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Experimental examination of the effects of military aviation fuel JP-8 and biodiesel fuel blends on the engine performance, exhaust emissions and combustion in a direct injection engine

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ABSTRACT

Biodiesels are the most popular fuels which can be used as an alternative fuel instead of diesel fuel in diesel engines. Low emission characteristics and high cetane numbers are the most significant advantages of biodiesels. However, JP-8 military aviation fuel which is a kerosene based fuel has a low viscosity, high lower heating value and very low freezing point. The usage of the fuel blends of biodiesel and JP-8 may be effective in improving the characteristics of biodiesel. In this study, JP-8 aviation fuel and sunflower methyl ester blends were tested at 7.5, 11.25, 15 and 18.75 Nm engine loads and at maximum torque speed (2200 min^{-1}) in a single cylinder, naturally aspirated, and direct injection diesel engine. In-cylinder pressure, ignition delay period, engine performance and exhaust emissions have been examined. As the engine load increases the specific fuel consumptions decrease for all test fuels. It was seen that NO_x emissions increased with the increase of the amount of biodiesel in the test fuels. CO emissions decreased as the amount of biodiesel fuel increased in the test fuels. Consequently, it was observed that JP-8 and biodiesel fuel mixtures can easily and efficiently be used in diesel engine.

1. Introduction

Energy demand increases each passing day in parallel with the economical and technological developments and increasing population in the world. Today, majority of energy requirement is covered by fossil fuels (petroleum) [1]. Petroleum reserves are rapidly consumed due to increasing requirements. Moreover, the usage of fossil fuels in engine vehicles increasingly threatens the eco-system of the world. So, researchers pay attention towards alternative energy sources. Today, exhaust emissions emitting from gasoline and diesel engines in the vehicles are carbon monoxide (CO), nitrogen oxides (NO_x), hydrocarbon (HC), particulate matter (PM), sulfur oxide (SO_x) and lead compounds [2,3]. In order to decrease these emissions, researchers have performed many investigations on the effects of vegetable oils as an alternative fuel in the engines. Rudolph Diesel had used peanut oil as fuel in the diesel engine [4]. Moreover, vegetable oil was used as fuel in many vehicles during World War II, but the most extensive researches on the subject coincide the 1970s when the petroleum crisis occurred [5]. Vegetable oils, which have different chemical structures compared to petroleum based fuels, cause the various problems when used directly as fuel in diesel engines [6]. The viscosity of vegetable oils is about 10 times more than the viscosity of diesel fuel. Their lower heating value is approximately 90% of the diesel fuel. Diesel fuel is substantially composed of paraffins and aromatics. But biodiesel is the ester which fatty acids made with glycerine. The type and amount of unsaturated fatty acids constituted the properties of vegetable oils. Besides biodiesel has higher oxygen content. High viscosity can lead to injection problems, poor atomization and incomplete combustion. While the vegetable oils do not have good properties on injection and ignition, they have high flash point and provide advantage on storage safety [6–8]. Besides the transesterification method was widely used in order to remove the negativities arising from vegetable oils, pyrolysis, microemulsion and refining methods are also used [9–12]. By these methods, vegetable oils are converted to biodiesel. Vegetable oils are esterificated with monohydric alcohol such as methanol and ethanol. This method is called transesterification [13]. The specifications of biodiesel shall be improved or it shall be used along with other fuels in order to expand the operating range in diesel engines [14–16].

At this point, kerosene based aviation fuel JP-8 is used along with biodiesel [17,18]. There are many studies in the literature regarding JP-8 aviation fuel [19–23]. JP-8 does not include naphtha. JP-8 has low cetane number and high boiling point as compared

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to benzen [24]. JP-8 aviation fuel is different from conventional diesel fuel due to its corrosion inhibitor content for fuel system which prevents icing at cold operating conditions. JP-8 aviation fuel can be used in motored vehicles for civil and military purposes because of such chemical specifications [25]. Lee and Bae tested the effects of JP-8 aviation fuel and diesel fuel in a single cylinder, common rail injection system diesel engine under heavy-duty conditions. As the evaporation characteristic of JP-8 aviation fuel is better than the standard diesel fuel. They have observed that it has a wider spray angle and short penetration depth compared to diesel. Moreover, they have observed that the premixed combustion section was improved due to good features of mixture formation of JP-8. The highest heat release of JP-8 aviation fuel was also obtained. While high HC and CO emissions were obtained with JP-8 aviation fuel, they have found that CO emissions decrease [17].

