Investigation of The Effects of Biodiesel Produced from Crambe Abyssinica Plant Using KOH and NaOH Catalyst on Combustion, Engine Performance and Exhaust Emissions

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Abstract

In this study, biodiesel fuel produced from crambe abyssinica plant using KOH and NaOH catalysts was mixed with standard diesel fuel and the effects on engine performance, combustion and emission were experimentally investigated. During the experiment, in-cylinder pressure data were specified for each test fuel and engine load. In addition, measurements of HC, NO_x, CO and smoke emissions were carried out. With the obtained experimental data, parameters such as heat release rate, combustion stages, thermal efficiency, indicated mean effective pressure (imep), ignition delay, ringing intensity and specific fuel consumptions were calculated and evaluated in MATLAB/Simulink environment. It was concluded that the highest thermal efficiency values were achieved with CAKB25 mixed fuel under all engine load conditions. It has been determined that using crambe abyssinica KOH catalyst (CAK) and crambe abyssinica NaOH catalyst (CAN) biodiesel fuel mixtures on diesel engine instead of standard diesel fuel improves CO, HC, and smoke emissions but increases NO_x values slightly.

Keywords: Crambe abyssinica, Biodiesel, Combustion, Engine Performance, Emissions

1. Introduction

Today, with the increase in the usage of motor vehicles, the risk of depletion of petroleum based fuels and air pollution problems have reached significant levels (Shrivastava et al. 2019; Elgarhi et al. 2020). In addition, the usage of internal combustion engines on motor vehicles cannot be dispensed with for many reasons. This situation has directed researchers to the production and use of more environmentally friendly renewable energy resources. At the basis of these renewable energy sources are biodiesel fuels that can be produced from vegetable oils and animal fats found in industry (Uyumaz et al. 2018; Yeşilyurt et al. 2020; Dharmaraja et al. 2018; Arunkumar et al. 2019; Likoza et al. 2016). One of the most important advantages of biodiesel is that biodiesel does not require any extra equipment for its usage in diesel engines. Although biodiesel fuel on diesel engines causes some decrease on engine performance compared to standard diesel fuels, remarkable improvement on emission values can be observed (Calam et al. 2020; Ardebili et al. 2020; Başaran 2020). Although biodiesel is produced from many edible oils, their social and economic sustainability is at risk and they cannot compete. In addition, it is predicted to negatively affect the hunger problem that occurs and may occur in underdeveloped and developing countries (Raju et al. 2018; Alagu et al. 2016). Researchers have sought an alternative, inedible raw material to biodiesel production (Allami et al. 2019; Zhong et al. 2016; Ramalingam et al. 2017). At this point, crambe abyssinica, one of these inedible raw materials, has been found to be suitable in many respects for biodiesel production (Costa et al. 2018). Crambe abyssinica is an annual plant and is used in the machinery industry, oil industry and biodiesel production due to its high oil efficiency between 35% and 60%. Almeida et al., in his study, stated that they obtained oil between 26% and 34% from crambe abyssinica plant grown in different years in Portugal (Almeida et al. 2019). The remaining part of the oil extracted crambe abyssinica plant is also widely used as a food product in livestock (Seyis et al. 2013).

