzle	160° x 0,22 x 4 holes

The engine was run for five minutes before each test in order to get stable data. To obtain each emission value, five measurements were carried out at the same intervals and their averages were taken. To minimize cyclical errors, all data received from the Kibox were recalculated using average values of 100 cycles. The values of maximum cylinder pressure (CP_{max}) and HRR_{max}, the start and end of combustion were calculated by using the KiBox Cockpit software.

Table 3. Basic features of the exhaust gas analyser and opacimeter

	Range	Accuracy
NO_x (ppm)	0~5000	1
Smoke Opocity (%)	0-100	±2
CO (%,v/v)	0~10	0.01

Equation (1) was used to calculate the heat release rate (HRR).

$$\frac{dQ_n}{d\theta} = \frac{k}{k-1} p \frac{dV}{d\theta} + \frac{1}{k-1} V \frac{dp}{d\theta} \quad (1)$$

Wall heat losses were not taken into account in calculating the HRR. The start of combustion (SOC) and the end of combustion (EOC) correspond to the point where the HRR is 5% and 90%, respectively. The difference between SOC and EOC is called combustion duration. The start of injection (SOI) is the crank angle at the fuel line pressure (207 bar) at which the injector starts to open. The difference between SOI and SOC is called ignition delay.

3. RESULTS AND DISCUSSION

3.1. Combustion Characteristics

The ignition delay is an important combustion parameters affecting the SOC, cylinder pressure and HRR. The variation of ID of the test fuels according to the engine load at 1500 rpm is seen in Figure 2. As the engine load increased, the gas temperature in the cylinder also increased, resulting in a shorter ignition delay. It is a fact that the high cetane number and high oxygen content of fuels cause a reduction in ID. The cetane numbers of DF and biodiesel blends are close to each other. However, oxygen content of the biodiesel is higher than that of diesel fuel. In addition, Sivalakshmi et al. [14] noted that low molecular weight gaseous compounds, separated from biodiesel during injection time at high temperature, could be ignited earlier thereby decreasing ID for biodiesel. Because of these reasons, it was found that ID was reduced slightly with the addition of biodiesel.

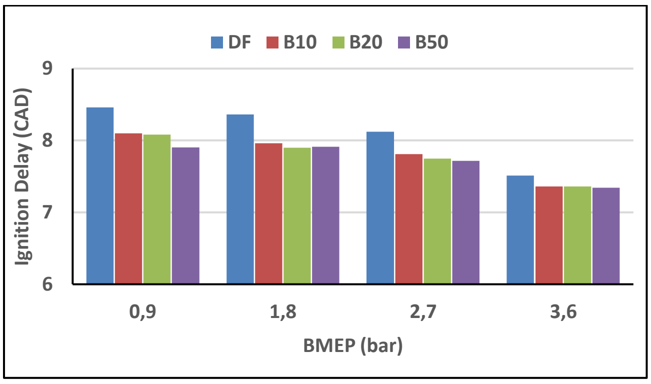

Figure 2. Change of ID according to engine loads at 1500 rpm.

The change in CD of the test fuels according to the engine load at 1500 rpm is seen in Figure 3. Since the amount of fuel injected increased with the rise in load, it was observed that the combustion duration of all test fuels increased at high loads. Biodiesel blends were found to have longer combustion duration when compared with DF for all engine loads because of the less fuel burning in premixed combustion mode and more fuel burning in diffusion combustion mode. The combustion duration generally increases in the order of DF, B10, B20, and B50.

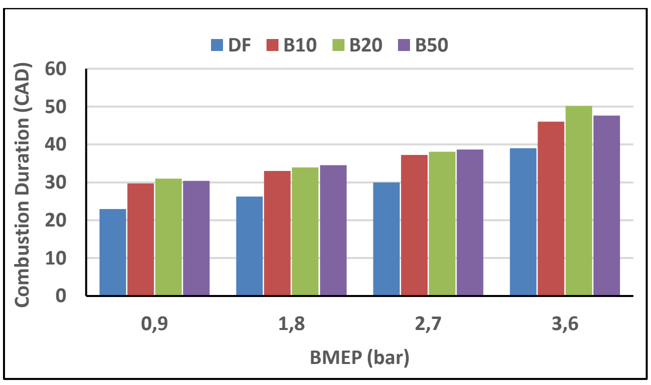

Figure 3. Change of CD according to engine loads at 1500 rpm.

