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Original Research Article

An Experimental Investigation on The Effects of Waste Olive Oil Biodiesel on Combustion, Engine Performance and Exhaust Emissions

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ABSTRACT

In this study, the biodiesel obtained from the waste olive oil by transesterification method has been mixed with a 30% of diesel fuel as volume and tested with a single cylinder direct injection diesel engine. The main purpose of this study is to obtain purer biodiesel from waste olive oil using methyl alcohol (CH_3OH) and sodium hydroxide (NaOH) as catalyst in the transesterification method and research performance, combustion and emission characteristics in detail in a direct injection diesel engine. The combustion, engine performance and exhaust emission values have been also compared with diesel fuel. The test engine was operated at a constant speed of 2200 rpm and different engine loads such as 3.25 Nm, 7.5 Nm, 11.25 Nm, 18.75 Nm. According to the experimental results, the thermal efficiency of biodiesel is lower by about 1% to 5% than diesel. CO is lower about 37.5 % with biodiesel than that of diesel at 18.75 Nm. CO_2 is higher 41% with biodiesel than diesel at 11.25 Nm. NO_x was measured 9.5% higher than diesel fuel at 18.75 Nm. Soot emissions decreased by 37.5% compared to diesel.

Keywords: Biodiesel, Exhaust Emissions, Transesterification, Engine Performance, Combustion

1. Introduction

Nowadays, energy needs of countries are increasing rapidly due to industrialization and technological development. Due to the energy crisis, countries have begun to focus on developing alternative energy sources to meet their energy needs. Therefore, the development of renewable energy sources with low emissions that can be used in diesel engines is very important for both environment and human health [1]. One of the alternative fuels that can be used in compression-ignition engines is biodiesel obtained from vegetable or animal fats [2]. Direct usage of vegetable oils in diesel engines leads to problems such as insufficient atomization due to high viscosity, high levels of smoke emission, incomplete combustion, carbon deposits, ring adhesion and obstruction of the injectors. Despite the high flash point of vegetable oils, low volatility is another important disadvantage [3]. The density and viscosity of biodiesel fuels is higher than that of diesel and cold working properties are not better than standard diesel fuel. At this point, it can cause fuel injection system problems. It is possible to use biodiesel fuels in internal combustion engines by reducing disadvantaged aspects. Therefore, methods such as microemulsions, pyrolysis and transesterification have been performed to use vegetable oils as alternative fuels in diesel engines. Among these methods, transesterification have been widely used [4]. Transesterification reaction is alkaline catalyzed, acid catalyzed and enzyme catalyzed [4-7]. Dorado et al [8] investigated the chemical parameters of waste olive oil biodiesel. The results showed that using KOH and methanol gave better results than using NaOH and ethanol. Abed et al [9] researched the effects of cooking-oil biodiesel on engine performance and exhaust emissions of a Kirloskar make, single cylinder, four strokes, and direct injection diesel engine with a developing power of 5.775 kW at 1500 rpm. They have found that thermal efficiency of waste cooking-oil biodiesel were lower than that of diesel fuel and specific fuel consumptions were higher. Gao et al. [10] investigated the effects of waste cooking oil biodiesel on kinetics, fuel properties and engine performance and emissions. They reported that engine torque of biodiesel is lower than diesel fuel. Shen et al [11] evaluated the effects of waste cooking oil biodiesel blends on the exhaust emissions and fuel consumption of two light-duty diesel truck and two heavy-duty diesel trucks. The results showed that the fuel consumption and CO_2 emissions of biodiesel blends were higher than that of neat diesel fuel due to the lower calorific value of biodiesel blends. Yesilyurt [12] presented the effects of the fuel injection pressure on the performance and emission characteristics of a diesel engine fueled with waste cooking oil biodiesel-diesel blends. The experiments showed that engine torque, brake power, CO, UHC and smoke opacity decreased and BSFC, exhaust gas temperature, NO_x and CO_2 emissions increased for the biodiesel fuel blends. Akar et al [13] evaluated the effects of waste oil biodiesel on the engine performance and exhaust emission characteristics of a compression ignition engine. They have seen that using biodiesel deteriorated performance and emission parameters except CO. Rajak and Verma [14] presented the effects of edible and non-edible vegetable oil, animal fats, waste oil and alcohol in CI engine. The results showed that the PM emissions and smoke emissions reduced and NO_x pollutant increased with the usage of biodiesel. Gharehghani et al [15] evaluated the effects of waste fish oil biodiesel on diesel engine combustion characteristics and emissions at a single cylinder E6 Ricardo engine. They reported that in-cylinder pressure increased, heat release rate duration shortened with fish oil biodiesel. Garcia-Martin et al [16] produced waste cooking oil biodiesel in an oscillatory flow reactor (OFR) and tested engine performance of a TDI diesel engine. The results showed that biodiesel yield in OFR was higher than in stirred tank reactor. The particle number concentration decreased for especially B50 fuel (38%). Dorado et al [17] investigated the effects of waste olive oil methyl ester in a Diesel direct injection Perkins engine at several steady-state operating conditions. They have found that CO, CO_2, NO and SO_2 decreased and NO_2 increased with the usage of olive oil biodiesel methyl ester when compared to the conventional diesel fuel. Lapuerta et al [18] evaluated the effects of the alcohol type used in the production of waste cooking oil biodiesel and tested the fuels in a 2.2 l, common rail injection diesel engine. The results showed that fuel consumption slightly increased and sharp reduction in total hydrocarbon emissions, smoke opacity and particle emissions when compared to diesel engine. Lapuerta et al [19] investigated the effects of cooking oil biodiesel on particulate emissions of DI diesel commercial engine at a typical road conditions. Smoke and particulate matter emissions were sharply decreased as the biodiesel concentration was increased. Kalligeros et al [20] tested different biodiesel (rape seed oil, sunflower oil, olive oil)/marine diesel blends in single cylinder, indirect injection diesel engine and detected fuel consumption and exhaust emissions. The reduction of HCs and CO emissions were explained with the faster evaporation and more stable combustion. Dorado et al [21] evaluated waste olive oil methyl ester as a fuel for diesel engines during 50-h short term performance test in a disel direct injection Perkins engine. They have seen that slight power loss and brake-specific fuel consumption increase. Energy conversion efficiency remained constant or showed slight increase for waste olive oil methyl ester when compared to No.2 Diesel fuel. David et al. [22] investigated the mutagenicity emission factors of canola oil and waste vegetable oil biodiesel. ıt was seen that utagenic potencies of canola and waste vegetable oil emissions are higher than B0. Nawel et al. [23] aimed to reduce energy and reactant consymptions to obtain green process. They showed that maximum oil conversion of 100%, 2.69 kJ minimum energy consumption and 77.36% maximum green chemistry balance with KOH catalyst concentration of 2 wt%. Carmano-cabello et al. [24] intended to optimize biodiesel production with lipid fraction included in solid food waste. They analyzed biodiesel quality with cold properties (below-3 °C). Conversion efficiency should be enough high in transesterification method in view of purer and reasonable biodiesel production. So, production process and catalysts are evident in order to obtain good biodiesel. This phonomena directly affects the combustion phasing and performance. So, there is knowledge gap in this field. In the current study, biodiesel was obtained from waste olive oil by transesterification method using methyl alcohol (CH_3OH) and sodium hydroxide (NaOH) as catalysts. High level of biodiesel addition to pure diesel deteriorated the properties of test fuel, injection and combustion characteristics due to lower calorific value and higher viscosity and density. Hence, the produced biodiesel was mixed with 30% by volume of diesel fuel. So, B30 fuel was experimentally tested in a single-cylinder, natural-aspirated, direct-injection diesel engine and its effects on combustion, engine performance and exhaust emissions.