Costa et al., achieved the production of biodiesel using the transesterification method and sodium hydroxide (NaOH) catalyst with the oil. They obtained biodiesel from the seeds of the crambe abyssinica plant and reached the conclusion that it meets the EN14214 standards according to the analysis (Costa et al. 2019). Rosa et al., produced biodiesel that meets EN14214 standards using the transesterification method and potassium hydroxide (KOH) catalyst (Rosa et al. 2014). Gülüm et al., produced biodiesel from corn oil using KOH and sodium hydroxide (NaOH) catalysts. It has been observed that the biodiesel fuel produced with both catalysts meets EN 14214 and ASTM D 6751 standards (Gülüm et al. 2015). Mahlia et al., in their detailed patent research on biodiesel have realized that biodiesel production by transesterification method is simpler and lower costs compared to other methods (Mahlia et al. 2020). Rajkumar et al., aimed to reduce NO_x emission by using biodiesel in diesel engines. They investigated the performance, combustion and emission characteristics of the diesel engine using experimental and modeling methods, using different combinations of biodiesel fuels produced from Karanja and coconut oils. They concluded that NO_x emission decreased with the increase in the ratio of fuel produced with coconut in the biodiesel blend. Karanja has presented a reduction of approximately 18% in NO_x emission with the hydrogenation process of biodiesel fuel (Rajkumar et al. 2019). Abed et al., investigated the effect of biodiesel fuels produced from jatropha, palm, algae and waste cooking oil on the values of CO, CO₂, NO_x, HC and smoke emissions released as a result of the use of the diesel engine. They have tested biodiesel fuels at different concentrations such as B10 and B20 on a single-cylinder diesel engine under different load conditions. In all biodiesel fuels, CO, HC and smoke emissions were found to be lower than standard diesel fuel. It has been concluded that biodiesel fuel produced from waste cooking oil emits more CO₂ emission in B10 and B20 concentrations than other biodiesel and standard diesel fuels (Abed et al. 2019). Uyumaz et al., produced biodiesel fuel from waste tyre. They investigated the effects of W10 biodiesel fuel on the single-cylinder diesel engine, in-cylinder pressure, ignition delay (ID), and combustion time and engine performance. As a result of the tests, it was concluded that using W10 biodiesel fuel instead of standard diesel fuel causes higher in-cylinder pressure and heat transfer. Although W10 biodiesel fuel used on diesel engine showed performance values close to standard fuel, they observed an increase in BSFC (brake specific fuel consumption) by 18.5% at 11.25 Nm engine load (Uyumaz et al. 2019). Uyumaz et al., investigated the effect of L10, L20 and L30 fuel mixtures, which contain linseed biodiesel fuel and diesel fuel mixture, on engine performance, specific fuel consumption and emission values. The usage of biodiesel fuel blends has revealed reductions on CO and soot emissions. They concluded that if the engine was loaded with 18.75 Nm and the L30 fuel was used, the CO emission decreased approximately 36.2%, whereas NO_x emission increased by 12.7%. They evaluated it as L10 as the most suitable fuel (Uyumaz 2020). Uyumaz et al., carried out detailed performance tests and combustion analysis of OP10 and OP20 mixture of biodiesel produced from poppy oil on a single cylinder diesel engine at 2200 rpm and different engine loads. In the test results, it was observed that biodiesel blends increased the in-cylinder pressure and heat release rate amount. Under full load conditions of the engine, NO_x emission increased by 2.9% and 5.98%, CO emission decreased by 14% to 17.2% with OP10 and OP20 biodiesel fuels compared to diesel fuel, and thermal efficiency was 5.73% and 13.05% found that it decreased (Uyumaz et al. 2020).

In the literature, there is limited number of studies on the engine performance, combustion and emission values of biodiesel fuel produced from crambe abyssinica plant involving KOH and NaOH catalysts. It is considered that the current study is aimed to clarify this issue. In this study, the effects of biodiesel fuel produced from crambe abyssinica plant using KOH and NaOH

catalysts on engine performance, combustion and emission values were investigated by mixing with standard diesel fuel.

2. Experimental Setup and Procedure

Test fuels consisting of different biodiesel concentration and standard diesel, on the engine were examined on a test setup containing a single cylinder diesel engine. Test setup basically consists of diesel engine, dynamometer, emission device, smoke meter, precision balance, thermocouple, data processing card and computer. In addition to these, a sensitive pressure sensor is used to measure the in-cylinder pressure and an encoder capable of measuring 0.36 °CA precision in determining the engine crank angle. The schematic representation of the test setup where the experiments are carried out is shown in Figure 1.

Fig. 1 Schematic representation of the test setup

All fuels were tested at 3.75, 7.5, 11.25 and 15 Nm load. The technical specifications of the test engine are shown in Table 1. No structural changes were made to the diesel engine during the test processes.