Rakopoulos et al. tested the effects of JP-8 aviation fuel and diesel fuel in direct and pre-chamber diesel engines on exhaust emissions [21]. Arkoudeas et al. experimented and compared biodiesel fuel obtained from the sunflower methyl ester and olive oil ester, (10%, 20% and 50% mixtures with the JP-8 aviation fuel) and the JP-8 aviation fuel in a diesel engine. They determined that biodiesel fuel has shown similar characteristics, and they have observed improvement in the particulate matter emissions with JP-8 fuel [26]. Anastopoulos et al. have tried to improve the lubrication characteristics of kerosene. In their study, they have tested 10 different mono-carboxylic acid esters and kerosene in a direct injection diesel engine. They have concluded that all the tested esters improve the lubrication characteristic of kerosene [27]. Kouremenos et al. investigated the effects of diesel fuel and JP-8 in high speed ordinary four-cycle engine. They have found that no significant differences occurred in the generated emission levels for both fuels. Moreover, they have specified that combustion pressure, intensity increased and combustion stability deteriorated when they used the JP-8 aviation fuel. When JP-8 and diesel fuel were compared, they have observed that JP-8 caused pressure fluctuations (oscillations) in the fuel injection systems depending on the different physical specifications of JP-8 [20]. Durbin et al. tested different fuels (low sulfur diesel fuel, vegetable oil/yellow-grease biodiesel fuel, yellow-grease biodiesel, soy-based biodiesel and military JP-8 fuel) in order to determine the effects on HC, CO, PM and NO_x emissions. They have seen that higher biodiesel blends of fuels showed higher unburned hydrocarbons and CO. But PM emissions reduced with higher biodiesel blends of fuels. Moreover THC and CO emissions increased using JP-8 fuel compared to low sulfur diesel fuel. They used the cetane improver with biodiesel. But they observed that the effects of cetane improver were negligible [28]. Labeckas et al. studied on four-stroke, four-cylinder, and naturally aspirated diesel engine fuelled with diesel fuel and its 5 vol.% (E5), 10 vol.% (E10), and 15 vol.% (E15) blends with anhydrous (99.8%) ethanol (E). In addition, they prepared the blends of ethanol–diesel–biodiesel (E15B) including 15 vol.% of ethanol, 5 vol.% of biodiesel and 80 vol.% of diesel fuel. They researched the effects of ethanol and RME addition to diesel fuel on combustion, engine performance, start of injection and emissions. They showed to develop brake thermal efficiency of 0.362 when the engine fuelled with E15B. Furthermore, NO_x and HC emissions reduced with the addition of ethanol to diesel fuel for richer mixtures [29]. Adaileh and AlQdah experimented the diesel, B5 and B20 test fuels (including 5%, 20% biodiesel respectively) in a diesel engine fuelled with waste vegetable oil at engine speed of 1200–2600 rpm. They have seen that biodiesel showed significant reductions in CO and HC emissions except for NOx. It was also seen that specific fuel consumption increased by 5.95% with biodiesel owing to low heating value. NO_x emissions increased when B5 and B20 are used instead of diesel fuel [30]. Keskin et al. investigated the effects of cotton oil soapstock biodiesel–diesel fuel blends in a diesel engine. They determined that engine torque and power output decreased with cotton oil soapstock biodiesel–diesel fuel blends by 5.8% and 6.2%, respectively. It was also observed that specific fuel consumption increased up to 10.5% when cotton oil soapstock–diesel fuel blends used. Smoke emissions decreased up to 46.6% with blend fuels depending on the amount of biodiesel at maximum engine torque speed [31]. Lee et al. performed an experimental study to analyze the combustion process of JP-8 and diesel fuel in a heavy duty diesel engine. ıt was shown that JP-9 emitted less smoke emissions compared to NO_x and HC emissions. They characterized combustion process by means of image analysis focusing on the luminosity intensity and its spatial distribution. According to test results JP-8 had a longer inition delay compared to diesel fuel. They have seen that flame luminosity of diesel fuel was stronger than JP-8 fuel. They have also implied that diesel fuel had more diffusion dominant combustion [32]. Allen et al. investigated the autoignition characteristics JP-8 and camelina hydroprocessed renewable jet fuel in a rapid compression machine. They measured ignition delays at 670–750 K and low pressures (7 and 10 bar). It was shown that ignition properties of renewable jet fuel were distinct from JP-8. In addition ignition delay for hydroprocessed renewable jet fuel was shorter than JP-8 fuel. Hydroprocessed renewable jet fuel had earlier onset of ignition and more vigorous heat release compared to JP-8. They explained the reason of this case by contenting full paraffinic of hydroprocessed renewable jet fuel [33].

Biodiesel and JP-8 fuel mixtures can be used efficiently in diesel engines. The purpose of this study is to analyze the combustion using biodiesel and JP-8 fuel mixtures and diesel fuel. In this study, the experiments were conducted in a single cylinder, four-stroke, direct injection diesel engine fuelled with biodiesel based sunflower seed oil methyl ester and JP-8 fuel mixtures and diesel fuel at 7.5, 11.25, 15 and 18.75 Nm engine loads at maximum torque speed. The effects of different test fuels on combustion, engine performance and exhaust emissions were investigated at different engine loads.

2. Material and method

In this study, six different test fuels have been tested. The mixture percentages of test fuels are given in Table 1. Test fuels have been obtained by mixing sunflower methyl ester based biodiesel and JP-8 aviation fuel by volume. In order to determine the test fuels on combustion, engine performance and exhaust emission in a single cylinder, four-stroke, and direct injection diesel engine were used. For this purpose, the test engine was operated at a maximum engine torque speed of 2200 min^{-1} and 7.5, 11.25, 15, and 18.75 Nm engine loads. The experiments were performed at constant engine coolant and oil temperature. They were kept constant in order to provide stable operating conditions during the tests. The technical specifications of the test engine are given in Table 2.

High viscosity of biodiesel and low lubrication characteristics are the most significant disadvantages of JP-8 aviation fuel. Insufficient lubrication properties of the JP-8 aviation fuel may damage the engine parts and fuel injection system. Some characteristics of test fuels are given in Table 3. The measurements of the characteristics of fuels have been performed at the laboratories of Turkish Petroleum Refineries (TUPRAS).

The schematic view of the experimental set-up is shown in Fig. 1. Cussons P8160 DC dynamometer was used which can absorb a power of 10 kW in 4000 min^{-1} in the control of engine speed and loads. Engine

Table 1
Percentages of test fuels and their abbreviations.