2. Experimental Setup and Procedures

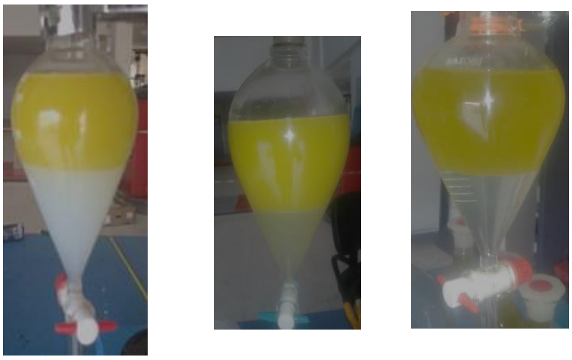
The engine test and emission measurements of the biodiesel produced by the transesterification method from waste olive oil were carried out.

2.1. Biodiesel production

In the current study, biodiesel production was carried out by transesterification method from waste olive oil. Methyl alcohol (CH_3OH) and sodium hydroxide (NaOH) as catalyst were used during production. In the biodiesel purification process, water washing method was applied. Diesel fuel is mixed with 30% olive oil methyl ester and tested in a single cylinder, direct injection engine and its effects on engine performance and exhaust emission were investigated. Glass balloon, cooler, flat glass balloon, separating hopper, kinematic viscometer, density measuring device, cetane number and cetane index measuring device, water determination measuring device, sulfur amount measuring device, magnetic stirrer, magnetic fish, precision balance were used in biodiesel production. The mixture and heating process are carried out by adding the crude oil into the methoxide after the methanol and the catalyst are completely mixed and the methoxide is formed. Raw oil and methoxide on magnetic cooker is allowed to be mixed by rotating magnetic fish by means of magnetic heater at 40 rpm for 1 hour at 60ºC temperature. Temperature makes easier to occur transesterification reaction. During these heating operations, the temperature control is kept under observation with a thermometer for a certain period of time and the magnetic heater is reached to the required temperature. The aim of this process is to prevent the methanol from starting to boil about 63-67 ºC. Refrigerant pipe (cooler) circulates water. In this way, it prevents the evaporation of the evaporating alcohol due to the increase of heat by condensing with the help of cold water. The mixture was allowed to rest for an average of 12 hours at the end of the mixing process. The biodiesel and glycerin decomposed at the end of this waiting period. Glycerin was withdrawn from the biodiesel by separating funnel due to the difference of the density of glycerin and biodiesel. The produced biodiesel must be washed to remove unreacted alcohols, fatty acids and catalyst material. Otherwise, these substances in the fuel may cause abrasive effect on rubber or rubber connected engine parts.

Biodiesel, separated from glycerin, is subjected to biodiesel washing process to separate the fatty acids and catalyst residues of alcohols remaining in biodiesel by keeping for half an hour in 100 ^{o}C boiled water in separating funnel.

As shown in Figure 1, the washing process was repeated using the bubble method by adding boiled water in the biodiesel.



a) First washing b) Second washing c) Third washing

Figure 1. The washing process of biodiesel

This process continues until the color of the water becomes clear. Thus, the purest biodiesel is obtained. After separating biodiesel from water, biodiesel is applied to drying process to remove water completely from the biodiesel. Drying process continues at evaporation temperature of water above 100 ºC in the 110-120 ºC temperature range until evaporation ends. It was put into the beaker on the heated magnetic stirrer and it was heated by stirring with the magnetic fish.

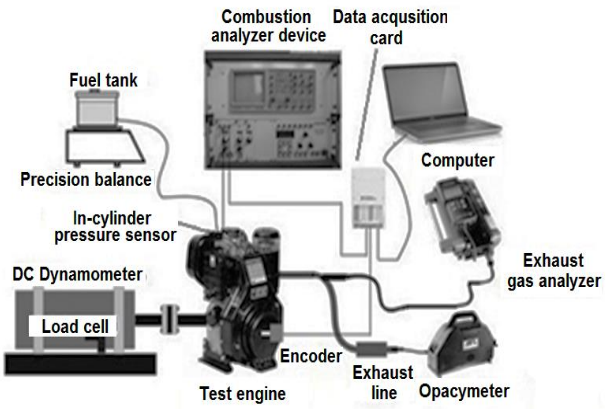


Figure 2. Schematic view of the experimental setup

2.2. Engine tests and emissions measurement

The effects of waste olive oil biodiesel on combustion, performance and exhaust emissions have been investigated experimentally. For this purpose, the test engine was operated at 2200 rpm and at different loads (3.75, 7.5, 11.25, 15 and 18.75 Nm). The schematic view of the experimental setup is shown in Figure 2. The experiments were carried out at Gazi University, Faculty of Technology, Automotive Engineering Department Internal Combustion Engines Laboratory.