Table 1 Test engine specifications

Model	Antor/6LD400
Engine type	Direct injection, naturally aspirated
Number of cylinder	1
Bore x Stroke [mm]	86 x 68
Cylinder volume [cm ³]	395
Compression ratio	18:01
Maximum power [kW]	5.4 @ 3000 rpm
Maximum torque [Nm]	19.6 @ 2200 rpm
Geometry of combustion chamber	U type
Fuel injection system	PF Jerk type fuel pump

Injection nozzle	0.24 [mm] x 4 hole x 160°
Nozzle opening pressure [bar]	180
	24 before top dead center
Injection timing [°CA]	(BTDC)

7 different fuel mixtures were tested in the test setup. The engine was brought to operating temperature for each experiment, the necessary controls were made and the conditions were kept stable for all experiments in order to enhance durability. During the experiments, in-cylinder pressure signal covering 50 engine cycles with a sensitivity of 2000 pulse/cycle was received. The received signals are transformed into pressure data with a data processing card and transferred to the computer. In order to increase the precision and accuracy of the test results, the average of 50 cycles was evaluated and the evaluation was made over a single cycle.

Exhaust emissions were measured with EGAS-2M model exhaust gas analyzer produced by Environment SA firm. The technical specifications and measurement sensitivity of gas analyzer are shown in Table 2.

Table 2 Technical specifications for EGAS-2M emission measuring device

Model	Graphite 52M	Topaze 32M	MIR 2M
Gases	THC	NO-NO _x	CO-CO ₂ -O
Measurement procedure	HFID	HCLD	NDIR-paramagnetic
Linearity	< %1	< %1	< %1
Measuring range	0-10/30000 ppm	0-10/10000 ppm	0-500/10000ppm (CO) 0-1/20 % (CO ₂) 0-5/25 % (O ₂)
The lowest measurement	0.05 ppm	0.1 ppm	<2% (For full scale)
Response time	<1.5	<2 s	<2 s

During the experiments, smoke was measured with the AVL DiSmoke 4000 brand / model smoke meter whose technical specifications are given in Table 3.

Table 3 Smoke meter specifications

Device model	AVL DiSmoke 4000	
	Partial flow	
Measurement method	Opacity	K value
Operating range	0–100%	Accuracy 0.1%
Accuracy [m ⁻¹]	0–99,99	0,01

The measurement of the in-cylinder pressure on the diesel engine is provided by the pressure sensor whose technical specifications are given in Table 4.

Table 4 Pressure sensor specifications

Model	AVL 8QP500c quartz
Operating range [bar]	0-150
Measurement precision [pC/bar]	11,96
Linearity [%]	±0.6
Natural frequency [kHz]	>100

As a result of the tests, engine in-cylinder pressure values depending on the crankshaft angle were obtained for all fuel types. Parameters such as combustion, heat release rate, combustion stages, thermal efficiency, imep, ID, RI and specific fuel consumption are calculated in MATLAB / Simulink environment with the obtained test data.

One of the most important factors in the examination stages of fuel characterization is the heat release rate behavior during the combustion process. The heat release rate calculation in the combustion process depending on the crank angle is provided by the equation given in Eq. 1, derived from the first law of thermodynamics. In this equation, leaks are neglected, heat transfer from the cylinder walls ($\dot{Q}_{heat}$) is included (Solmaz 2020). Here, $\dot{Q}$ is the heat release rate, P is the pressure inside the cylinder, V is the cylinder volume, k is polytropic index and θ is the crankshaft angle.

$$\frac{d\dot{Q}}{d\theta} = \frac{k}{k-1} P \frac{dV}{d\theta} + \frac{1}{k-1} V \frac{dP}{d\theta} + \frac{d\dot{Q}_{heat}}{d\theta} \quad (1)$$

Calculation of the heat transfer from the cylinder walls in the heat transfer equation is given in Eq. 2 and its unit is defined as J/°CA. In the equation, n is the engine speed, h_c is the heat convection coefficient, T_g is the mean gas temperature inside the cylinder and T_w is the combustion chamber wall temperature.

$$\frac{d\dot{Q}_{heat}}{d\theta} = \frac{1}{6n} h_c A_2 (T_g - T_w) \quad (2)$$

For the calculation of the heat conduction realized in the combustion chamber, the equation (W/m²K) created by Woschni seen in Eq. 3 was taken as reference (Aydoğan 2020). The c_m in the equation denotes the average piston speed.