The variations of cylinder pressure and HRR of the test fuels with respect to the crank angle under the different loads at 1500 rpm are seen in Figure 4. Since the amount of fuel injected into the cylinder increases with the increase of the engine load, the cylinder pressure increased at higher loads and reached their peak values with a delay for all fuels. The experimental results showed that the biodiesel blends have similar combustion characteristics to diesel fuel. Because of the almost same ignition delay at 0.9 and 3.6 bar, the peak cylinder pressure and HRR of the test fuels were observed to be almost equal at 0.9 and 3.6 bar, except for the low CP_{max} value of B50 at 3.6 bar. Because of the rapid combustion of the fuel accumulated in the combustion chamber during the longer ignition delay of DF, the CP_{max} and HRR_{max} values of DF were higher than those of the biodiesl blends at 1.8 and 2.7 bar.

The heat release rate provides quantitative information about the progress of the combustion and defines the rate of the chemical energy release of fuel during the combustion. The two main stages of combustion (premixed and diffusion) are clearly seen in the HRR graphs. Rapid burning of the fuel accumulated in cylinder during ID period in premixed combustion stage leads to a high HRR_{max}. Next, the diffusion combustion phase, in which the combustion is controlled by the fuel-air mixing speed, occurs. Because of the higher ID and the rapid burning of fuel accumulated in the cylinder, premixed combustion of DF was more intense than diffusion one at engine loads of 1.8 and 2.7 bar, which led to higher HRR_{max}.

As seen in Figure 4, HRR_{max} increased up to 2.7 bar with the increased load but decreased again at 3.6 bar for all the fuels. The reason for this reduction is the shorter premixed combustion and longer diffusion combustion phase at 3.6 bar, thus peak HRR values reduced. This result is similar to Ref [15].

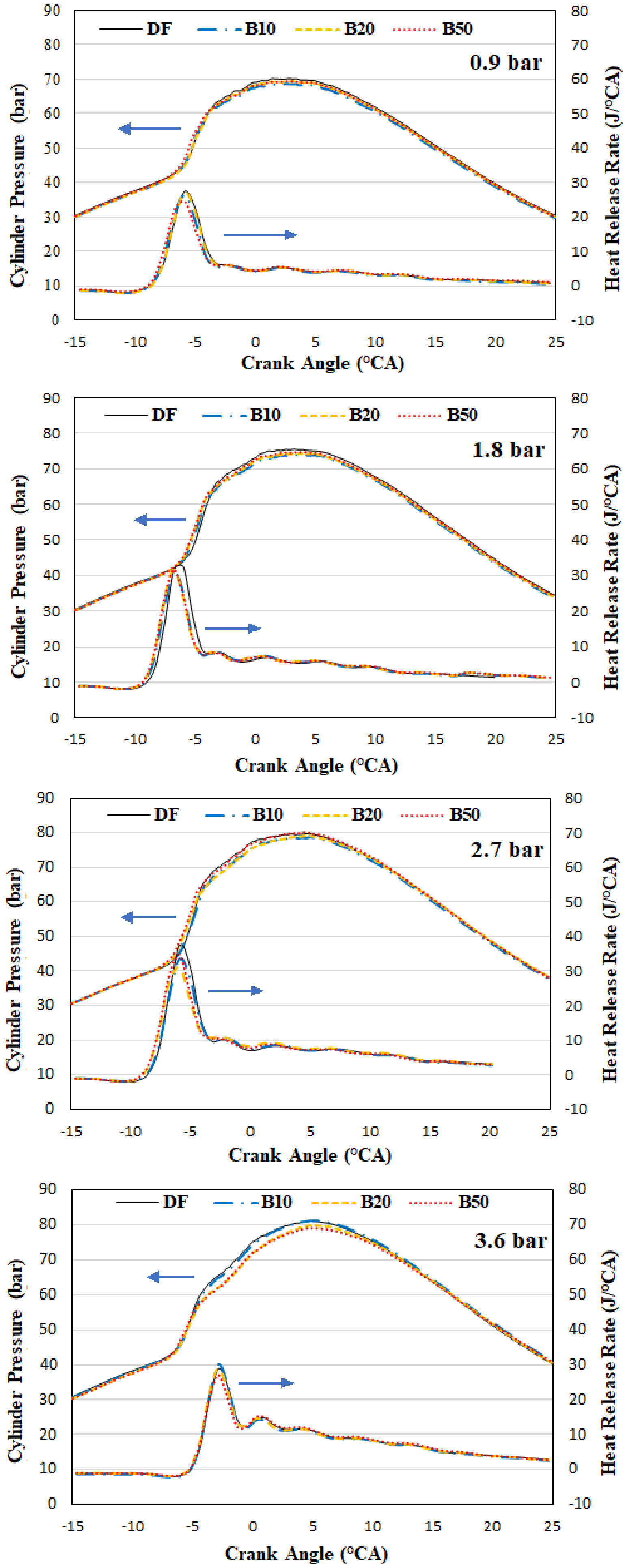

Figure 4. Change of the cylinder pressure and HRR according to crank angle under different loads at 1500 rpm.

