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# Experimental investigation on the combustion, performance and exhaust emission characteristics of poppy oil biodiesel-diesel dual fuel combustion in a CI engine

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ABSTRACT

In this study, a single cylinder, four-stroke, naturally aspirated with compression ratio of 18:1 direct injection diesel engine was run with opium poppy oil biodiesel-diesel fuel blends. The effects of diesel and biodiesel-diesel fuel blends were investigated experimentally on combustion, performance and emissions. Experiments were conducted with standard diesel fuel and opium poppy oil biodiesel-diesel fuel blends (OP10 and OP20) at maximum brake torque speed of 2200 rpm and five different engine load including 25%, 50%, 75% and 100%. This study focuses on the detailed performance and combustion analysis with opium poppy biodiesel under different engine load and speeds. Test results showed that in-cylinder presssure and heat release rate increased with the increase of engine load when biodiesel fuel blends were used. ID period increased with the usage of biodiesel. Thermal efficiency decreased by about 5.73% and 13.05% with OP10 and OP20 compared to diesel respectively owing to lower calorific value of opium poppy oil biodiesel at 75% engine load. $NO_x$ increased 2.9% and 5.98% with OP10 and OP20 according to diesel at full load. On the contrary, CO decreased 14% and 17.42% with OP10 and OP20 compared to diesel at full load.

## 1. Introductıon

Diesel engines are widely used due to higher thermal efficiency. However, NO, CO and soot emissions produced from diesel engines severely damages environment, human health and atmosphere. At this point, the usage of biodiesel in diesel engines is attracted by researchers. Biodiesel provides not only reasonable power output but also helps to reduce exhaust emissions significantly compared to diesel fuel [1–4].

Researchers are intensifying their work on alternative energy sources to decrease the dependency of petroleum-based fuels and the harmful exhaust gases caused by increasing of the number of vehicles in the World. Diesel engines are widely used in heavy industrial applications and transportation due to the higher compression ratio and thermal efficiency than spark plug ignited engines. However, NO and smoke emissions of diesel engines threaten the enviroment and human health. Although particle filter, EGR and SCR are used to decrease the harmful emissions, they do not decrease the need of ecofriendly fuels due to their high production costs and yield losses [5–11,52]. The usage of biodiesel fuels have important potential to solve these problems in diesel engines. As it is known, biodiesel has higher cetane number than diesel fuel. High cetane number increase the quality of combustion by decreasing the ignition time in the compression ignited engines [12–14]. Besides, the content of oxygen in the chemical structure and the lower value of sulfur in biodiesel decrease the exhaust emissions in diesel engines [14–17]. But the content of oxygen in biodiesel fuels decreases the thermal energy of the fuel. At the same time, there might be some problems in injection systems and injectors due to the high viscosity and denstiy of biodiesel fuel. Especially, occlusion and corruption in fuel injection systems can occur in cold working conditions.

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**Nomenclature**

| | |
|---|---|
| ABDC | After bottom dead center |
| AME | Annona methyl ester |
| ATDC | After top dead center |
| BBDC | Before bottom dead center |
| BMEP | Break mean effective pressure |
| BSFC | Brake specific fuel consumption |
| BTDC | Before top dead center |
| BTE | Break thermal efficiency |
| CA | Crank angle |
| CO | Carbon monoxide |
| $CO_2$ | Carbon dioxide |
| COME | Cottonseed oil methyl ester |
| DI | Direct injection |
| EGR | Exhaust Gas Recirculation |
| EVC | Exhaust valve closing |
| EVO | Exhaust valve opening |
| FBP | Final boiling point |
| HC | Hydrocarbon |
| ID | Ignition delay |
| IBP | Initial boiling point |
| IVC | Intake valve closing |
| IVO | Intake valve opening |
| ITE | Indicated thermal efficiency |
| MPRR | Maximum pressure rise rate |
| NO | Nitrogen oxide |
| OP | Opium Poppy |
| $P_{max}$ | Maximum in-cylinder pressure |
| R | Universal gas constant |
| RI | Ringing intensity |
| ROME | Rapseed oil methyl ester |
| SFC | Specific fuel consumption |
| SCR | Selective Catalytic Reduction |
| $T_{max}$ | Maximum in-cylinder temperature |
| $\gamma$ | Politropic index |
| $dQ$ | Heat release rate |
| $dQ_{heat}$ | Heat transfer to the cylinder wall |
| $d\theta$ | The variation of crank angle |
| $\frac{dP}{dt}$ | Maximum pressure rise rate |