$$h_c = 130 V^{-0.06} P^{0.8} T_g^{-0.4} (c_m + 1,4)^{0.8} \quad (3)$$

The net work performed throughout the cycle is calculated in Eq. 4 and the Indicated mean effective pressure is calculated with Eq. 5 (Yılmaz 2019). Equivalent V_{stroke} refers to the cylinder volume.

$$W_{net} = \int_{CA0^\circ}^{CA720^\circ} P dV \quad (4)$$

$$Imep = \frac{W_{net}}{V_{stroke}} \quad (5)$$

The equation where thermal efficiency is calculated by using experimental data is shown in Eq. 6. When using standard diesel and biodiesel fuel mixtures as test fuel, two different fuel amounts (m_{D0} , m_{Bn}) and lower calorific value (Q_{D0} , Q_{LH-Bn}) are entered into the divisor index due to the difference in the lower heating values of the fuel.

$$\eta_T = \frac{W_{net}}{m_{D0} Q_{LH-D0} + m_{Bn} Q_{LH-Bn}} \quad (6)$$

One of the most important issues to be examined in diesel engines and fuels is RI. Eq. 7 gives the intensity of ringing encountered by the engine during one cycle. T_{max} and P_{max} in 7 in the equation represent the maximum in-cylinder temperature and pressure, and γ represent the polytropic coefficient values.

$$RI = \frac{1}{2\gamma} \frac{\left(\beta \left(\frac{dP}{dt} \right)_{max} \right)^2}{P_{max}} \sqrt{\gamma \cdot R \cdot T_{max}} \quad (7)$$

The model of each subsystem was created by providing the input of the equations between Eq. 1 and Eq. 7 to the MATLAB / Simulink simulation program environment. The visual of the Simulink model created is given in Figure 2. All analyzes were carried out depending on the crank angle. Graphics were created by exporting the results obtained in MATLAB / Simulink environment to an external file.

Fig. 2 Simulink model visual

2.1. Production of Biodiesel

By using KOH and NaOH catalysts, biodiesel fuels produced from *crambe abyssinica* plant were mixed with standard diesel fuel and fuels of different concentrations were obtained. Standard diesel fuel is called D0, the mixture of biodiesel obtained with KOH catalyst and standard diesel fuel is CAK, and the mixture of biodiesel obtained with NaOH catalyst and standard diesel fuel is CAN. Transesterification parameters are as follows 6:1 Methanol to oil molar ratio, KOH catalyst 0.8 g (w/w), NaOH catalyst 0.40 g (w/w), 57°C reaction temperature, and 60 min reaction time. For separation of the methyl esters and glycerol, the mixture was left in separating funnel for 8–10 h. After removing glycerin from the separating funnel, obtained sample was washed with hot distilled water (about 85°C) several times until the washing water became clear. Drying procedure of biodiesel was occurred at 120°C for 20 min to remove any remaining water.

Fuels are named D0 (100% diesel), CAKB25 (biodiesel produced with 25% KOH catalyst + 75% diesel), CAKB50, CAKB75, CAN25 (biodiesel produced with 25% NaOH catalyst + 75% diesel), CAN50 and CAN75. The experiments were carried out under 2200 rpm and 3.75, 7.5,

11.25, and 15 Nm load conditions in a diesel engine. During the experiments, the measurements of the in-cylinder pressure were performed depending on the crank angle. Emission values were directly measured with different fuel mixtures at different engine load conditions. The obtained in-cylinder pressure data were evaluated using mathematical equations in MATLAB / Simulink environment by obtaining parameters such as heat release rate, amount of heat transfer from the cylinder, RI, thermal efficiency, imep, ID, start and end of combustion positions.

The properties of CAK, CAN and standard diesel fuel are shown in Table 5. Among the values in Table 5, density, viscosity, water and sulfur ratio values were obtained from the analysis result and the lower calorific value was taken from the reference source (Rosa et al. 2014). 6 different fuels with 25%, 50% and 75% concentration of CAN and CAK biodiesel fuel were created with standard diesel fuel.