3.2. Engine Performance

3.2.1. Brake Specific Fuel Consumption

The change in BSFC of the fuels according to the engine load at 1500 rpm is seen in Figure 5. It was found that the biodiesel blends had higher BSFC values at all loads due to the lower heating value of biodiesel. The BSFC values followed a similar trend for all test fuels and reached their minimum value at 2.7 bar, and slightly increased again at 3.6 bar. The reason for this increase is that the shorter premixed combustion and the longer diffusion combustion occur at 3.6 bar. The heating value of the fuels has a significant effect on the BSFC. Since the heating value of the biodiesel is lower, more fuel must be injected for obtaining the same BMEP. Among the biodiesel blends, B50 had the highest, B10 had the lowest BSFC.

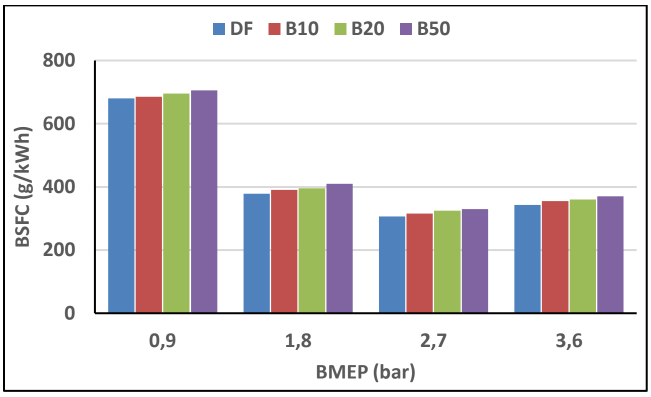

Figure 5. Variation of BSFC versus engine loads at 1500 rpm.

3.2.2. Brake Thermal Efficiency

BTE is an indication of the ability of the combustion of the fuels and is a useful parameter that gives information about how efficiently the energy of the fuel is converted into mechanical output [16]. The change in BTE of the test fuels according to engine load at 1500 rpm is seen in Figure 6. There is an inversely proportional relationship between BTE and BSFC values of a fuel. Biodiesel has a higher BSFC value because it has a lower heating value than diesel fuel. Therefore, the addition of biodiesel decreased the BTE value. DF has the highest BTE at all loads because it has the lowest BSFC value and it is followed by B10, B20, and B50 fuel blends respectively. It was also observed that BTE curves followed a similar trend and reached their peak values at 2.7 bar for all test fuels but slightly decreased at 3.6 bar. This result is similar to other study in Ref. [17].

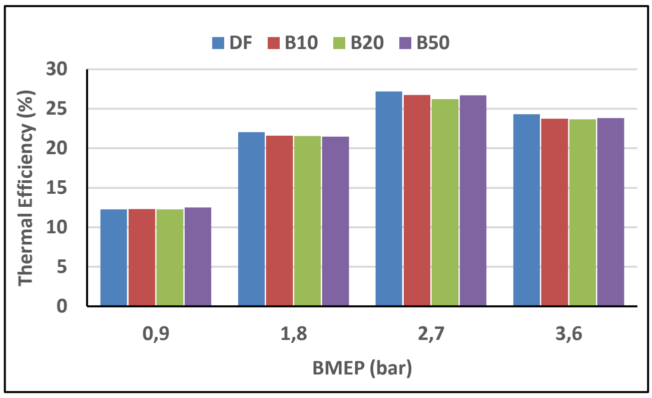

Figure 6. Change of BTE according to engine loads at 1500 rpm.

3.3. Exhaust Emissions

3.3.1. Nitrogen Oxides

The changes in NO_x emissions of the test fuels at different engine loads are shown in Figure 7. The results showed that the biodiesel addition to DF at all loads caused a slight increase due to the more oxygen content of the biodiesel, which increases the maximum temperature in the cylinder. NO_x formation is generally dependent on oxygen concentration and temperature in the cylinder. NO_x emissions of test fuels increased in parallel with the increase in the injected fuel as the engine load increased up to 2.7 bar, but it decreased again at 3.6 bar. The probable cause of this decline was that the shorter premixed combustion and the longer diffusion combustion occurred at 3.6 bar. In this case, since maximum pressures and temperatures did not increase as much as 2.7 bar, HRR_{max}, and consequently, NO_x emissions decreased. Similar findings were obtained in Refs. [17,18].