The experiments were conducted in a single cylinder, direct injection, naturally aspirated diesel engine. The technical specifications of the test engine are given in Table 1.

Table 1. The technical specifications of the test engine [25]

Model	Antor / 6LD400
Engine Type	Direct injection, naturally aspirated
Cylinder number	1
Bore x Stroke [mm]	86 x 68
Swept volume [cm^3]	395
Compression ratio	18:1
Maximum power [kW]	5.4 @ 3000 rpm
Maximum torque [Nm]	19.6 @ 2200 rpm
Combustion chamber geometry	ω type
Fuel injection system	PF Jerk type fuel pump
Injection nozzle	0.24 [mm] x 4 hole x 160°
Injection timing [ºCA]	24 BTDC

The experiments were carried out at constant intake air inlet (25 °C) and engine oil temperatures (80 °C ± 1). The tests and measurements were performed after the engine was reached to operating temperature. The test engine was coupled with Cussons P8160 test bed and loaded with a DC dynamometer as shown in Figure 3. The dynamometer can able to absorb 10 kW of power at 4000 rpm. Engine load can be determined by strain gauge load cell. Magnetic pick-up sensor was used to measure engine speed. Engine oil and intake air inlet temperatures were measured with K-type thermocouples and temperatures were kept constant during the experiments. AVL 8QP500c quartz model water cooled pressure sensor was used for measuring cylinder pressure. After the cylinder pressure data is taken from the pressure sensor, the data was amplified with Cussons P4110 combustion analyzer and the analog data was transferred to the data acquisition card.

Diesel and B30 (30% biodiesel-70% diesel) test fuels were used in the experiments. The properties of test fuels are shown in Table 2.

Analog cylinder pressure data has been converted to digital data using National Instruments USB 6259 brand data acquisition card. Digital pressure data was then recorded on the computer. The encoder was mounted on the crankshaft, which produces 360 pulses per revolution. It was also used to determine the engine speed and the top dead center (TDC).

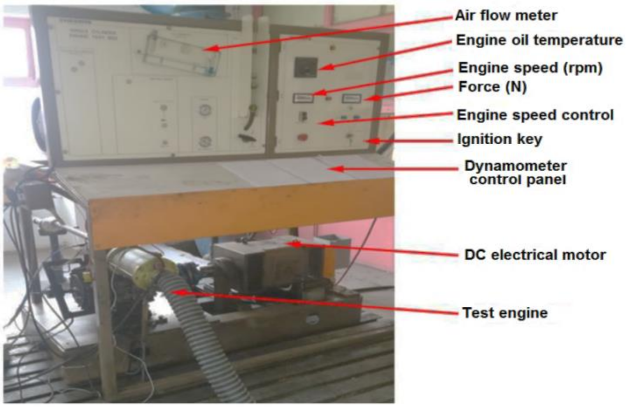


Figure 3. Test setup

Table 2. The properties of test fuels

Properties	Diesel	Biodiesel
Calorific value [kJ/kg]	45343	39488
Density [kg/m^3 @15 °C]	842	921
Kinematic viscosity [cst @ 40 °C]	2.44	3.6-4.42
Freezing point [°C]	<-5	6
Cetane number	>50	57

Crank angle information was measured at intervals of 0.36°CA. The cylinder pressure of consecutive 50 cycles were averaged to reduce cyclic differences. For combustion analysis, combustion characteristics such as heat release, start of combustion and combustion duration were determined by using cylinder pressure data. Exhaust emissions in the tests were measured with the Testo exhaust gas analyzer with the technical specifications given in Table 3.

Table 3. Technical specifications of Testo Exhaust gas analyzer

Combustion products	Operating range	Accuracy
O_2 [vol.%]	0–25	±2 mV
CO [ppm]	0–10000	5 ppm (0–99 ppm)
CO_2 [vol.%]	0–50	±0,3 vol.% +1 mV.% (0–25) vol.%)
HC [%]	0.01–4	<400 ppm (100–4000 ppm)
NO_x [ppm]	0–3000	5 ppm (0–99 ppm)

Table 4. Technical specifications of AVL Di-Smoke 4000

Analyzer	AVL DiSmoke 4000	
Measurement method	Partial flow	
	Opacity	K value
Operating range	0-100 %	0.1 %
Accuracy [m^{-1}]	0-99.99	0.01

Soot emissions in the tests were measured with the AVL Di-Smoke 4000 opacimeter given the technical specifications in Table 4.

In the experiments, the variables such as thermal efficiency and combustion characteristics were determined at constant engine speed. Heat release rate was determined using cylinder pressure depending on the first law of thermodynamics. When calculating the heat release, it is assumed that the in-cylinder charge mixture is the ideal gas. It is thought that there are no gas leakages. Equation (1) was used to calculate the heat release rate.

$$\frac{dQ}{d\theta}=\frac{k}{k-1}P\frac{dV}{d\theta}+\frac{1}{k-1}V\frac{dP}{d\theta}+\frac{dQ_{heat}}{d\theta} \qquad (1)$$

In Equation (1), dQ represents heat release, dQ_{heat} heat transferred to cylinder walls, dθ crank angle, and k ratio of specific heat. Cylinder pressure and fuel line pressure must be determined for combustion analysis. Figure 4-a shows the change in fuel line pressure due to the crank angle. The first derivative of cylinder pressure, heat release and cylinder pressure is shown in Figure 4-b.