The quality of combustion decreases with unfulfilled vaporization in combustion chamber as a result of getting worse of the atomization properties of the fuel due to the high viscosity and density of biodiesel fuel. Among the biofuels, opium poppy seems to be useful and reasonable as alternative fuel in diesel engines. Opium poppy oil is industry plant and one-year culture plant. The alkaloids contained in the opium poppy capsules are utilized. It highly contains oil. Opium poppy production is currently carried out in Turkey as a result of provisions of laws and regulations. Produced opium poppy was processed in the factories. Processed opium poppy is exported to the needy countries. The majority of produced opium poppy have been used for medical purposes. Uyumaz et al. [6], tested waste tyre oil biodiesel-diesel fuel blends (B10) and neat diesel in a four stroke direct injection diesel engine. Authors found that SFC of B10 fuel was higher than neat diesel about 20% at 3.75 Nm and 7.5 Nm load condition. In addition, maximum smoke were obtain at full load conditions. Also, CO emissions decreased with the usage of B10 fuel. Labeckas and Slavinskas [18] investigated the engine performance and exhaust emissions of canola biodiesel-diesel fuel blends. Authors reported that SFC increased about 18.7–23.2% due to low thermal energy of biodiesel fuel. They also found that NO production of biodiesel-diesel fuel blends increased the production of NO emissions. Usta et al. [19] investigated the effects of biodiesel produced from hazelnut and waste sunflower oil mixture and diesel fuel blends on engine performance and exhaust emissions in an indirect injected, four stroke diesel engine. It was reported that biodiesel fuel blends can be used without any modification and pre-heating in diesel engines. Özsezen et al. [20] tested the raw sunflower oil in an indirect injected and four stroke diesel engine. Authors found that brake torque and power decreased 1.36% and 1.35%, respectively with the usage of raw sunflower oil. On the other hand, SFC increased 4.98% at full load conditions. At the same time, HC, $CO_2$ and smoke opacity decreased 33.6, 2.05 and 4.52%, respectively, with the usage of sunflower oil. Can [21] tested waste fried oil biodiesel-diesel fuel blends (B5 and B10) in one cylinder, four stroke, direct injection diesel engine at four different engine loads (0.48–0.36–0.24–0.12 MPa). Author specified that ignition delay (ID) decreased and the combustion was advanced. Author also detected that SFC increased and thermal efficiency decreased by B5 and B10 fuel blends. Xue [22] searched the effects of biodiesel produced from waste renewable oil on combustion, engine performance and exhaust emissions. Author found very small differences on combustion characteristics like ingition delay, pressure increase rate, maximum pressure and heat release. Aksoy [23] blended diesel fuel and biodiesel fuel produced from opium poppy oil 50% by volume and investigated engine performance and exhaust emissions. Authors' experiments showed that engine torque and power decreased 4 and 5.73%, respectively. Author reported that CO and NO emissions decreased by 50% opium poppy oil biodiesel and 50% diesel fuel blend. Altın et al. [24] searched engine performance of opium oil and different oils and bodiesel fuels and the blends with diesel fuel. Authors found that engine torque and power decreased and SFC increased by opium poppy oil biodiesel. Beside this, CO and NO emissions decreased wtih the usage of opium poppy oil biodiesel. Aksoy [25] aimed to investigate the biodiesel evaluation of the opium poppy seeds. Author preferred alkali catalyzed (NaOH) single-phase reaction to produce biodiesel. Author determined the methanol ratio and catalyst concentration as 20 wt% and 0.5 wt%, respectively. Author also determined the some properties of opium poppy oil biodiesel. Maximum in-cylinder pressure increased with the addition of biodiesel in the fuel mixtures at full load. Moreover, maximum heat release has been obtained with JP-8 (J100) test fuel for all engine loads due to close calorific value of JP-8 aviation fuel [26–29]. In-cylinder pressure increased with waste cooking oil biodiesel-diesel fuel blends (B20, B50, B75) and shorter ID and combustion duration were determined [30]. On the other hand, lower peak heat release was obtained with biodiesel due to higher ignition delay (ID) and worse fuel evaporation especially at low engine loads [31]. Senthil et al. [32] found that heat release rate increased with neat diesel compared to annona methyl ester (AME) due to longer ID. Shi et al. [33] showed that in-cylinder pressure increased with the increase of engine load. Moreover, slight decrease was seen on cylinder pressure with biodiesel-diesel fuel blend (B20) compared to B10 at constant EGR rate. Liu et al. [34] showed slight increase was seen on heat reelase with biodiesel and polyoxymethylene dimethyl ethers-biodiesel fuel blend (B85P15) than diesel in premixed combustion stage, because oxygen molecules improved the oxidation reactions. Combustion characteristics were improved with rice bran oil methyl esters (ROME) compared to alga extracted oil methyl esters (AME). At full load maximum in-cylinder pressure increased with ROME and AME compared that neat diesel [35]. Waste fish oil biodiesel presented higher in-cylinder pressure with shorter heat release duration compared to diesel [36]. Higher in-cylinder pressure was obtained with castor biodiesel-diesel fuel blends compared to diesel for all engine loads [37]. Combustion duration decreased with the addition of mustard biodiesel into the diesel fuel compared to diesel fuel owing to higher oxygen content of biodiesel [38]. Gnanasekaran et al. [39] implied that combustion duration and heat release rate increased with the advancing of injection timing with fish oil biodiesel. Altın et al. [40] found that engine torque decreased by about 10% with raw sunflower seed oil, raw soybean oil, and opium poppy oil at 1300 rpm engine speed compared to neat diesel. Tosun et al. [41] implied that brake torque and power output decreased 7.02% and 7.95% with opium poppy oil biodiesel compared to diesel respectively.

In this study, the effects of opium poppy oil biodiesel-diesel fuel blends (OP10 and OP20) on combustion characteristics in a single cylinder, four-stroke, naturally aspirated direct injection diesel engine at maximum torque speed (2200 rpm) and loads of 25%, 50%, 75% and 100% were investigated. Opium poppy oil that has been produced great quantities in Turkey. In addition, it has great potential due to higher oil content for using as alternative fuel in diesel engine. However, there are limited experimental researches in view of detailed combustion, performance and emission analysis with opium poppy oil biodiesel. So, the determination of the effects of opium poppy oil biodiesel on combustion, performance and exhaust emissions is worthful. The variation of in-cylinder pressure, heat release rate, ID, combustion duration and ringing intensity (RI) with opium poppy oil biodiesel-diesel fuel blends were researched and compared with neat diesel fuel.

## 2. Experımental setup and procedures

The experiments were performed with neat diesel, OP10 and OP20 fuels at 2200 rpm engine speed and 25%, 50%, 75% and 100% engine loads. The engine was heated up to operating temperature to obtain stable conditions during the experiments. Technical properties of the one cylinder diesel engine used in experiments are given in Table 1. Experimental setup is shown in Fig. 1. The test engine was conducted to Cussons P8601 DC dinamometer which can absorb 10 kW power at 4000 rpm. The engine speed and engine power can be controlled with control panel of the dinamometer. The tests were performed at constant oil temperature of 85 °C in order to prevent cyclic variations. Before each experiments, the test engine was heated up with neat diesel fuel. Table 2 shows the test fuels and abbreviations. The properties of the test fuels are seen in Table 3.

A program which was written in Matlab was used to specify the combustion analysis. Heat release rate, combustion phasing and RI were obtained with the handled in-cylinder pressure. Heat release rate was calculated with equation (1) below which was gained from the first law of thermodynamic by assuming the gas inside the combustion chamber is an ideal gas and omitting the cylinder gas leackage [42–44].

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

Here, $dQ$, $dQ_{heat}$ and $d\theta$ represent the heat release rate, the heat transfer from combustion chamber to cylinder walls and crank angle, respectively. Cylinder pressure and temperature rapidly increase during compression stroke. Whole charge mixture tends to combust suddenly at high in-cylinder pressure and temperature. So, in-cylinder pressure rise rate increases for each crank angle degree. It results in knocking tendency. The life of engine shortens as a result of knocking. The calculation of ringing intensity is a paramater to establish engine knocking. RI was calculated by the equation (2) as below [42–47].

$$RI = \frac{1}{2\gamma}\frac{\left(\beta\left(\frac{dP}{dt}\right)_{\max}\right)^2}{P_{\max}}\sqrt{\gamma.\ R.\ T_{\max}} \tag{2}$$

$\gamma$, $P_{\max}$, $T_{\max}$ and $\frac{dP}{dt}$ define the politropic index, maximum in-cylinder pressure, maximum in-cylinder temperature and maximum pressure rise rate.

The variations of exhaust emissions such as $NO_x$, CO and $CO_2$ were measured using Testo exhaust gas analyzer. The technical specifications of the Testo exhaust gas analyzer are seen in Table 4.

Soot emissions were measured with AVL Di-Smoke 4000 smoke meter which of the technical properties are seen in Table 5.