Table 5 The properties for standard diesel, CAK and CAN biodiesel fuels

Test fuels	CAK	CAN	Diesel
Density @ 15 °C (g/cm ³)	879.70	880.03	841.75
Kinematic viscosity (40 °C mm ² /s)	6.492	6.716	3.354
Water content (ppm)	378.49	403.58	-
Sulfur (mg/kg)	1.3	1.3	-
Lower heating value (J/kg)	43616	43616	44250

3. Results and Discussion

In this study, by using KOH and NaOH catalysts, biodiesel fuel produced from *crambe abyssinica* plant was mixed with standard diesel fuel and the effect on engine performance, combustion and emission values was investigated by experimental methods. During the experiments of fuels, in-cylinder pressure measurements were made at a precision of 0.36 ° crank angle at 2200 rpm and 3.75, 7.5, 11.25 and 15 Nm load conditions of the diesel engine. In addition, exhaust emission measurements were made for each fuel type and engine load. In-cylinder pressure values obtained depending on the crankshaft were analyzed with different equations in MATLAB / Simulink environment, and values such as imep, RI, thermal efficiency, burning time and ID in the combustion cycle of the engine were determined. The heat release rate obtained by burning the standard diesel fuel under the condition of 3.75 Nm engine torque, in-cylinder pressure, first and second derivative graph and the start of combustion with the start of injection are given in Figure 3. Fuel injection advance is 24 °CA and injection starts at 336 °CA. As the fuel combustion start, it is determined by accepting the crankshaft angle that the heat release rate changes from negative value to positive value.

Fig. 3 Heat release rate, in-cylinder pressure, 1st and 2nd derivative graphs

Combustion stages of standard diesel fuel under 3.75 Nm engine torque condition are shown in Figure 4. CA10, CA50 and CA90 mean that 10%, 50% and 90% of the charge mixture that is burned versus crank angle. Since the combustion after CA90 point cannot be well detected due to heat transfer to the cylinder wall and after burning process in combustion studies, and it is accepted as the end of combustion.

Fig. 4 Standard diesel fuel combustion stages

The graphics containing the in-cylinder pressure and heat release rate values obtained by burning D0, CAK and CAN fuels at different engine loads are given in Figure 5. The highest pressure and heat release rate value was obtained with standard diesel fuel when the engine load was 3.75, 7.5 Nm and with CAKB25 mixed fuel at 11.25 and 15 Nm engine load. Under all engine load conditions, it was observed that the maximum cylinder pressure values gradually decreased as the biodiesel concentration rate increased. It was determined that with the same concentration of CAK fuel series, lower maximum pressure values were achieved compared to the CAN fuel series. It was seen two stages of combustion called pre-combustion and diffusion combustion as seen in heat release rate variation. It can be found from Figure 5 that lower heating value caused to obtain lower in-cylinder pressure and heat release with the addition of biodiesel. It can be also mentioned that biodiesel showed better performance with the increase of engine load in view of in-cylinder pressure and heat release rate. Biodiesel cannot be well atomized and vaporized compared that D0 due to higher density and viscosity. It can be explained that longer combustion process showed positive effect with the combustion of biodiesel fuel blends in view of complete combustion.

Fig. 5 Cylinder pressure and heat release rate graphs

There is significant relationship between thermal efficiency and CA50. If CA50 is obtained nearly after top dead center (ATDC), higher thermal efficiency can be observed. Thermal efficiency and CA50 values determined in different fuel mixtures and engine loads are shown in Figure 6. It was concluded that the highest thermal efficiency values were achieved with CAKB25 fuel blend for each engine load. While the highest thermal efficiency value was obtained in fuel types with a biodiesel concentration of 25%, a gradual decrease was observed in fuel types with a 50% and 75% concentration. The highest thermal efficiency was computed as 21.32% with CAKB25 at 11.25 Nm engine load. It has been determined that the use of standard diesel fuel has higher thermal efficiency than CAK75 and CAN 75 fuel mixtures under all engine load conditions due to higher heating value. It was seen that there is good agreement between thermal efficiency and CA50.

Fig. 6 Thermal efficiency and CA50 values

The imep values obtained as a result of the experiment with different fuel mixtures and engine loads are shown in Figure 7. Imep can be defined as engine performance indication. It was observed that the highest imep value was obtained in CAKB25 as 4.49 bar at 15 Nm engine load.