Abbreviation	Percentages of fuel
B25	25% biodiesel + 75% JP-8
B50	50% biodiesel + 50% JP-8
B75	75% biodiesel + 25% JP-8
B100	100% biodiesel
J100	100% JP-8
Diesel	100% diesel

speed was measured by magnetic pick-up sensor. In-cylinder pressure was measured with water cooled AVL 8QP500c piezoelectric in-cylinder pressure sensor connected with amplifier. At each test, in-cylinder pressure data were measured and recorded by National Instruments USB 6259 data acquisition card (DAQ) with 0.36°CA crank angle intervals. In order to decrease cyclic variations under a specific operating condition, the average of consecutive 50 different cycles was calculated and the in-cylinder pressure has been obtained. The technical specifications of in-cylinder pressure sensor are given in Table 4.

VLT 2600 S opacimeter was used for CO measurement. Testo 350 XL gas analyzer has been used for exhaust emission measurements. The technical specifications of VLT 2600 S opacimeter and emission gas analyzer are given in Tables 5 and 6 respectively. Stopwatch and electronic scale which are able to measure with sensitivity of 0.01 g were used for measuring fuel consumption. Exhaust gas temperature was measured using K-type thermocouple mounted on exhaust line.

Merriam brand Z50MC2-4F model laminar air flow meter was used for the measurement of consumed air by the test engine. Air volume at the laminar air flow meter is measured with pressure difference. Air volume changes linearly depending on the pressure difference. Besides, indicated mean effective pressure was calculated by using network and cylinder volume in the experiments. In-cylinder pressure data were used in order to calculate the heat release rate. Heat release rate has been calculated by Eq. (1) by applying the first law of thermodynamics [34].

$$\frac{dQ}{d\theta} = \frac{k}{k-1} P \frac{dV}{d\theta} + \frac{1}{k-1} V \frac{dP}{d\theta} \tag{1}$$

The first law of thermodynamics was applied for the control volume in the combustion chamber. So, the single zone combustion model was used in order to calculate the net heat release rate. While calculating the net heat release rate, the heat transfer from the cylinder to cylinder walls has been neglected. Heat losses are also affected by the engine speed. As engine speed decreases, there will not be enough time for heat transfer from cylinder to the cylinder walls. It affects the pressure rise rate. Higher engine speeds result less heat losses. It can be also said that higher temperatures obtain and earlier combustion occurs. Heat losses also cause the higher HC emission owing to in complete combustion. In fact, heat losses are a small fraction of fuel energy. But heat losses cannot be converted to useful work due to the heat transfer to the combustion chamber walls. It also belongs to combustion chamber geometry and combustion chamber surface-area-to-volume ratio. Furthermore, leaner air/fuel ratio decreases the burned gas temperature at the end of combustion. It results in the reduction of heat losses to the cylinder walls. It has been deemed that the in-cylinder mass did not change due to gas leakage in segments and valves. Moreover, it has been assumed that charge mixture taken to the cylinder is ideal gas and the thermodynamic characteristics of the combustion chamber are stable [34–36]. The temperature, pressure and the composition of the charge mixture in the cylinder changes in each cycle and cyclic variations occur. The variation of coefficient of indicated mean effective pressure has been calculated as in Eq. (2).

$$COV_{imep} = \frac{\sigma_{imep}}{\overline{X}} \times 100 \tag{2}$$

In Eq. (2), σ_{imep} defines the standard deviation of indicated mean effective pressure for consecutive 50 cycles and $\overline{X}$ defines the average of indicated mean effective pressures [34].

Table 2
Technical specifications of test engine.

Engine type	Four stroke, direct injection
Number of cylinder	1
Cylinder bore [mm]	86
Stroke [mm]	68
Displacement [cc]	395
compression ratio	18/1
Maximum brake torque at 2200 rpm [Nm]	19.6
Injection pressure [bar]	180
Injection advance [°CA, BTDC]	24

3. Results and discussion

Tests were conducted in a single cylinder, direct injection diesel engine, at maximum torque speed (2200 min^{-1}). In Fig. 2, the effects of diesel, biodiesel and JP-8 fuel mixtures on cylinder pressure and heat release at different engine loads are being observed. When the heat release is examined, sudden combustion and diffusion combustion phases are observed. It is noticed that at 7.5 Nm engine load, the combustion is advanced as the amount of biodiesel in the test fuels increases, and the combustion is retarded when J100 and B100 fuels are used. At all loads, maximum heat release has been obtained with J100 fuel. Higher heating value of J100 compared to biodiesel causes the increase of heat release. The cetane number of JP-8 is lower than diesel. It causes longer ignition delay and higher heat release. In addition, the heating value of JP-8 is close to the diesel fuels. More fuel mass is accumulated during the longer ignition delay period. Then all charge mixture starts to ignite at a sudden combustion phase immediately. It results higher heat release rate. It was observed that maximum cylinder pressure and heat release decreased with B75 at 7.5 and 11.25 Nm engine load. It is possible to say that maximum cylinder pressure increases as the biodiesel amount in the fuel mixtures increases at full load (18.75 Nm). It can be also concluded that there is no significant difference in cylinder pressure between B100 and diesel fuel. More heat was released with B75 compared to diesel fuel. It can be explained that more oxygen content of biodiesel and higher heating value of JP-8 increased the heat release rate compared to diesel. At full load, maximum cylinder pressure has been obtained with B100 fuel. Moreover, it was observed that combustion was advanced as the amount of biodiesel increased in the test fuels. Maximum cylinder pressure was obtained with J100 fuel at part loads.