## 3. Results and dıscussıon

The main objective of this study is to determine the effects of opium poppy oil biodiesel on combustion, engine performance and exhaust emissions in a diesel engine. For this purpose, the test engine was operated at different engine loads including 25%, 50%, 75% and 100%. Moreover, full load speed characteristics were determined between 1750 and 3000 rpm engine speed operating range in a diesel engine fueled with opium poppy oil biodiesel-diesel fuel blends. So, the effects of opium poppy oil biodiesel on cylinder pressure and heat release rate at different engine loads are shown in Fig. 2. The test engine delivers maximum brake torque at 2200 rpm. It can be clearly implied that in-cylinder pressure increased with the usage of biodiesel. Furthermore, in-cylinder pressure increased with the increase of engine load. Maximum in-cylinder pressure was measured slightly after top dead center 10 °CA with biodiesel fuel blends. It can be also mentioned that maximum in-cylinder pressure was determined later with biodiesel fuel blends according to D100. The higher viscosity and density of biodiesel resulted in later maximum in-cylinder pressure compared to D100. Although biodiesel has lower calorific value compared to D100, higher in-cylinder pressure was determined with biodiesel fuel blends. Oxygen molecules improves the oxidation reactions resulting in higher in-cylinder pressure. It was also seen that earlier heat release rate was determined with OP10 and OP20 according to D100. It can be found that higher oxygen concentration leads to earlier combustion reactions. This situation causes to release heat earlier during combustion. Moreover, higher in-cylinder prassure was obtained due higher particle size of injected biodiesel. Because, injected biodiesel by mass increases due to higher density of opium poppy biodiesel. Thereby, the addition of more biodiesel into pure diesel was not preferred owing to poor injection characteristics of biodiesel in the current study. Since the percentage of biodiesel increased in the fuel mixtures, droplet size of injected biodiesel increased. Consequently, fuel consumption increases and atomization of biodiesel was deteriorated.

Combustion stages can be determined via normalization of cumulative heat release between 0 and 1. Thus, CA10, CA50 and CA90 can be computed. In addition, combustion duration (CA10-90) can also be determined. CA10 and CA90 refer to crank angle where the 10% and 90% of charge mixture completed to combust respectively. In the current study, combustion duration was computed between CA10 and CA90 as crank angle. ID is defined depending on crank angle where the first flame kernel is formed during combustion. ID was determined between start of injection and the positive value of heat release rate depending on crank angle. ID is evident for compression ignition because whole combustion is affected by ID [12]. Fig. 3 shows the ID versus engine load with test fuels. It was seen that ID decreased with the increase of engine load. Higher fuel concentration in the combustion chamber resulted in higher in-cylinder temperature. This phonomena

**Table 1**
The technical specifications of the test engine.

| | | |
|---|---|---|
| Model | | Antor/6LD400 |
| Engine type | | DI-Diesel engine, natural aspirated, air cooled |
| Cylinder number | | 1 |
| Bore × stroke [mm] | | 86 × 68 |
| Displacement [cm3] | | 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] × 4 holes × 160° |
| Nozzle opening pressure [bar] | | 180 |
| Fuel delivery advance angle [°KA] | | 24 BTDC |
| Valve timings | IVO/IVC [°KA] | 7.5 BTDC/25.5 ABDC |
| | EVO/EVC [°KA] | 21 BBDC/3 ATDC |

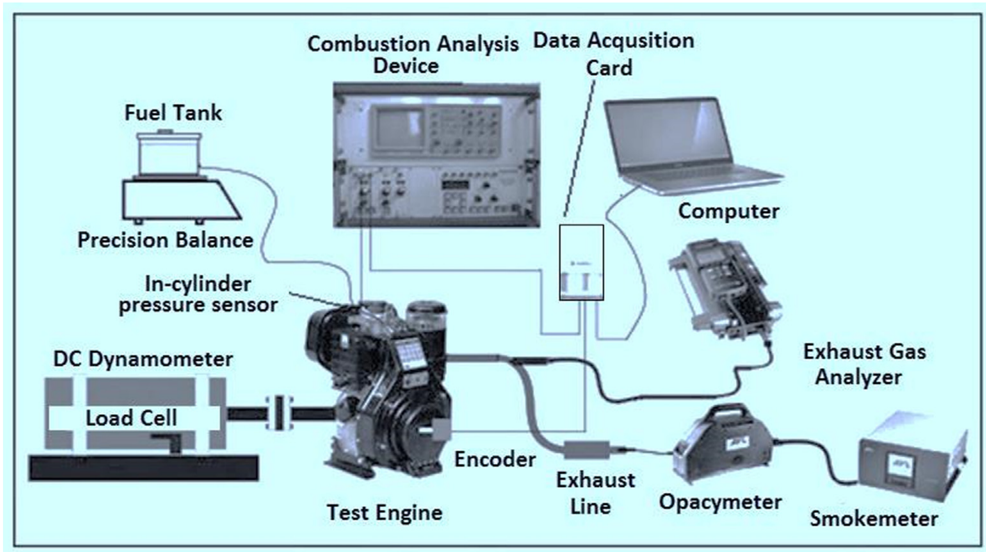


**Fig. 1.** Experimental setup.

**Table 2**
Test fuels and abbreviations.

| Abbreviation | Fuel mixtures |
|---|---|
| D100 | 100% diesel |
| OP10 | %10 Opium Poppy Oil Biodiesel + 90% diesel |
| OP20 | %20 Opium Poppy Oil Biodiesel + 80% diesel |

**Table 3**
Properties of the test fuels.

| Property | Diesel fuel | Opium poppy biodiesel fuel |
|---|---|---|
| Density @ 15 °C ($g/cm^3$) | 0.830 | 0.890 |
| Water content (ppm) | 35.45 | 75.89 |
| Lower heating value (MJ/kg) | 42.6 | 40.187 |
| Cetane index | 56.80 | 54.60 |
| Kinematic viscosity @ 40 ($mm^2/s$) | 2.89 | 4.682 |
| Flash point (°C) | 67 | 151 |

**Table 4**
The technical specifications of the Testo exhaust gas analyzer.

| Combustion products | Operating range | Accuracy ( ± digit) |
|---|---|---|
| $O_2$ | 0–25 [vol.%] | ± 0.8% |
| CO | 0–10000 [ppm] | ± 5% of mv [100–2000 ppm]<br>± 10% of mv [2001–10000 ppm]<br>± 10 ppm [0–99 ppm] |
| $CO_2$ | 0–50 [vol.%] | Calculated from $O_2$ |
| $NO_x$ | 0–3000[ppm] | ± 5% of mv [100–1999.9 ppm]<br>± 10% of mv [2000–3000 ppm]<br>± 5 ppm [0–99 ppm] |

**Table 5**
The technical properties of the AVL Di-Smoke 4000 smoke meter.

| Analyzer | AVL Di-Smoke 4000 | |
|---|---|---|
| Measurement method | Partial flow | |
| | Opacity | k value |
| Operating range | 0–100% | 0.1% |
| Accuracy [$m^{-1}$] | 0–99,99 | 0,01 |

causes to shorten ID, because fuel could be easily ignited at higher engine loads owing to higher in-cylinder temperature. Minimum ID was determined at 100% engine load at 2200 rpm. Moreover, the usage of biodiesel caused to increase ID. Higher viscosity and density of biodiesel deteriorated the atomization of the biodiesel fuel blends. The particle size of fuel also increases with the usage of biodiesel. Hence, fuel can not be properly atomized and vaporized. So, ID increased with biodiesel fuel blends. ID is affected by cetane number. When the amount of sunflower oil biodiesel increased in the fuel blends, ID decreased with the increase of engine load due to higher cetane number and viscosity of biodiesel [26]. Similar results were obtained ID increased with neutralized waste cooking oil biodiesel-diesel fuel blend (B10), mustard oil biodiesel and castor biodiesel fuel blends compared to diesel due to higher density and viscosity of biodiesel.