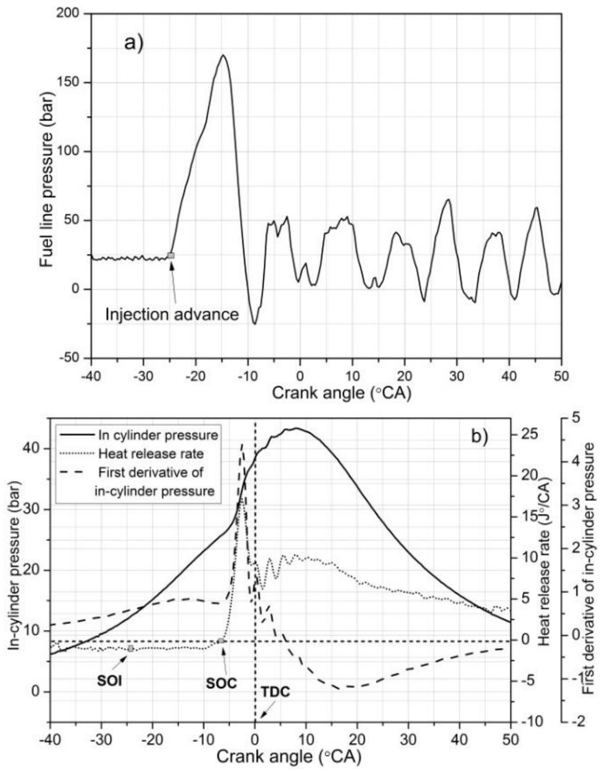


Figure 4. a) Fuel line pressure b) The first derivative of cylinder pressure, heat release and cylinder pressure

In the experiments, the start of combustion was accepted as the point where the heat release increased from negative value to positive value according to the crank angle. The time between the start of the injection and the start of the combustion indicates the ignition delay. The original start of injection timing of the test engine is 24 °CA BTDC. It is difficult to determine when the combustion ends. However, the end of combustion is defined as the point at which 90% of the mixture is completed. The heat from the cylinder to the walls depends on the thermodynamic conditions in the combustion chamber [26-28].

3. Results and Discussions

3.1. Cylinder pressure change

As engine load increased, in-cylinder pressures increased. The maximum in-cylinder pressure was obtained ATDC at high engine load. When B30 was used at 18.75 Nm load, it was seen that the in-cylinder pressure is higher than diesel (about 44 bar). When comparing with diesel, B30 presented higher in-cylinder pressure with the increase of engine load. It can be implied that high density of biodiesel caused to increase in-cylinder pressure due to higher density during injection. The variations of in-cylinder pressure are shown in Figure 5. Besides, higher oxygen content of biodiesel improves oxidation reactions in the combustion chamber. Hence, fuel can be well combusted in the presence of sufficient oxygen. At higher engine loads, more time is required in order to complete combustion. So, combustion is retarded. This phonomena is especially noticed with B30 due to higher viscosity and density of waste oil biodiesel. Uyumaz et al. [29] showed that the addition of sunflower oil biodiesel caused to increase in-cylinder pressure. Uyumaz [30] noticed that in-cylinder pressure increased with the increase of mustard oil biodiesel addition into pure diesel. He found that the highest in-cylinder pressure was measured with M30 at 15 Nm engine load.

Uyaroğlu et al. [31] mentioned that no big differences were seen on cylinder pressure between hazelnut oil–diesel (H30), corn oil–diesel (C30), soybean oil–diesel (S30), sunflower oil–diesel (Su30), canola oil–diesel (Ca30) and Crambe abyssinica oil–diesel (Cr30) and diesel fuel blends. They have also stated that eralier combustion occured with biodiesel–diesel fuel blends. Uyumaz [30] noticed that there was no big difference between diesel and mustard oil biodiesel-diesel blends on heat release rate. Can et al [32] showed that the addition of soybean biodiesel into diesel resulted in higher in-cylinder pressure ad heat release with EGR. On the other hand, there are studies in the literature that biodiesel addition showed lower in-cylinder pressure and heat release [33-36].

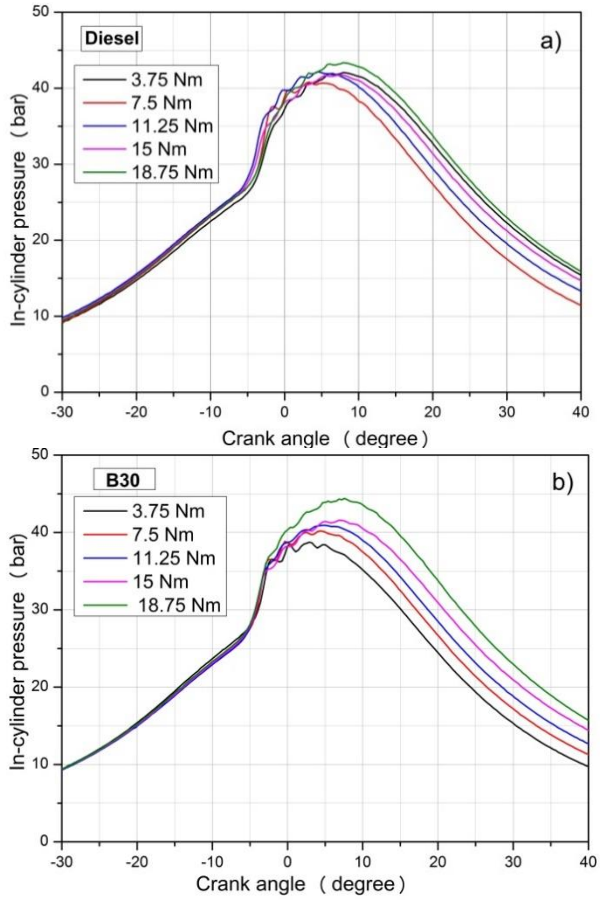


Figure 5. a) The variations of in-cylinder pressure of diesel fuel b) The variations of in-cylinder pressure of B30 fuel

The longer the ignition delay, the more fuel is injected into the cylinder before combustion. This causes fuel accumulation in the cylinder and the sudden combustion of excess fuel molecules increase the maximum in-cylinder pressure [37]. At lower loads, lower fuel is sent to the cylinder and the maximum in-cylinder pressure is getting lower. As the amount of fuel injected increases with the increase of engine load, in-cylinder gas pressure increases. As the engine load increases, the maximum cylinder gas pressure moves away from the TDC [38].