Indicated mean effective pressure refers the average pressure applied on the piston during a cycle for the engine in order to give a maximum power output. In Fig. 3, variations of coefficient of indicated mean effective pressure and equivalence ratio versus on engine load are shown. The temperature, pressure, composition, homogeneity of the charge mixture in the cylinder and thermodynamic characteristics of combustion chamber change by cycle-to-cycle. The change of characteristics of the mixture due to the amount of heat transferred from the combustion chamber to cylinder walls and gas leakages arising at segment, piston and valves cause cyclic variations. It is required for the cyclic differences not to exceed 10% in the internal combustion engines [34]. As mentioned in [37] cylinder pressure fluctuates during the cycle. The differences between the cylinder pressures inherently lead to variations of imep values. During the last stage of compression and the first stage of combustion, the differences become more evident. It is attributed to load fluctuations. In addition flow varies cycle to cycle in the engine and turbulence is not homogenous in the cylinder. The cyclic variations can be appeared when the engine speed is high especially at HCCI operation mode [36–38]. In Fig. 3(a), the variation of coefficient of indicated mean effective pressure versus engine load is given. It is observed that cyclic variations decrease as the engine load increases. Combustion stability is one of the most important factors affecting the cyclic variations. Combustion stability is affected by the dilution effect of fresh charge mixture. As the engine load decreases the combustion stability deteriorates when residual gas is trapped in the combustion chamber or EGR applies especially at idle conditions. Residual gas

Table 3
Characteristics of test fuels.

Fuel type	Method	Diesel	JP-8	Methyl ester
Density (g/cm^3, 15 °C)	ASTM D 1298	0.8372	0.7950	0.8893
Viscosity (cSt, 20 °C)	ASTM D 445	2.8 (40 °C)	3.87(−20 °C)	4.391(40 °C)
CFPP (°C)	ASTM D 2386	−5	−48.5	−2
Flash point (°C)	ASTM D 93	73	41	110
Lower heating value (kJ/kg)	ASTM D 2015	10,450	10,200	9189
Cetane number	ASTM D 976	54	45	58

slows down the rapid heat release rate. It might cause the later combustion. It also affects the cyclic variations [34]. Maximum COV_{imep} has been calculated as 10.77% with J100 fuel at 11.25 Nm engine load. Minimum COV_{imep} has been obtained as 4.84% with B100 fuel at 15 Nm engine load. It was noticed that there was no significant difference in COV_{imep} between diesel and B100 at light and medium engine loads. The low cetane number of JP-8 caused the increase of ignition delay period and the higher cyclic variations especially at low loads. When Fig. 3(b) is examined, it is found that equivalence ratio increases with the increase of engine load. At the same load, the highest equivalence ratio was obtained with J100 fuel. The lower density of J100 causes the decrease of quantity of fuel mass which is taken to the cylinder. In contrast, it results higher equivalence ratio when B100 fuel was used because of its higher density. Moreover, slight decrease of equivalence ratio has been found using biodiesel and JP-8 mixtures. As the equivalence ratio increases, the variation of coefficient of indicated mean effective pressure decreases. It can be mentioned that equivalence ratio decreases with the increase of the amount of biodiesel in the test fuel. Higher equivalence ratio was obtained with diesel compared to biodiesel JP-8 fuel mixtures. It can be also implied that leaner mixtures in the combustion chamber increase the cyclic variations.

In diesel engine, one of the most important parameters is the ignition delay period. As the ignition delay period increases, the amount of fuel which accumulated in the combustion chamber increases in order to ignite. All of fuel and air in the combustion chamber try to contribute to combustion suddenly. So, it causes knocking combustion in the engine. Ignition delay period is affected from the cetane number of the fuel and is being calculated by the determination of fuel injection timing and starting point of combustion [34]. Fig. 4 depicted the ignition delay period. The onset of combustion can be determined by the crank angle where the in-cylinder heat release value increases from negative to positive value as a result of combustion. The period in between the injection advance in terms of crank angle and the point where the heat release reaches a positive value gives the ignition delay period. As in Fig. 4 the start of injection and onset of combustion are evident. The engine was operated at 2200 min^{-1} engine speed. Ignition delay period was calculated in crank angle. Then 2200 min^{-1} engine speed was converted to a crank angle multiplied by 360°CA. It is also possible to determine the crank angle per 1 s. As it is known the ignition delay period in the crank angle we determined the ignition delay period in time by proportioning the crank angle to time.

In Fig. 5, the variation of ignition delay versus engine load is given. It is observed that the ignition delay period decreases as the engine load increases for all test fuels. The increase of energy amount taken to cylinder causes the increase of pressure and temperature at the end of combustion. Thus the first flame kernel occurs earlier. It is also observed that the ignition delay decreases as the amount of biodiesel increases in the test fuels. It is concluded that ignition delay shortens owing to the higher amount of cetane number of biodiesel than the JP-8 fuel. Besides, higher amount of oxygen molecules within the biodiesel improve the ignition conditions and accelerate combustion reactions. Consequently, the ignition delay period shortens. Maximum ignition delay period has been calculated with the J100 fuel.

Specific fuel consumption specifies the consumed fuel amount per unit power at unit time. The variation of specific fuel consumption is one of the most significant parameters in the determination of the engine performance. The variation of specific fuel consumptions with different test fuels versus engine load is given in Fig. 6. As the engine load