It was found that the same concentration CAK biodiesel fuel mixture had a higher imep value at 3.75, 11.25 and 15 Nm engine loads than CAN fuel mixtures. More charge mixture is taken to the cylinder resulting in higher heat release. So, the in-cylinder pressure exerted on the piston in a cycle increases. Hence, imep increases.

Fig. 7 Comparison of imep values

The variations of combustion duration values versus engine load are shown in Figure 8. Combustion duration was determined between the start of combustion and CA90. It has been observed that the shorter combustion was achieved with standard diesel for all engine load conditions. It was determined that combustion duration decreased with CAK fuel blends at the same concentration according to CAN fuel blends. The longest combustion duration was computed with CANB75. When standard diesel fuel was used at 3.75 Nm, it was determined that the lowest combustion duration was 95.911 °CA and the CANB75 fuel mixture presented the highest combustion duration at 15 Nm. The increase of engine load caused to take more charge mixture resulting in longer time to complete combustion. Higher viscosity and density of biodiesel fuel blends make difficult to atomize and vaporize fuel molecules. Thus, more time is needed to complete combustion.

Fig. 8 Comparison of CA0-90 combustion duration

ID is directly dependent on fuel property that is cetane number. Higher cetane number causes the shorter ID. The time elapsed between the fuel start of injection and the start of combustion is called the ID (Li et al. 2019). ID values obtained by burning D0, CAK and CAN fuels at different engine loads are given in Figure 9. It has been observed that the ID of the standard diesel fuel is the highest under 3.75, 7.5 and 11.25 Nm torque conditions of the engine, and close to the maximum under 15 Nm load condition. Higher ID is among the expectations that it will cause

knock due to the increase of accumulated fuel during combustion. At 11.25 Nm engine load, the lowest ID values were determined. Surprisingly, biodiesel fuel blends present lower ID compared that diesel fuel in spite of higher density and viscosity of biodiesel.

Fig. 9 Comparison of ID values

RI is a combustion parameter depending on engine speed, maximum pressure rise rate. RI values obtained with test fuels at different engine loads are shown in Figure 10. It is concluded that the use of standard diesel fuel on the engine is more prone to knock than other blended fuels. The higher ID values of standard diesel fuel compared to other fuels strengthen the accuracy of this expected result. With the increase of biodiesel concentrations, an increase in the RI of CAK and CAN mixed fuels was observed. It has been observed that CAK fuel blends at the same concentration have lower RI than CAN fuel blends. The highest RI was calculated with DO as 0.03171 MW/m² at 7.50 Nm engine load. It can be mentioned that RI decreased with the increase of engine load. Combustion of higher charge mixture allowed obtaining higher in-cylinder temperature and pressure. So, combustion conditions are improved across the combustion chamber.

Fig. 10 Comparison of RI values

Brake specific fuel consumption values obtained as a result of measurements performed at different fuel mixtures and engine loads are shown in Figure 11. The result is that the lowest BSFC values are provided with standard diesel fuel under all engine load conditions. As the biodiesel concentration of the standard diesel fuel was increased, the BSFC value increased due to lower heating value and higher density. The same concentrated CAN mixed fuel series has been found to have a lower BSFC value at all engine loads than the CAK blended fuel series. The highest BSFC was computed with CAKB75 as 616.691 g/kWh at 3.75 Nm engine load. The

lowest BSFC values for all fuel types were achieved at a torque value of 11.75 Nm at the engine load. BSFC first decreased and then started to increase with the increase engine load for all test fuels. At medium engine loads heat losses and gas leakages decrease resulting in lower BSFC.

Fig. 11 Effect of different biodiesel blended fuels on specific fuel consumption under different engine load conditions