3.2. The heat release rate change versus crank angle

When heat release rate is examined, the maximum heat release rate was obtained as 23 J/°CA at 7.5 Nm engine load at 3 °CA BTDC with diesel fuel. In the case of B30 fuel, the heat release rate is approximately 22 J/°CA at the same engine load at the same crank angle.

The variations of heat release rate are shown in Figure 6. Heat release was observed to reach the maximum level at 7.5 Nm engine load for both test fuels.

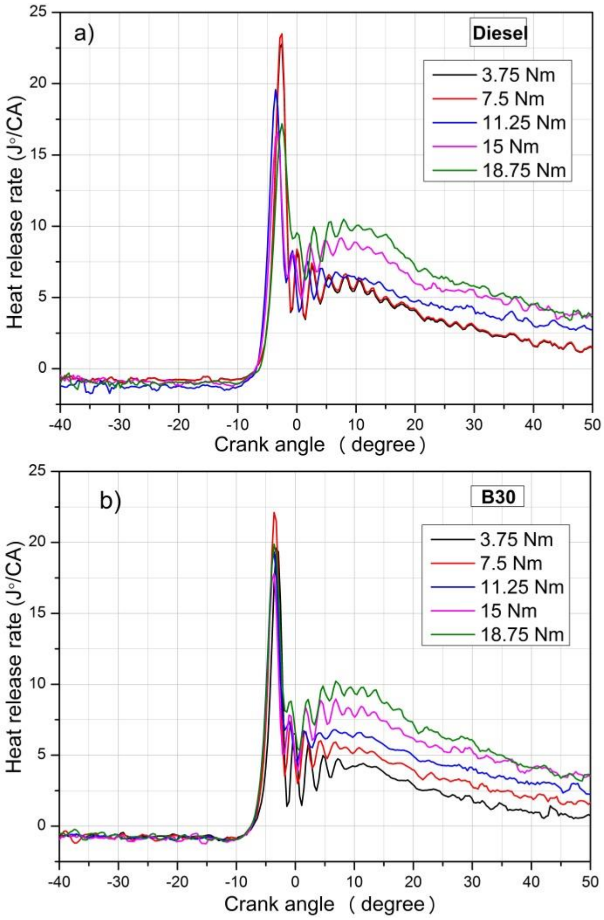


Figure 6. a) The variations of heat release rate with diesel b) The variations of heat release rate with B30

The heat release rate is similar in the same engine torque and crank angle for both fuel. In terms of the crank angle, sudden combustion period was completed at the same position versus crank angle for both fuels. Çelik et al. [39] presented that maximum heat release rate was obtained at nearly the same crank angle position with the addition of n-heptan into diesel.

3.3. Ignition delay change versus engine load

As the engine load increases, the ignition delay is shortened. Minimum ignition delay was obtained about 0.73 ms and 0.86 ms for diesel and B30 respectively at full load. The ignition delay of B30 fuel is longer. Combustion of more fuel molecules lead to increase in-cylinder temperature and combustion improves. Thus, fuel evaporates earlier and the ignition delay decreases. Figure 7 shows ignition delay change versus engine load. Lower calorific value, high viscosity and density of biodiesel increased the ignition delay [40]. Çelik et al. [40] found that the shortest ignition delay was obtained with PH8. Uyaroğlu et al. [31] found the shortest ignition delay period with hazelnut oil biodiesel-diesel fuel blends at full load. In another study, Çelik et al. [41] investigated the effects of organic based manganese addition into diesel. They showed that the highest ignition delay was obtained with pure diesel. Besides ignition delay period shortened with the addition of manganese.

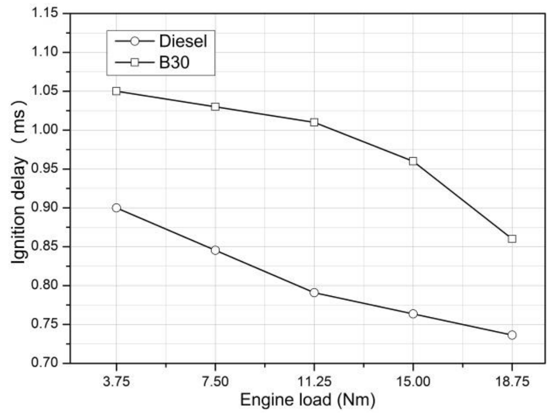


Figure 7. Ignition delay

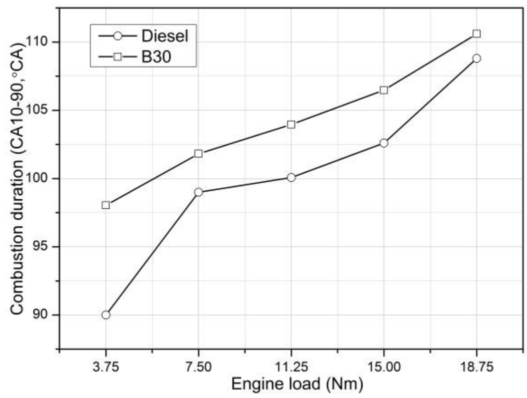


Figure 8. The variations of combustion duration

3.4. Combustion duration versus engine load

With the increase of the engine load for both fuels, the combustion duration increases. At 18.75 Nm engine load, combustion duration was determined about 108 °CA and 111 °CA with diesel and B30 respectively. At low engine speeds, the difference in combustion duration is high between the two fuels. When the engine load increases, the difference between the combustion duration of both fuels begins to decrease. The high density and viscosity of the B30 increases the combustion duration. The combustion duration is desired to be completed at the time of diffusion combustion period and at earlier crank angle. At the same engine torque value, the combustion duration of the B30 fuel is higher than that of diesel fuel. The change in combustion duration is shown in Figure 8. Higher density and viscosity of biodiesel prolonged the combustion duration, because physical delay period increased before the combustion. More time and sufficient temperature is needed in the combustion chamber in order to start combustion with biodiesel-diesel blends.