Fig. 4 shows the combustion duration at different engine loads with test fuels. Since more charge mixture is taken to the cylinder, more time is required in order to complete combustion at higher engine loads. So, it is clear to mention that combustion duration increased with the increase of engine load. Maximum combustion duration was determined at 100% engine load. As the biodiesel fraction increases in the fuel mixture, combustion is completed earlier. The longest combustion duration was computed with D100. The reason of this situation is that oxidation reactions improve with the addition of biodiesel due to more oxygen concentration of biodiesel. Thus, fuel molecules can be easily oxidized and in-cylinder temperature increases towards the combustion chamber. So, combustion duration decreased with the addition of biodiesel in the fuel mixture.

The variation of RI versus maximum pressure rise rate (MPRR) is seen in Fig. 5. RI is affected by engine speed and MPRR. It can be concluded from Fig. 5 that RI increased as the MPRR increased. It was found that D100 presented the highest RI. Minimum RI was computed with OP10. Apart from that, ignition conditions improve because, biodiesel fuel blends consist of more oxygen especially with OP20. Maximum RI was computed as 9.44 $MW/m^2$ with D100 at 100% engine load.

Fig. 6 depicts the indicated mean effective pressure (imep) versus consecutive 50 cycles at 100% engine load. Imep is defined the averaged in-cylinder pressure exerted to the piston during a cycle [51]. The highest imep values were determined with D100 at constant engine speed of 2200 rpm. It can be said that the lowest imep values were determined with OP20. Imep is highly affected by heating value of fuel. As it is known, calorific value of biodiesel is lower than diesel. So, the force exerted to the piston decreased as a result of combustion of