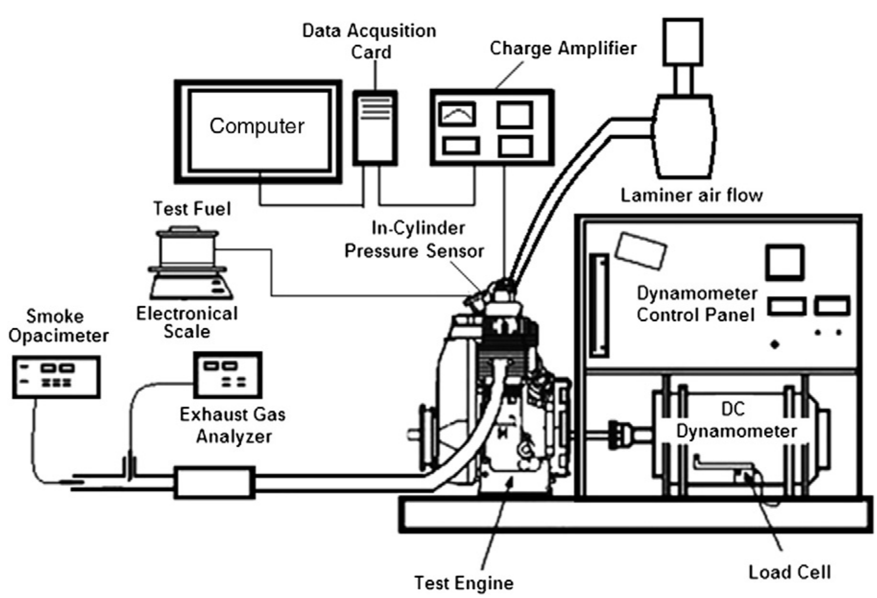

Fig. 1. The schematic view of the experimental set-up.

Table 4
Technical specifications of in-cylinder pressure sensor.

Model	AVL 8QP500c quartz
Operating range [bar]	0–150
Measurement precision [pC/bar]	11.96
Linearity [%]	±0.6
Natural frequency [kHz]	>100

Table 5
Technical specifications of VLT 2600 S opacimeter.

Parameter	Operating range	Accuracy
Soot intensity [%]	0–99	0.01
k soot factor [m^{-1}]	0–10	0.01
Engine speed [rpm]	0–9999	1

Table 6
Technical specifications of Testo 350 XL emission gas analyzer.

Combustion products	Operating range	Accuracy
Oxygen (O_2) [vol.%]	0–25	+/−0.2 mV
Carbon monoxide (CO) [ppm]	0–10000	5 ppm (0–99 ppm)
Carbon dioxide (CO_2) [vol.%]	0–50	±0.3 vol.% +1 mV.% (0–25 vol.%)
Hydrocarbon (HC) [%]	0.01–4	<400 ppm (100–4000 ppm)
Azot Oksit (NOx) [ppm]	0–3000	5 ppm (0–99 ppm)

increases, the specific fuel consumption decreases. So, specific fuel consumption increases for all test fuels. Minimum specific fuel consumption was calculated with the diesel fuel due to the highest heating value of diesel fuel. It is found that as the amount of biodiesel in the test fuels increases specific fuel consumption also increases. Biodiesel's viscosity is high and heating value is low. When the amount of biodiesel in the test fuels increases, the engine consumes more fuel in order to provide the same effective engine power, thus, the specific fuel consumption increases. The specific fuel consumption decreases with J100 due to its higher heating value than biodiesel. Maximum specific fuel consumption has been calculated with the B100 fuel. As mentioned in [26] specific fuel consumption increased with the increase of biodiesel in the blend fuels.

In Fig. 7, the effects of engine load on exhaust gas temperatures are depicted with different test fuels. As shown in Fig. 7, it can be concluded that the exhaust gas temperature increases as the engine load increases for all test fuels. The increase of charge mixture taken to the cylinder increases the end of combustion gas temperature and exhaust gas temperature. Minimum exhaust gas temperatures were measured with J100 fuel, and the maximum exhaust gas temperatures were measured with B100 fuel. The increase amount of biodiesel in the test fuels increases the oxygen content of test fuel, therefore, the end of combustion gas temperature increases. It results higher the end of combustion temperature. It can be also concluded that exhaust gas temperature is higher when biodiesel JP-8 fuel mixtures are used. Besides the viscosity of biodiesel is higher than JP-8. This results more injection of biodiesel to the cylinder. In conclusion, high exhaust gas temperatures are obtained with biodiesel. Later combustion occurred with JP-8 fuel at all engine