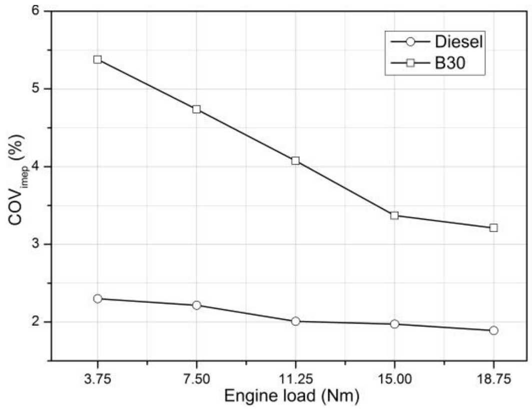


Figure 9. The cyclic variations depending on the engine load

3.5. Cyclic variations

It is observed that the cyclic variations are higher with biodiesel at the same engine load. However, as the engine load increases, the difference between the two fuels decreases. Variation of the mixture composition, the temperature and pressure differences in the cylinder cause cyclic variations. In the literature, the coefficient of cyclic variations (COV) should not exceed 10% for stable engine operation. Whole residual exhaust gas could not be discharged from the combustion chamber and they remain in the combustion chamber for the next cycle. So, In-cylinder charge composition varies at the end of compression stroke. This phonomena causes incomplete combustion and it results in cyclic variations. Figure 9 shows the cyclic variations depending on engine load. It can be said that more stable combustion was seen with diesel compared that B30. B30 showed resistance for physical and chemical decomposition before the combustion. This phonomena resulted in higher ignition delay period and combustion duration. Prolonged ignition delay caused to sudden combustion with B30 compared to diesel. So, unregulated combustion is seen with B30.

3.6. Variation of maximum pressure rise rate versus engine load

The difference between the pressure rise rates is higher at low engine loads. This difference decreases with the increase of engine load. It is seen that the pressure rise rates are close each other at 11.25 Nm engine load. Crank angle where the maximum in-cylinder pressure was obtained and pressure rise rate can be computed while 50 consecutive cycles are averaged versus crank angle. Knocking limit can be defined by pressure rise rate in the internal combustion engines for stable operation. Knocking is seen when the maximum pressure rise rate exceeds 10 bar/°CA in the internal combustion engines. The first derivative of in-cylinder pressure gives the pressure rise rate. The pressure rise rate is determined by taking numerical derivative according to the first degree of the cylinder pressure values versus crank angle. If the maximum pressure rise rate did not exceed 10 bar/°CA, no knocking was observed. Figure 10 shows the maximum pressure rise rates versus engine load.

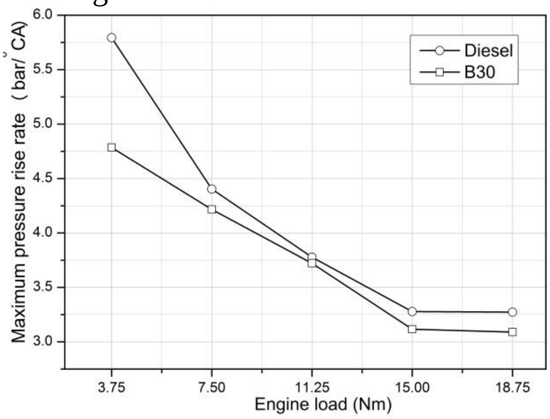


Figure 10. The change of maximum pressure rise rate

3.7. Cumulative heat release

The cumulative heat release is used to calculate the amount of mass burned versus the crank angle. Mass burned fraction can be determined by normalizing the heat release between 0-1300 J. The cumulative heat release obtained during the combustion period as a result of combustion gives us information about the percentage of burning fuel when normalized treatment between 0-1300 J. The crank angle, where the cumulative heat release reaches 10 % (130 J), is considered to be the crank angle where 10 % of the in-cylinder fuel amount is completed. The cumulative heat release reaches 90% (1170 J), which corresponds to the crank angle at which 90% of the fuel amount completes the combustion. Usually the time taken for combustion is defined as the time between 10% and 90% of the burning fuel in terms of the crank angle. As the crank angle increases, the cumulative heat release is increased parabolically. The cumulative heat release at the same engine speed and the same engine load at the same crank angle in the use of the B30 fuel shows a similar trend to diesel fuel. Figure 11 shows the cumulative heat release.

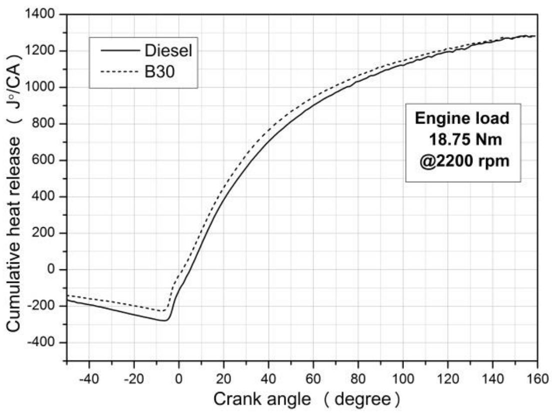


Figure 11. Cumulative heat release

3.8. The variation of CA50 versus engine load

CA50 is the crank angle where corresponds to 50% of the cumulative heat release. If CA50 is obtained far away from TDC, İndicated thermal efficiency decreases. Figure 12 shows that CA50 of the B30 fuel is higher than that of diesel fuel at specified engine load. This shows that the point where half of the mixture burns is longer than the TDC. This effect causes to decrease of thermal efficiency of B30.

3.9. The variation of indicated thermal efficiency versus engine load

Indicated thermal efficiency of diesel is higher than B30 fuel. However, while the indicated thermal efficiency of diesel fuel is 28% at 11.25 Nm engine load, the indicated thermal efficiency of B30 fuel is 26%. At the same engine loads, indicated thermal efficiency is better max 5% and min 1% at diesel fuel. Figure 13 shows the variations of indicated thermal efficiency versus engine load. Due to the high viscosities of fuels, it has more latent heat for evaporation and the indicated thermal efficiency is low. High viscosity and density cause the formation of larger droplets during the spraying of the fuel and the insufficient mixture with the air by evaporating [42]. Can et al. [32] showed that B20 presented similar thermal efficiency with diesel without EGR. It can be emphasized that more fuel fraction is needed for the same power output. So, brake specific fuel consumption increases and thermal efficiency decreases with the usage of biodiesel [11,12].