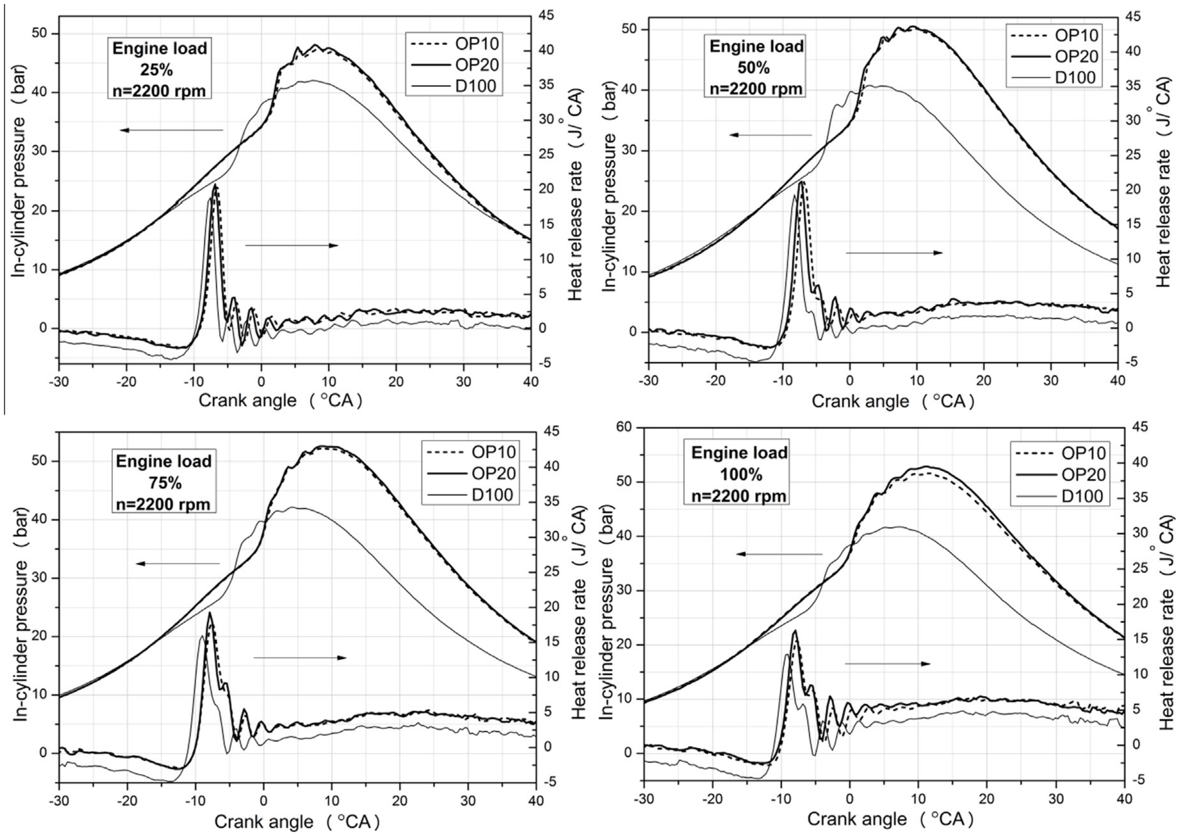


**Fig. 2.** The effects of opium poppy oil biodiesel on cylinder pressure and heat release rate.

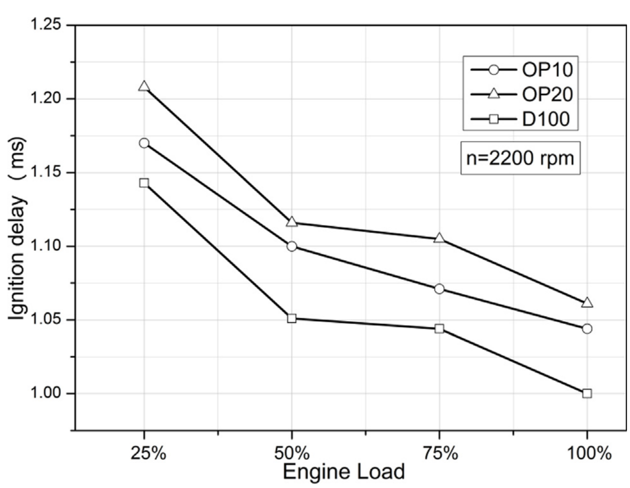


**Fig. 3.** ID versus engine load.

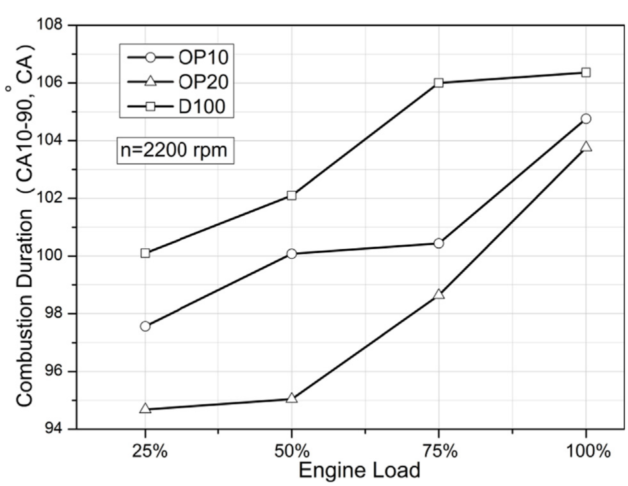


**Fig. 4.** Combustion duration.

biodiesel fuel blends. Hence, imep decreased. The heating value of test fuel decreased when the biodiesel was added into the fuel mixture. Consequently, lower imep was determined with biodiesel-diesel fuel blends. Similar results were obtained with mustard oil biodiesel. Imep values decreased with the addition of mustard oil biodiesel because of the lower calorific value of biodiesel.

In the current study, experiments were also conducted at different engine speeds at 100% engine load. The effects of engine speed on engine performance and exhaust emissions were determined. Maximum brake torque was measured at 2200 rpm engine speed for all test fuels. As it is seen from Fig. 7 that brake torque decreased with the increase of engine speed. Gas leakages and heat losses increase at higher engine speed. Moreover, volumetric efficiency also decreases, because there is less time in order to deliver charge mixture into the cylinder depending on crank angle. So, brake torque decreased with the incrase of engine speed. Fig. 7 also explains that brake torque decreases with the addition biodiesel in the fuel mixture. The lowest brake torque was measured with OP20. Similar results were obtained with Fig. 7. Aksoy [23] determined that lower brake torque was obtained with 50% opium poppy oil – 50% diesel fuel mixture compared that neat diesel fuel.

Maximum brake torque was measured with D100 as 18.02 Nm at 2200 rpm. But brake torque decreased 11.09% at 3000 rpm with D100. Accordingly, brake torque decreased 16.1% and 15.06% with OP10 and OP20 at 3000 rpm compared to 2200 rpm respectively. Fig. 8 defines the maximum in-cylinder pressure versus engine speed at 100% engine load. Maximum in-cylinder pressure was obtained at 2200 rpm engine speed for all test fuels, because the most charge mixture was delivered to the cylinder in a cycle at 2200 rpm. After the 2200 rpm engine speed,

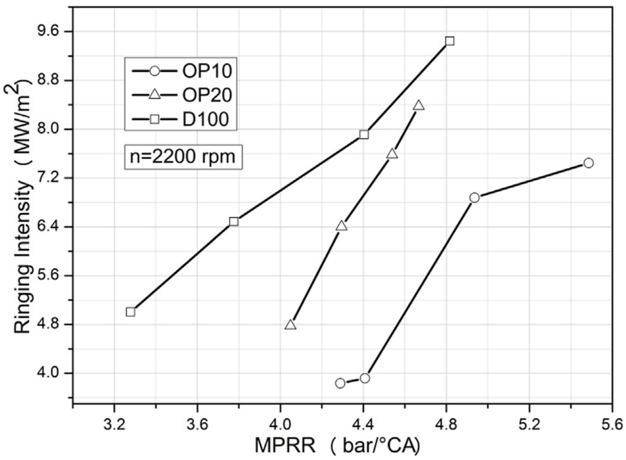


**Fig. 5.** The variation of ringing intensity versus MPRR.

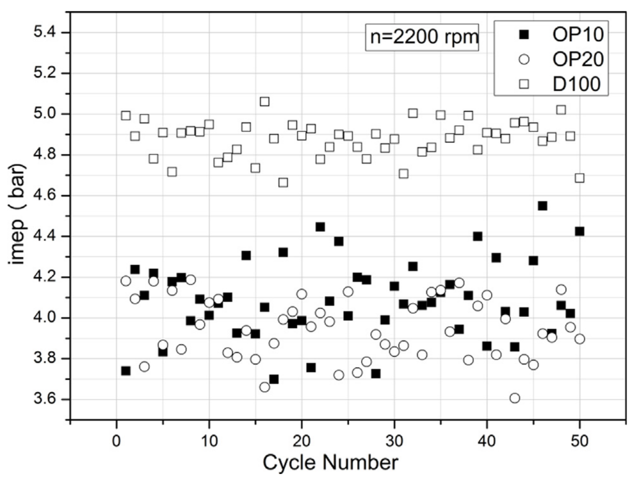


**Fig. 6.** Imep versus consecutive 50 cycles.

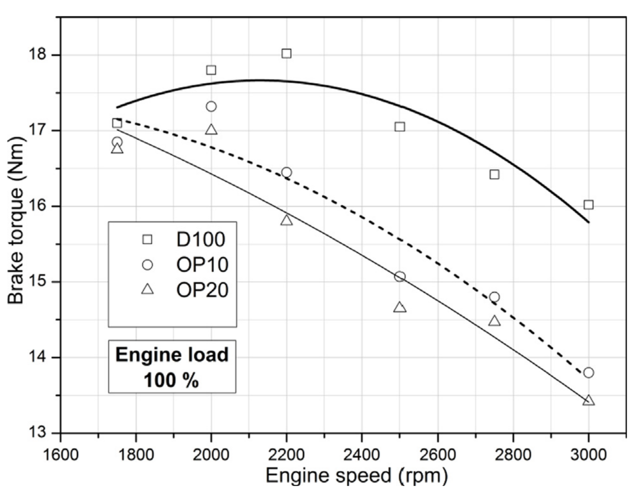


**Fig. 7.** Brake torque versus engine speed.

maximum in-cylinder pressure started to decrease due to more heat and flow losses.

Oxidation reactions improve with the usage of biodiesel. So, maximum in-cylinder pressure was obtained with OP20. Maximum in-cylinder pressures were obtained as 53.41 bar, 50.23 bar and 40.93 bar with OP20, OP10 and D100 respectively at 2200 rpm and 100% engine load.