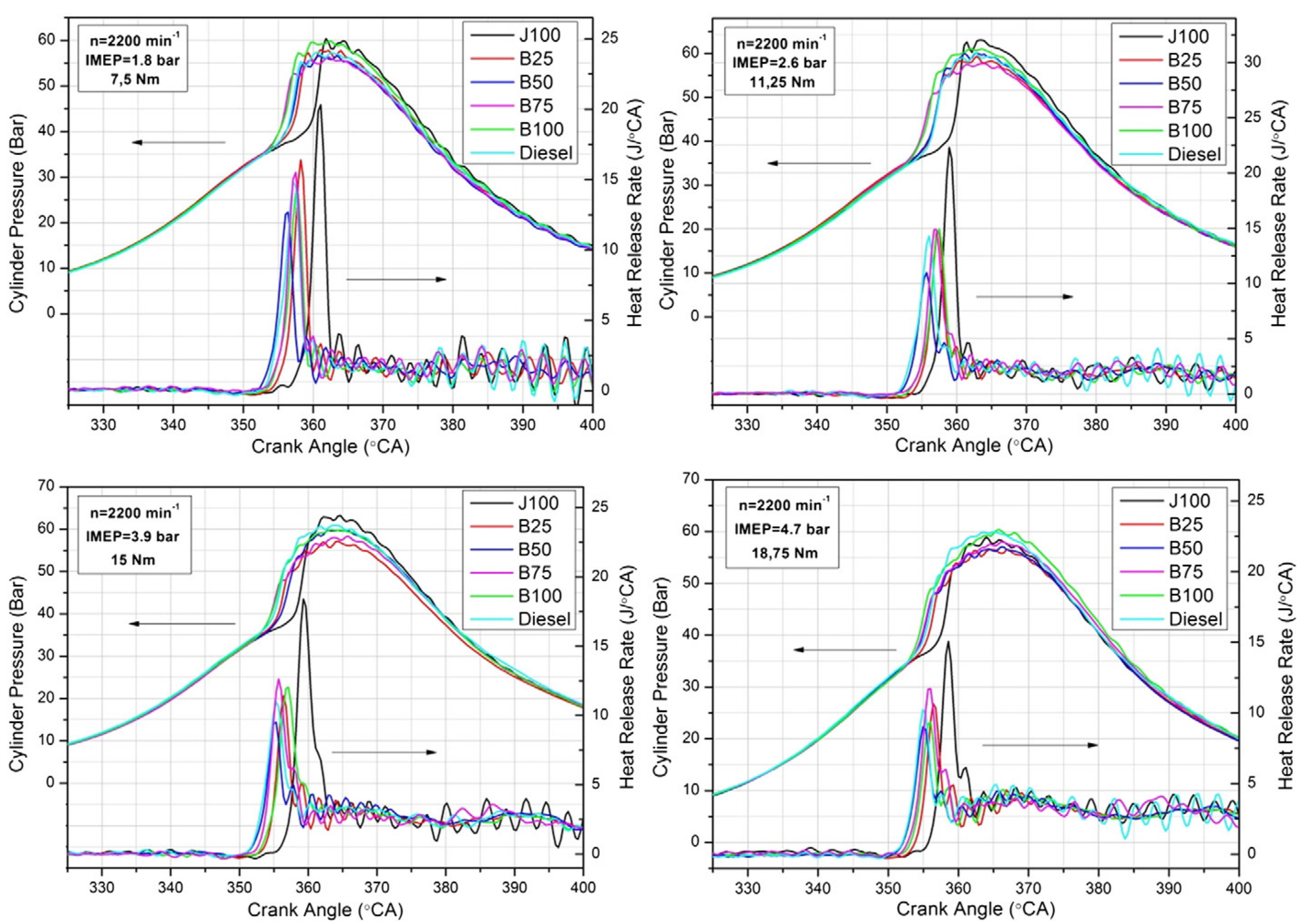

Fig. 2. Cylinder pressure of test fuels at different engine loads and their effects on heat release rate.

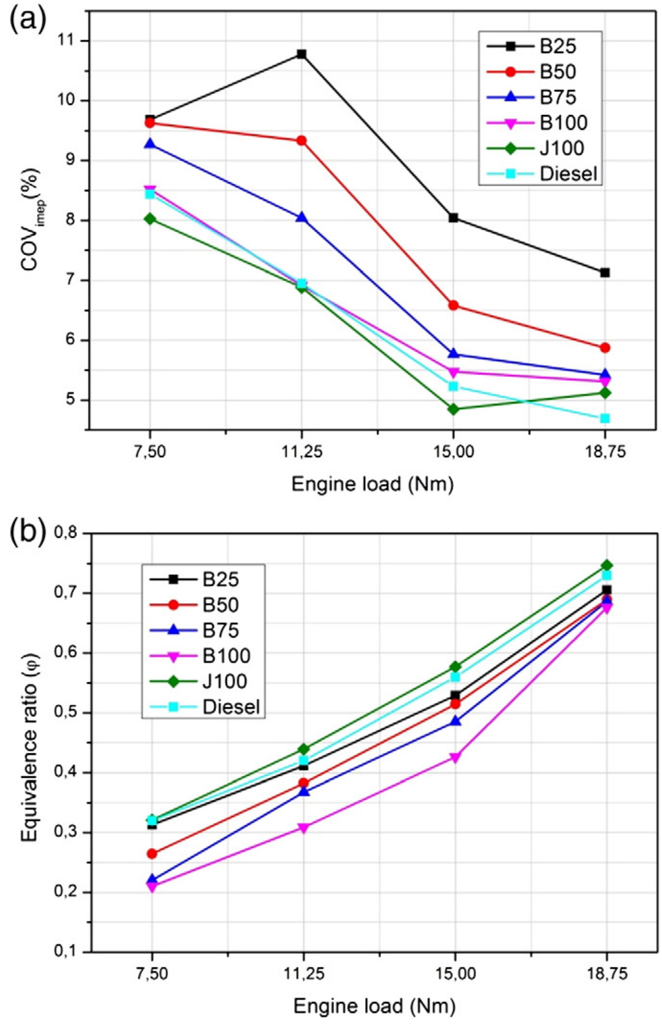

Fig. 3. Variations of coefficient of indicated mean effective pressure and equivalence ratio versus on engine load.

loads and combustion occurred on more diffusion combustion phase. So, combustion process was extended towards the expansion stroke. In the meantime, the piston is too far away from the top dead center and the in-cylinder temperature decreased. It causes lower exhaust gas temperature.

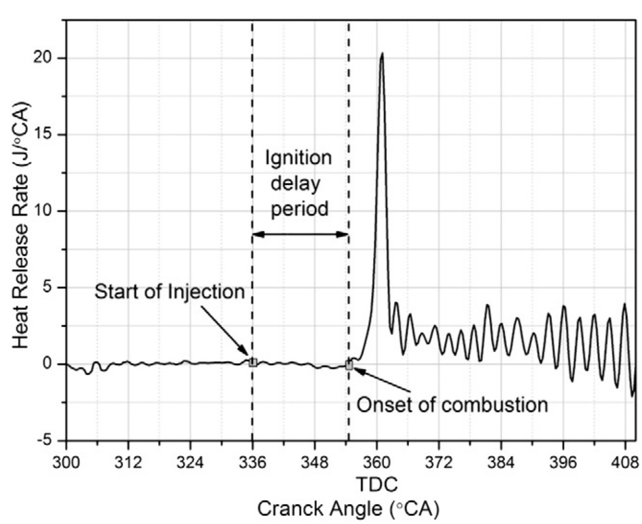

Fig. 4. Ignition delay period.