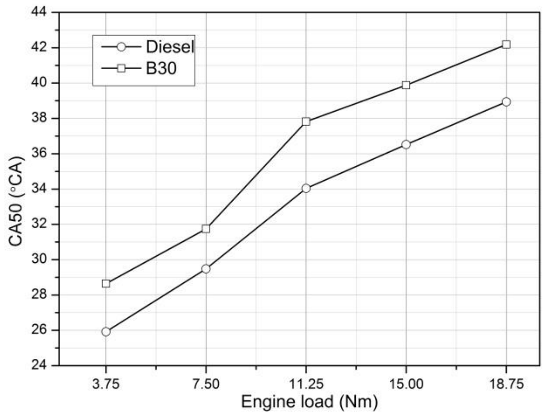


Figure 12. The variation of CA50

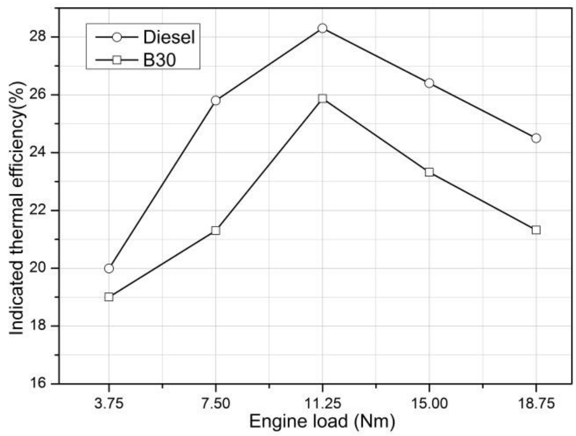


Figure 13. The variations of indicated thermal efficiency versus engine load

3.10. The variations of co and CO_2 versus engine load

CO is an incomplete combustion product due to insufficient temperature and oxygen content. In both diesel and B30 fuel use, CO emission rate increases depending on the engine load. Since the oxygen concentration decreases in the combustion chamber as the load increases, the formation of CO increases. However, this increase rate is higher in diesel fuel. The CO ratio is higher in diesel fuel than in biodiesel at 18.75 Nm, 3.75 Nm engine loads about 31 % and 12.5 %, respectively. The oxygen content in biodiesel mixtures reduces CO emissions by providing more efficient combustion in the region where combustion is rich [41,42]. CO_2 emissions also increase due to engine load in both fuels, but this increase is higher in diesel fuel than in B30 fuel. The B30 fuel contains oxygen, which improves the combustion process and improves the combustion quality. The CO_2 ratio of biodiesel is higher than the diesel about 41% and 20 % for the engine loads of 11.25 Nm and 3.75 Nm, respectively. The change in CO and CO_2 due to the motor load is shown in Figure 14. If the air-to-fuel ratio is controlled CO emissions may be reduced but the amount of CO_2 increases accordingly. The formation of CO_2 by reacting CO and oxygen depends on the combustion chamber temperature. Therefore, CO decreases with speed and CO_2 is increased [43]. Biodiesel has higher CO_2 emissions than diesel fuel due to the higher fuel consumption [44].

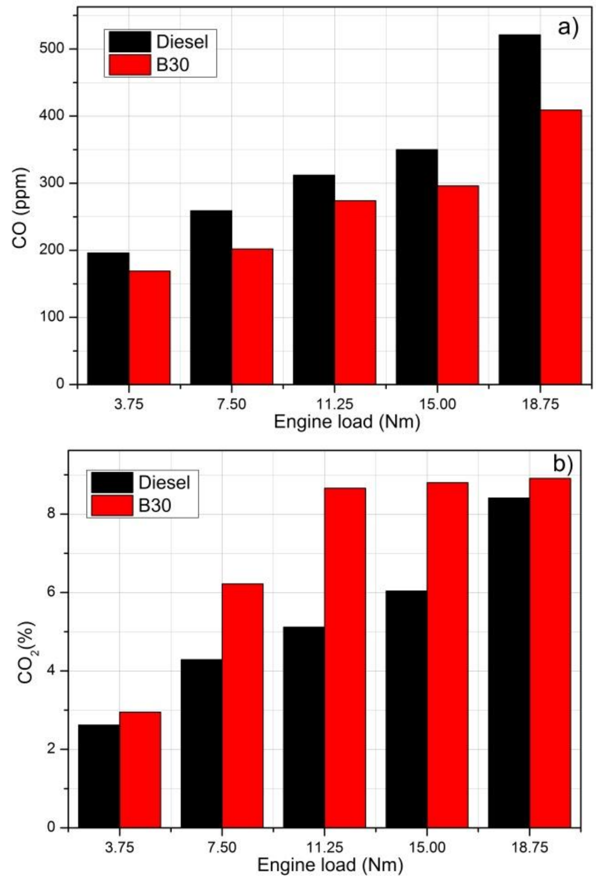


Figure 14. The variations of CO and CO_2 versus engine load

3.11. The variations of NOx versus engine load

NO_x emission for diesel and B30 fuel increase with the increase of engine load. However, NO_x emission in the same engine load is higher than the NO_x emission in diesel fuel when B30 fuel is used. The minimum NO_x difference is 0.75% at 11.25 Nm engine load and maximum 33% at 3.75 Nm. NO_x increases due to high combustion end gas temperatures. It is thought that nitrogen and oxygen molecules react and NO_x forms when the temperature is high during combustion. The engine load NO_x change is shown in Figure 15.