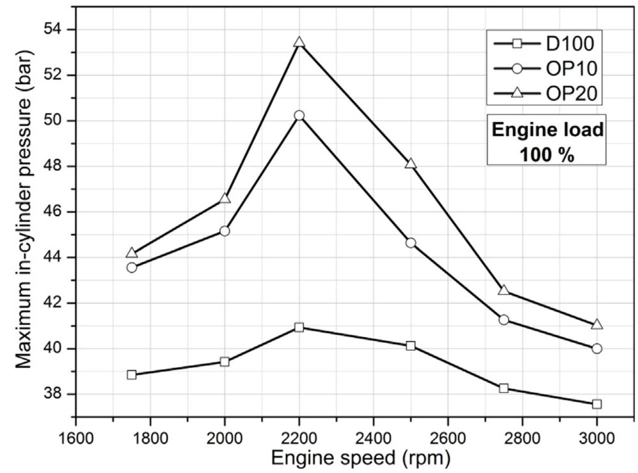


**Fig. 8.** Maximum in-cylinder pressure versus engine speed.

Fig. 9 shows the maximum in-cylinder temperature versus engine load. As engine load increased, more fuel molecules participated to the chemical oxidation reactions. So, maximum in-cylinder temperature increased with the increase of engine load. It was also found that the addition of biodiesel caused to obtain higher in-cylinder temperature as seen in Fig. 9. Longer ID and higher density of biodiesel lead to combust more fuel immediately during combustion. Higher in-cylinder temperature is obtained when the more biodiesel fuel blends are ignited. It can be also explained that higher oxygen content of biodiesel caused to more qualified combustion. Thus, in-cylinder temperature increased with biodiesel fuel blends. Maximum in-cylinder temperatures were obtained as 1236.62 K, 1170.51 K and 1081.85 K with OP20, OP10 and D100 respectively at 100% engine load. The increase of biodiesel fraction resulted in higher in-cylinder temperature owing to higher oxygen content, because the end of combustion gas temperature increases in the combustion chamber [30].

Fig. 10 shows the variations of indicated thermal efficiency (ITE) and SFC versus engine load with test fuels. ITE increased until 75% engine load for each test fuel. After 75% engine load, ITE decreased due to deterioration of combustion. Oxygen concentration decreased in the combustion chamber resulting in incomplete combustion at the highest engine load. This phonomena caused to decrease ITE. When Fig. 11-a was examined, maximum ITE was computed with D100 due to higher calorific value of diesel. ITE decreased with the adition of biodiesel. The lowest ITE was determined with OP20. ITE decreased by about 5.72% and 13.03% with OP10 and OP20 compared that D100 at 75% engine load. Higher brake thermal efficiency (BTE) was computed as 42.17%

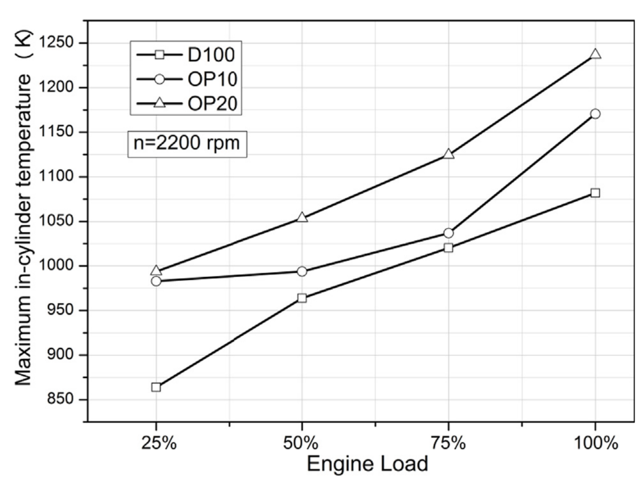


**Fig. 9.** Maximum in-cylinder temperatures.

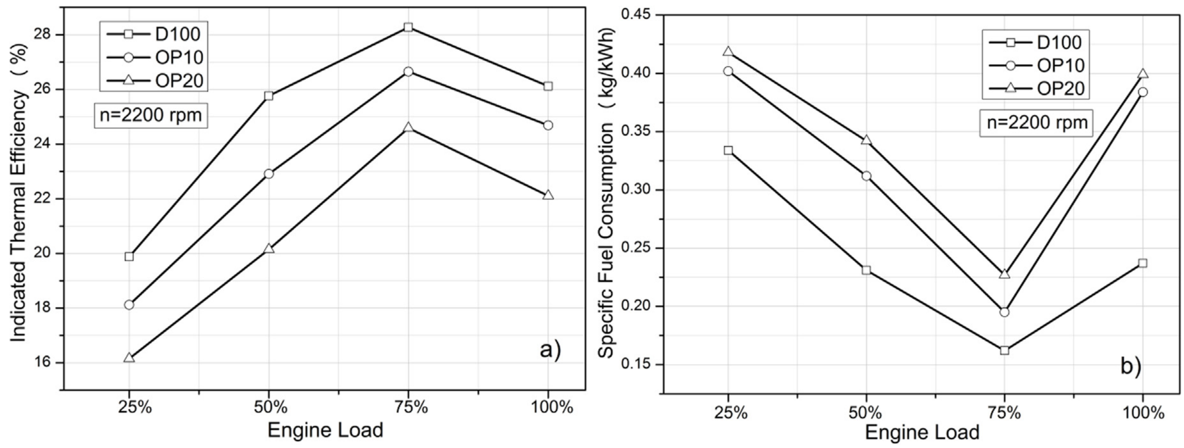


**Fig. 10.** a-ITE versus engine load b-SFC versus engine load.

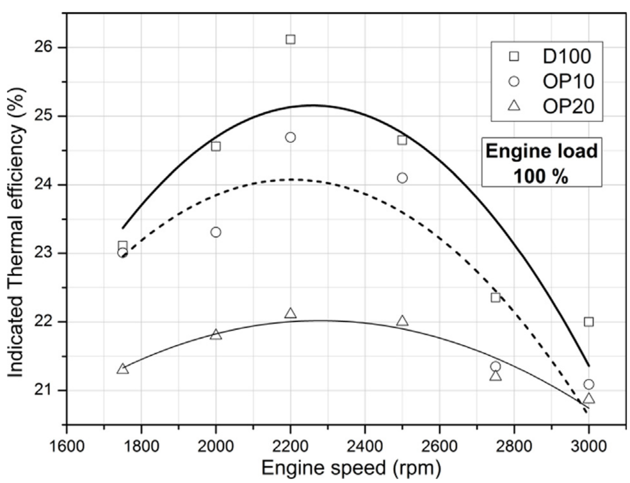


**Fig. 11.** ITE versus engine speed.

with B15 biodiesel-diesel fuel blend whereas lower SFC was obtained as 0.2 kg/kWh at 4.64 BMEP.

Similar results were obtained with sunflower oil biodiesel-diesel fuel blends at different engine loads. Similar results were obtained with opium poppy oil biodiesel, waste cooking oil biodiesel-butylated hydroxytoluene blend and mustard oil biodiesel [26,38,41,48].

Fig. 10-b defines the variation of SFC depending on engine load. SFC decreased with the increase of engine load until 75% engine load. At 100% engine load, SFC increased due to partial combustion. Firstly, there was a good agreement between ITE and SFC as seen in Fig. 10-a and -b. The lowest SFC was computed at 75% engine load where the highest ITE was determined for all test fuels. SFC increased by about 20.37% and 40.12% with OP10 and OP20 compared to D100 at 75% engine load. Fig. 11 shows the ITE versus engine speed at 100% engine load. As expected, the highest ITE was computed with D100 at 2200 rpm. ITE first started to increase with the increase of engine speed and then started to decrease depending on engine speed. At lower engine speeds, less charge mixture is ignited resulting in lower heat release. In addition, heat is transferred to the cylinder wall at lower engine speeds, because there is sufficient time in order to transfer heat. Heat losses decrease for each cycle with the increase of engine speed. So, ITE increases. However, flow and frictional losses increase at higher engine speeds resulting in lower ITE.