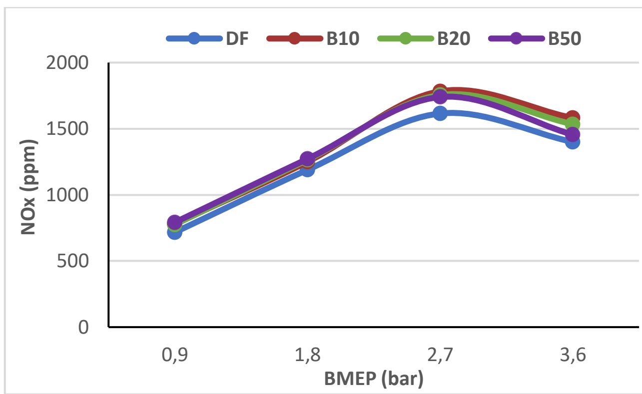

Figure 7. Change of NO_x emissions according to the engine loads at 1500 rpm.

3.3.2. Smoke Opacity

The formation of smoke depends on the engine condition, combustion chamber type and the physic-chemical properties of the fuel and occurs at the oxygen deficiency in the cylinder [19]. Oxygen deficiency is locally visible in the combustion chamber and increases with decreasing air/fuel ratio. The changes in smoke opacity of the test fuels according to the engine loads at 1500 rpm are shown in Figure 8. Biodiesel addition to DF was found to reduce smoke opacity at all engine loads because of the higher oxygen content and lower C/H of the biodiesel. The higher oxygen content of the biodiesel blends improves the combustion and reduces smoke formation. These results indicated that smoke opacity was severely reduced by the addition of the oxygenated fuel such as biodiesel in the DF.

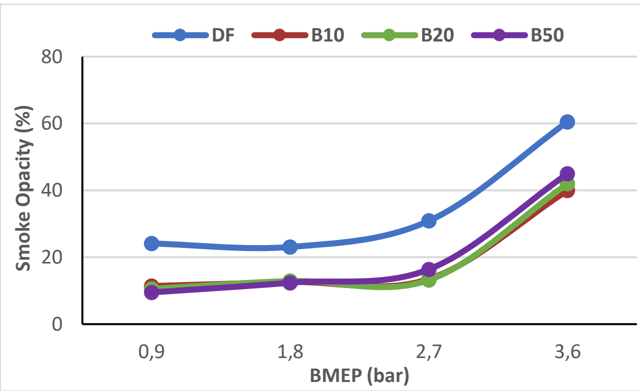

Figure 8. Change of smoke opacity according to the engine loads at 1500 rpm.

3.3.3. Carbon Monoxide

Fuel type, air/fuel ratio, combustion chamber shape, fuel atomization rate, injection pressure and timing as well as engine load and speed are the main factors affecting the CO formation. However, the air/fuel ratio is the most effective of these parameters. The changes in CO emissions of the fuels according to the engine loads at 1500 rpm are seen in Figure 9. It was observed that CO emissions were quite low for all fuels at low loads but increased at 3.6 bar because of the decrease in ratio of air/fuel with the rise in fuel injected into the cylinder. The addition of biodiesel with high oxygen content to DF was found to lead to a reduction in CO emissions at 3.6 bar. These results are similar in Ref. [17,20].

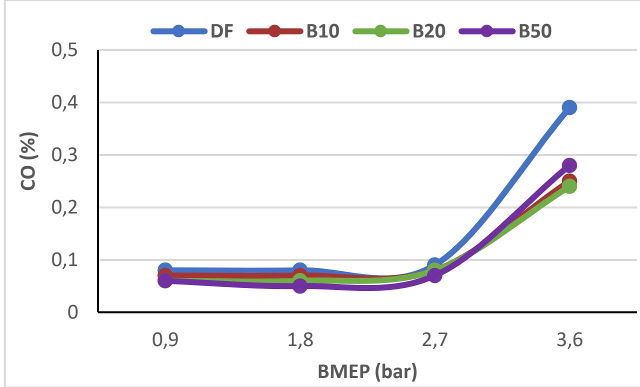

Figure 9. Change of CO emissions according to the engine loads at 1500 rpm.

4. CONCLUSIONS

In this study, the effect of canola oil biodiesel addition to DF on performance, exhaust emission and combustion characteristics of a single-cylinder diesel engine was examined under different engine loads and the following results were obtained:

1. The CP_{max} and HRR_{max} values of DF are higher than those of biodiesel blends at all engine loads due to the longer ID of DF.

2. DF has the lowest BSFC and the highest BTE compared to biodiesel blends at all engine loads.

3. The biodiesel addition to DF generally leads to a slight increase in NO_x emissions while causing a decrease in smoke and CO emissions.

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