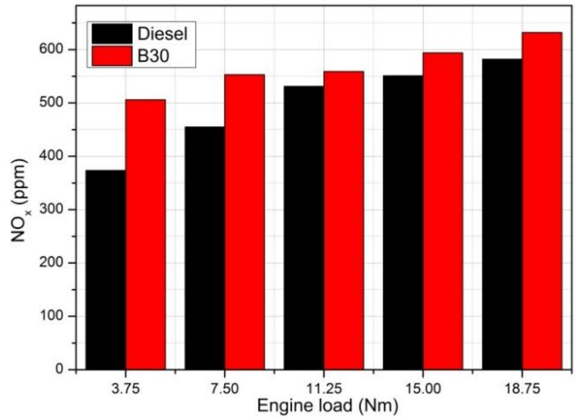


Figure 15. The variations of NO_x versus engine load

Depending on the oxygen content of biodiesel fuels, NO_x may show a tendency to increase according to diesel fuel values. Because oxygen increases the efficiency of the combustion and the combustion end gas temperature increases and can be effective in nitrogen gas to be converted into NO_x emissions [45]. The presence of 11% oxygen in biodiesel fuel improves the combustion performance and increases the combustion end temperatures, thus increasing the NO_x emission [45-46].

3.12. The variation of soot emissions versus engine load

Smoke emissions are increasing due to increase in engine load in both diesel and B30 fuels. Smoke emission rates for the same engine loads are higher in diesel fuel use. Oxygen content of waste olive biodiesel and low sulfur content cause smoke emissions to decrease. At full load, smoke emissions decreased by 37.5% in B30 fuel compared to diesel fuel. The change in smoke emissions is shown in Figure 16.

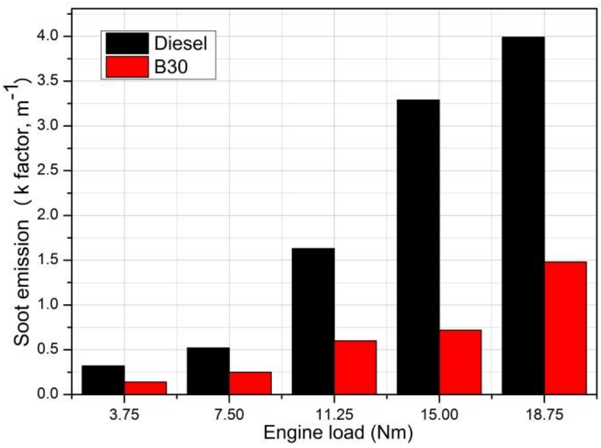


Figure 16. The variations of soot emissions versus engine load

The low sulfur content of the biodiesel, the presence of oxygen in the molecular content and the high air surplus coefficients increase the combustion efficiency, and the very small extent of SO_3 release is effective in the reduction of smoke emissions [44-46]. The presence of oxygen in the biodiesel and the low sulfur rate are effective in reducing the emission of smoke.

4. Conclusions

Detailed analysis should be required using purer biodiesel in view of performance, combustion and emissions in diesel engines. The properties of obtained biodiesel improved the injection process and combustion behaviour. So, detail investigation was aimed using waste oil biodiesel in order to determine performance, combustion and emissions in the present study. In this study, biodiesel was produced from waste olive oil by one-step transesterification method and B30 fuel was obtained by mixing 30% by volume of diesel fuel. B30 fuel was tested in a single-cylinder direct-injection diesel engine and engine performance and exhaust emission values were determined. Then, diesel fuel was tested in the same engine and combustion, engine performance and exhaust emission values were determined and the results obtained from both fuels were compared. In both fuel use, the heat release is similar in the same engine load at the same crank angle. Ignition delay is longer for B30 fuel than diesel fuel at the same engine loads. Ignition delay in diesel fuel is 0.725 ms at 18.75 Nm motor load and 0.875 ms for B30 fuel. At the same engine loads, the combustion time of the B30 fuel is longer than that of diesel fuel. As the engine load increases, the combustion times difference between diesel fuel and B30 fuel is reduced. The cyclic differences are greater in the use of B30 fuel for the same engine loads. The cyclical difference in diesel fuel at 18.75 Nm load is 2%, while in B30 fuel it is 3.25%. The cumulative heat release is similar in both fuels. Obtaining the CA50 value later than TDC for the B30 fuel decreases the indicated thermal efficiency. The thermal efficiency of diesel fuel is better than B30 fuel. However, while the thermal efficiency of diesel fuel is 28% at 11.25 Nm motor load, the thermal efficiency of B30 fuel is 26%. In the same engine loads, the thermal efficiency is better rate of max 5% and min 1% in diesel fuel. The CO ratio is higher in diesel fuel than in biodiesel for about 31% and 12.5 % at the load of 18.75 Nm and at 3.75 Nm, respectively. CO_2 emissions of B30 fuel are higher than diesel fuel. The CO_2 ratio of biodiesel is higher than the diesel for about 41% and 20% at the load of 11.25 Nm and 3.75 Nm, respectively. The NO_x emission rate is higher in B30 fuel than diesel fuel at the same engine loads. B30 fuel has a NO_x emission rate of more than 9.5% at 18.75 Nm engine load. The rate of smoke emissions is higher in diesel fuel. Smoke emissions decreased by 37.5% in B30 fuel compared to diesel fuel at 18.75 Nm load. As a result of the gradual decrease in petroleum-based fuels, price increases and the harmful effects on the environment, the search for a new fuel, which will be an alternative to these fuels, has become a popular subject. Biodiesel has become an important source of energy especially for the countries of agriculture because it is friendly to the environment and cheaper than that of diesel fuel. As a result, diesel biodiesel mixtures can be used in diesel engines as the nearest alternative fuel to diesel fuel even though they have lower efficiency than diesel fuel.

Abbreviations

ATDC After top dead center
BSFC Brake specific fuel consumption
CA Crank angle
CI Compression ignition
CO Carbon monoxide
CO_2 Carbon dioxide
COV Cyclic of variations

DC Direct current
NO_x Nitrogen oxides
OFR Oscillatory flow reactor
PM Particulate matter
SO_2 Sulphur dioxide
SO_3 Sulphur trioxide
TDC Top dead center
TDI Turbo direct injection
UHC Unburned hydrocarbon
dQ Heat release,
dQ_{heat} Heat transferred to cylinder walls,
dθ Crank angle variation
k Ratio of specific heat values

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