Fig. 12 shows the CO and $CO_2$ emissions versus engine load at 2200 rpm. CO is a incomplete combustion product. So the determinaton of CO is evident for biodiesel fuel blends. As seen in Fig. 12-a, CO increased with the increase of engine load. The increase of engine load caused to decrease oxygen concentration in the combustion chamber. Fuel molecules could not be oxidized due to lack of oxygen at higher engine loads. Consequently, CO is formed. It was also determined that CO reduced when the biodiesel fraction increased in the fuel mixture for each engine load. The main reason of this situation is that biodiesel has higher oxygen in its chemical stucture compared to diesel.

CO reduced by about 14% and 17.42% with OP10 and OP20 according to D100 at 100% engine load. On the other hand, $CO_2$ inreased when more biodiesel was added to the fuel mixture. Maximum $CO_2$ was measured with OP20 for all engine loads. It can be also implied that $CO_2$ increased with the increase of engine load. $CO_2$ increased 47.51%

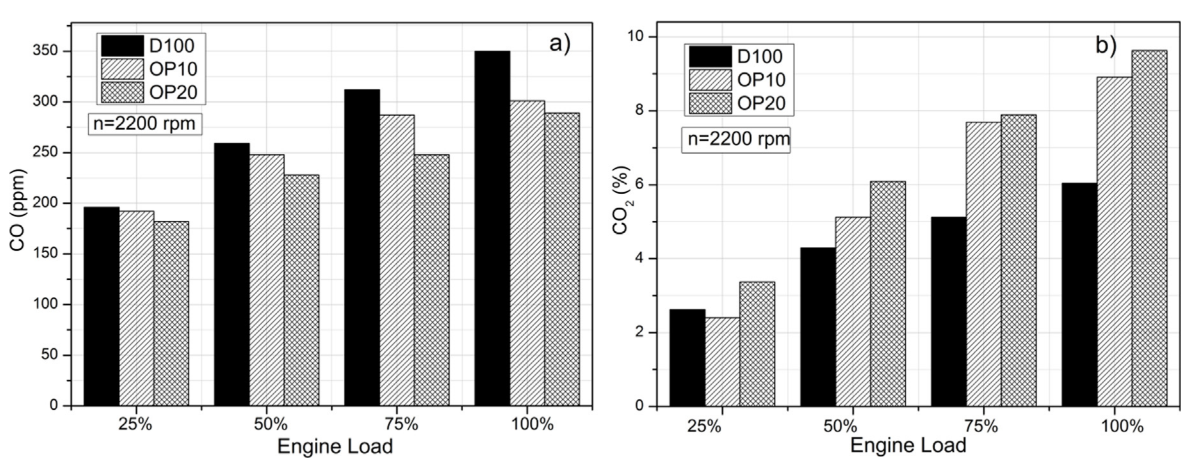


**Fig. 12.** CO and $CO_2$ emissions versus engine load.

and 59.43% with OP10 and OP20 compared that D100 at 100% engine load.

Fig. 13 defines $NO_x$ and soot emissions versus engine load. Oxygen and nitrogen molecules can react at higher temperatures resulting in $NO_x$ emissions. $NO_x$ which is one of the most harmful exhaust emissions in diesel engines that causes acid rain. So, reducing $NO_x$ is worth in diesel engines. When Fig. 13-a is examined, $NO_x$ increases with the increase of engine load. The amount of molecules involved in the reaction is increasing at higher engine loads. This causes to release more heat energy and higher in-cylinder temperature is obtained. Since higher in-cylinder temperature was obtained due to combustion of more charge mixture, oxygen and nitrogen molecules can react. Hence, $NO_x$ is formed. Besides, $NO_x$ increased with the increase of biodiesel fraction in the fuel mixture. The highest $NO_x$ was measured with OP20 for all engine load. Higher oxygen content of opium poppy oil biodiesel caused to react oxygen and nitrogen molecules at higher in-cylinder temperatures. Another harmful exhaust emission for diesel engine is soot emissions. $NO_x$ increased 2.9% and 5.98% with OP10 and OP20 according to D100 at 100% engine load. Inhomogeneity of charge mixture, rich mixtures, fuel injection problems lead to release soot emissions in diesel engines.

Soot increased with the increase of engine load for each test fuel as seen in Fig. 13-b. In-cylinder temperature is lower at lower engine loads due to less charge mixture. Fuel concentration in the combustion chamber increases at higher engine load. This deteriorates the oxidation reaction resulting in soot emissions. Soot reduced with the increase of biodiesel addition. Minimum soot was measured with OP20 for all engine loads. Oxygen content of biodiesel prevents to release soot emissions because fuel molecules can be properly ignited due to higher oxygen concentration with the usage of biodiesel. This effect reduces soot emissions. At 100% engine load, soot reduced by about 20.36% and 34.34% with OP10 and OP20 compared to D100 respectively.

Low soot emissions were seen with biodiesel due to more oxygen content as previous works [34,37,38,49,50]. In addition, biodiesel has no aromatics. The results showed that B85P15 50% reduction compared to biodiesel at high engine load. Similarly HC and CO decreased generally with biodiesel. Fig. 14 presents the $NO_x$ and soot emissions versus engine speed. $NO_x$ decreased with the increase of engine speed at 100% engine load. Flow and heat losses increase at higher engine speeds. Cylinder could not be charged with fresh air-fuel mixture because of less time at higher engine speed. In addition, gas leakages increased resulting in decreasing volumetric efficiency at higher engine speeds. The amount of molecules that participate into the oxidation reactions decrease at higher engine speeds. These effects cause to release lower heat energy and in-cylinder temperature. Oxygen and nitrogen can not be reacted at lower in-cylinder gas temperatures. So, $NO_x$ reduces. Until 2200 rpm engine speed, sufficient charge mixture can be delivered into the cylinder and higher in-cylinder temperature is obtained during combustion. So, $NO_x$ increases with higher in-cylinder gas temperatures. 415, 435 and 445 ppm $NO_x$ were measured with D100, OP10 and OP20 respectively at 3000 rpm and 100% engine load. Otherwise, higher $NO_x$ was released at maximum torque speed of 2200 rpm. Biodiesel percentage also affects $NO_x$. Higher biodiesel fraction in the fuel blend resulted in lower calorific value of fuel mixture. Lower heat is released with the combustion of fuel blend. Higher biodiesel fuel blend could not be well atomized due to higher viscosity and density. The size of injected particles increases with the higher percentage of biodiesel. It results in lower combustion efficiency. Maximum in-cylinder temperature also decreases. Hence, NO formation reduces [23,24]. So, lower biodiesel fraction was selected in view of the higher combustion efficiency. Fig. 14-b shows the soot emissions depending on engine speed. It can be clearly said that soot reduces with the increase of engine speed. Homogeneity of the charge mixture improves with the increase of engine speed. Rich mixtures zones could not be formed in the combustion chamber with higher engine speed. So, whole charge mixture is ignited without formation of soot. At 3000 rpm, 9.43% and 21.22% reduction was observed on soot emissions with OP10 and OP20 compared to D100 respectively. Oxygen can also be used in piston and ring cavities in the combustion chamber at higher engine speed due to improved turbulence. Local rich mixture zones could not be formed in the combustion chamber. So, soot is reduced.