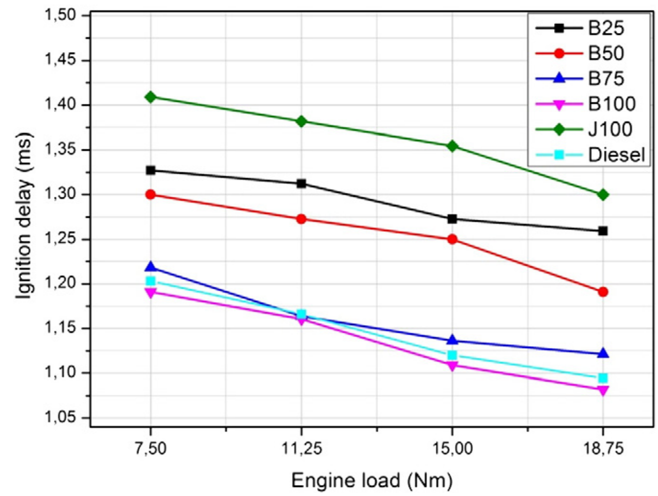

Fig. 5. The variation of ignition delay versus engine load.

In Fig. 8, the variation of NO_x emissions versus engine load is shown. NO_x emissions are generated depending on high gas temperatures at the end of the combustion. As the amount of charge mixture to the cylinder increases, in-cylinder temperature increases at the end of compression. Thus the end of combustion temperature increases. At high gas temperatures, nitrogen and oxygen molecules react and generate NO_x emissions. The higher oxygen content results higher end of combustion temperature. This is called thermal NO_x formation. The density of biodiesel is more than the JP-8 fuel. Moreover, more oxygen is included in its chemical structure compared to JP-8 fuel. More oxygen content of biodiesel improves the combustion reactions, and causes an increase to the end of combustion gas temperature. Minimum NO_x emissions have been measured with J100 fuel. As the amount of biodiesel within the test fuels increases, NO_x emissions increase. As shown in Fig. 8, NO_x emissions obtained with diesel are lower than biodiesel. The cetane number of diesel is lower than biodiesel but it has higher heating value than biodiesel. It can be said that the higher density and lower heating value of biodiesel result lower heat release rate compared to JP-8. But, NO_x emissions increased due to the higher oxygen content and density of biodiesel. The end of combustion temperature increases as more fuel mass is injected to the cylinder. So, this improves and accelerates the oxidation reactions between oxygen and nitrogen molecules. In reference [26], it is possible to say that similar test results were obtained. NO_x emissions increased especially with 50% sunflower oil biodiesel blend fuel. Highest heat release was obtained for all engine loads because,

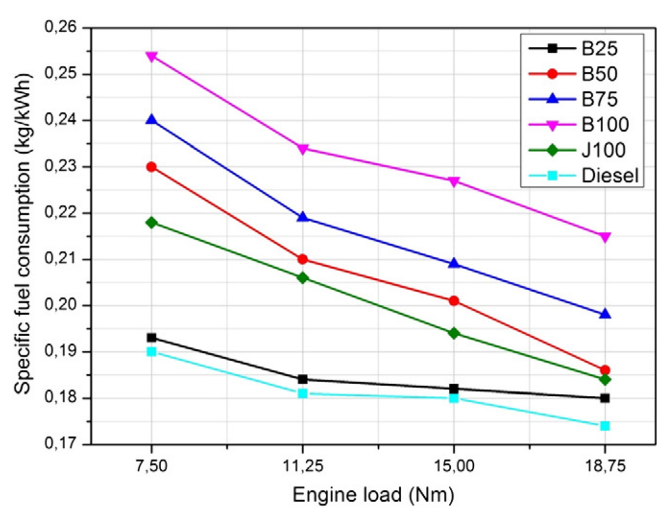

Fig. 6. The effects of engine load on specific fuel consumption.

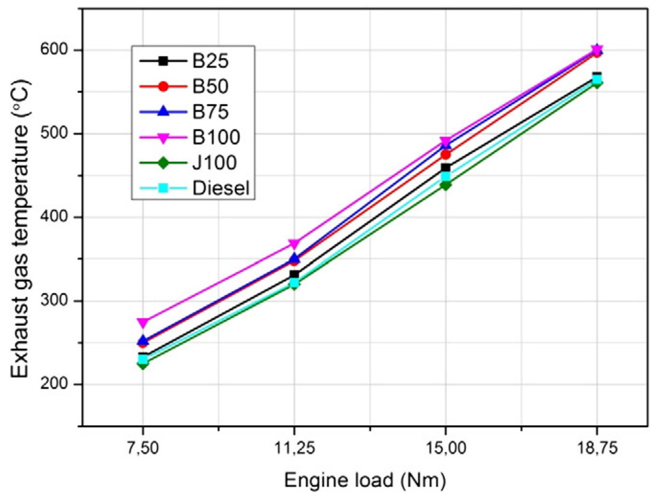

Fig. 7. The variation of exhaust gas temperature versus engine load.

the cetane number of JP-8 is lower than biodiesel and diesel fuel. It results longer ignition delay with JP-8 compared to other test fuels. More fuel is accumulated in order to ignite during the longer ignition delay period. Higher combustion temperatures are obtained with the result of combustion of more fuel thus, NO_x emission increases.

Fig. 9 illustrated the variation of CO and soot emissions versus engine load. CO emission is the harmful exhaust gas coming out as the result of incomplete combustion. Insufficient temperature and oxygen in combustion chamber increase the CO emissions [34,35]. CO emissions increase as the engine load increases for all test fuels. It is also observed that CO emissions decrease as the amount of biodiesel in the test fuels increases. Oxygen content of biodiesel improves the oxidation reactions, and more fuel is contributed to combustion. Having sufficient oxygen in the test fuels causes the generation of CO_2 emissions through the oxidation of carbon monoxide. CO and soot emissions increase with the increase of engine load. As the engine load increases, the oxygen concentration in the combustion chamber decreases, thus, oxidation reactions slow down. As a result of insufficient turbulence in combustion chamber, the homogenous air/fuel mixture is prevented, and it causes CO formations and increase of soot emissions. Soot emissions are solid particles coming out as the result of combustion. As the amount of biodiesel in the test fuels increases the soot emissions decrease. The oxygen content of biodiesel increases the combustion rate; hence, the generation of CO can be prevented. Higher heat release rate are obtained with B25, B75, and B100 compared to diesel fuel apart from at full load. It can be concluded that higher heat is released to the combustion chamber by using B25, B75, and B100 test fuels. It results better combustion rate due to warmer combustion chamber. CO emissions obtained with diesel are higher than biodiesel–JP-8 fuel mixtures for all engine loads. It can be noticed that there is no significant heat release rate difference between B100 and diesel fuel. Moreover, it is also possible to say that higher heat release rate is obtained with B100 compared to diesel fuel at part loads. Higher heat release rate increases the combustion chamber temperature. It accelerates the oxidation reactions. The highest CO and soot emissions are obtained with J100.

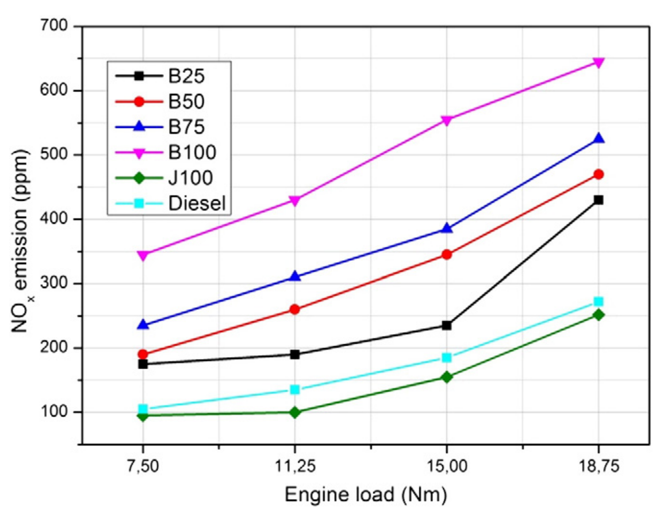

Fig. 8. The variation of NO_x emissions versus engine load.

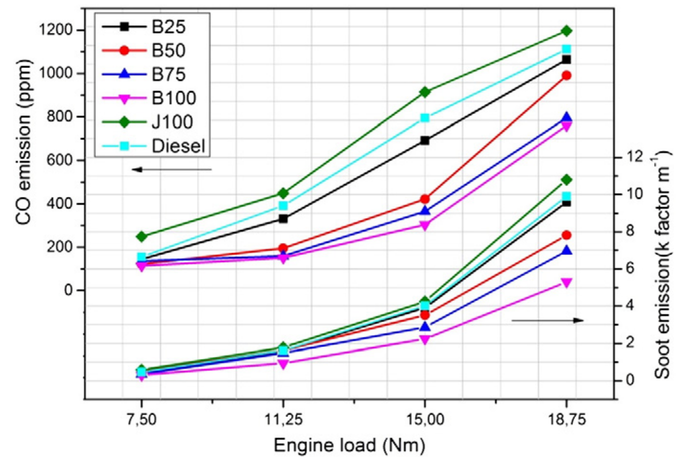

Fig. 9. The variation of CO and soot emissions.

4. Conclusions

The widespread use of diesel engines in transportation and shipment and the pollution of atmosphere and environment by the exhaust emissions caused by diesel engines cause the alternative fuel researches for these engines to be continued in a rapid manner. At this point, JP-8 aviation and biodiesel fuel mixtures can be used as alternative fuel in diesel engines. The effects of six different test fuels (B25, B50, B75, B100, J100 and diesel) on combustion, engine performance, exhaust emissions in a single cylinder, and direct injection diesel engine were experimented at maximum torque speed (2200 min^{-1}) and four different engine loads (7.5, 11.25, 15 and 18.75 Nm). The test results showed that maximum cylinder pressure increased as the amount of biodiesel increased and maximum cylinder pressure was obtained with B100 fuel at full load (18.75 Nm). Moreover, the combustion was advanced as the amount of biodiesel fuel increased. Equivalence ratio increased and COV_{imep} decreased as the engine load increased. Leaner mixture in the combustion chamber increases the cyclic variations. Ignition delay decreased as the engine load increased. As a result of an increase of energy taken to the cylinder, the pressure and temperature in combustion chamber increase, and the ignition delay period shortens. Besides, the ignition delay has shortened as the amount of biodiesel in the test fuels increases. The low cetane number of JP-8 fuel causes the increase of ignition delay period. The specific fuel consumption per unit power increased when biodiesel was used because of higher density and low heating value. As the engine load increases, specific fuel consumption decreases for all test fuels. Oxygen content of biodiesel improves the oxidation reactions. For this reason, combustion gas temperature increases at the end of combustion. Consequently, NO_x emissions increase but CO emissions decrease as the amount of biodiesel in the test fuels increases. Higher oxygen content of biodiesel accelerates the oxidation reactions, and causes the reaction of carbon monoxide. Consequently, it has been observed that JP-8 and biodiesel fuel mixtures can easily and efficiently be used in diesel engine. It has been observed that the engine's performance might be increased and exhaust emissions can be decreased as a result of the usage of different fuel mixtures.

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