## 4. Conclusions

Opium poppy oil biodiesel has drawn great attraction in view of considerable engine performance and reduced exhaust emissions compared to diesel in diesel engines. Although viscosity and density of opium poppy oil biodiesel is higher than diesel, calorific value is near to neat diesel. So, it has important potential to use in diesel engines. The aim of this study is to determine the effects of opium poppy oil biodiesel on combustion, engine performance, $NO_x$, CO and soot emissions with different engine loads (25%, 50%, 75% and 100%.) and engine speeds of 1750, 2000,2200, 2500, 2750 and 3000 rpm. There is no detailed experimental investigation with opium poppy oil biodiesel on combustion and engine performance. Detailed combustion and performance analysis were performed in the current study. This study aims to fill this knowledge gap. Higher in-cylinder pressure was observed with opium poppy oil biodiesel compared that diesel. ITE decreased by about 5.72% and 13.03% with OP10 and OP20 compared that D100 at 75% engine load. In addition, lower imep was obtained with OP10 and OP20 according to D100. ID increased with the usage of biodiesel fuel blends due to higher viscosity and density of opium poppy oil biodiesel. Combustion duration decreased with biodiesel fuel blends according to diesel. The most remarkable effect of opium poppy oil biodiesel was

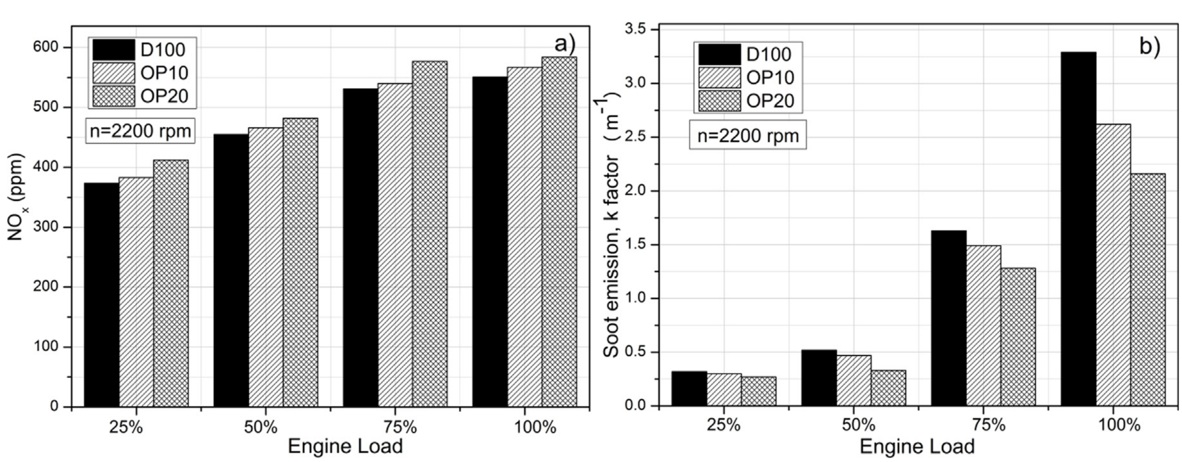


**Fig. 13.** $NO_x$ and soot emissions versus engine load.

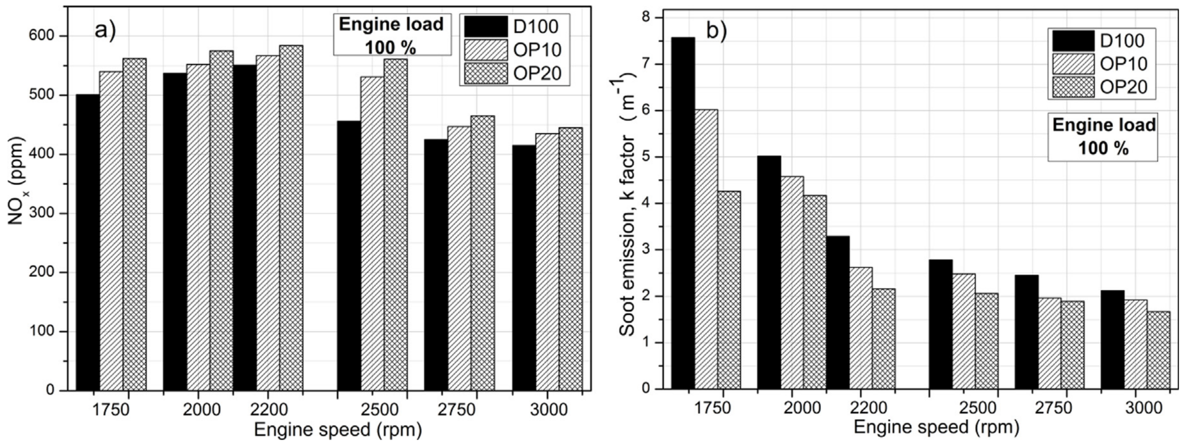


**Fig. 14.** $NO_x$ and soot emissions versus engine speed.

observed on CO and soot emissions. CO reduced by about 14% and 17.42% with OP10 and OP20 according to D100 at 100% engine load. Similarly, 20.36% and 34.34% reduction was measured on soot with OP10 and OP20 compared to D100 at 100% engine load. However $NO_x$ increased 2.9% and 5.98% with OP10 and OP20 compared to D100 at 100% engine load. $NO_x$ decreased by about 23.8% and 23.28% with OP20 at 3000 rpm compared to 2200 rpm at 100% engine load. Opium poppy oil biodiesel has presented considerable engine performance compared to neat diesel. As a result, opium poppy oil biodiesel can be efficiently used in diesel engines without detailed modification in diesel engines.

## CRediT authorship contribution statement

**Ahmet Uyumaz:** Writing - original draft, Conceptualization. **Bilal Aydoğan:** Writing - original draft, Conceptualization. **Emre Yılmaz:** Investigation, Resources, Data curation. **Hamit Solmaz:** Investigation, Resources, Data curation, Formal analysis. **Fatih Aksoy:** Supervision, Project administration, Methodology. **İbrahim Mutlu:** Project administration, Methodology. **Duygu İpci:** Writing - review & editing. **Alper Calam:** Writing - review & editing, Validation.

## Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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