Incomplete combustion is seen owing to lower oxygen concentration and in-cylinder temperature allowing CO formation that is combustion product. The variations of CO versus engine load is shown in Figure 12-a. Lower CO was measured at low engine loads because less charge mixture is burned. It can be emphasized that sufficient oxygen molecules exist due to lower fuel concentration in the combustion chamber. Hence, CO formation is reduced. In contrast with this phenomenon, CO increased at higher engine loads because of higher fuel fractions in the charge mixture. This situation caused to deteriorate chemical reactions between fuel and oxygen molecules. So, CO formation is seen again. Biodiesel fuel blends presented lower CO formation due to higher oxygen concentration. Figure 12-b represents the NO_x variations. NO_x increased with the increase engine load owing to combustion of higher charge mixture. High combustion temperatures are observed at high loads. Oxygen and nitrogen molecules are reacted each other at higher in-cylinder temperature resulting in NO_x formation. The highest NO_x was measured with CAKB75 as 341.656 ppm at 15 Nm engine load. It was also seen that NO_x increased with the increase of biodiesel fraction in fuel blends. The lowest NO_x was measured with diesel for all loads. Minimum NO_x was determined with D0 as 22.489 ppm at 3.75 Nm. Smoke emissions are generally resulted from rich charge mixture at higher engine loads. As seen in Figure 12-c, higher smoke was measured at medium and higher engine loads for all test fuels. It was evaluated that lower smoke was measured with biodiesel fuel blends compared to diesel. The highest smoke was determined as 2.29 m^{-1} and 2.59 m^{-1} for D0 at 11.25 and 15 Nm engine loads respectively. Similar change was seen with HC emission as shown in Figure 12-d. HC is emitted due to incomplete combustion because fuel and oxygen molecules could not well react near the cylinder wall and piston cavity in the combustion chamber. The highest HC was determined as 2400.27 ppm with D0 at 15 Nm engine load. It was seen that HC reduced with the usage of biodiesel fuel blends. Oxygen concentration caused to improve oxidation reactions according to D0 with biodiesel.

Fig. 12 Effect of different biodiesel blended fuels on emissions under different engine load conditions

Conclusion

In this study, the effects of biodiesel fuels produced by using KOH and NaOH catalysts from *crambe abyssinica* plant, which is in the non-edible plant group, on combustion, engine performance and emission were experimentally investigated. The tests of standard diesel fuel (D0) and 6 different fuel mixtures (CAK25-50-75 and CAN25-50-75) were carried out at 2200 rpm engine speed and 3.75, 7.5, 11.25 and 15 Nm engine load conditions. The data obtained as a result of the experiments were analyzed in MATLAB / Simulink environment and parameters such as combustion heat release rate, combustion stages, and thermal efficiency, indicated mean effective pressure, ignition delay, ringing intensity (RI) and specific fuel consumption were calculated and evaluated. The study result is summarized as follows:

- It has been proved that the CAN and CAK blended fuel series with 25%, 50% and 75% biodiesel concentration can be used without any structural changes on the diesel engine.
- The highest thermal efficiency values were obtained by using CAKB25 mixed fuel under all engine load conditions. In CAK and CAN mixed fuels, the highest thermal efficiency value was obtained when the concentration was 25%, while a gradual decrease was observed in the fuel types with a concentration of 50% and 75%. It was determined that the use of D0 fuel at all engine load conditions has higher thermal efficiency than CAK75 and CAN75 fuel mixtures.
- It was observed that the highest imep value was obtained in CAKB25 fuel at all engine loads. It was found that the same concentration CAK biodiesel fuel mixture had a higher imep value at 3.75, 11.25 and 15Nm engine loads than CAN fuel mixture.

- It was found that the lowest burning time was provided with D0 fuel when the engine load was 3.75 Nm with 95.911 °CA, and the highest combustion time was provided with CANB25 fuel when the engine load was 15 Nm with 116.275 °CA.
- It was concluded that the ignition delay of D0 fuel was the highest in the engine's 3.75, 7.5 and 11.25 Nm torque conditions, and close to the maximum under 15 Nm load condition. It is concluded that the use of standard diesel fuel on the engine is more prone to knock than other blended fuels. The high ignition delay values of standard diesel fuel compared to other fuels and being compatible with the literature strengthens the accuracy of this result.
- It has been determined that the BSFC value of D0 diesel fuel is lower than CAN and CAK fuel mixtures under all engine load conditions. The same concentrated CAN fuel series has been found to have a lower BSFC value at all engine loads than the CAK fuel series. The lowest BSFC values for all fuel types were achieved at a torque value of 11.75 Nm at the engine load.
- It is concluded that using CAK and CAN biodiesel fuel mixture on diesel engine instead of standard diesel fuel reduces CO, HC and smoke emission values. In all engine load conditions of CAK and CAN fuel mixtures, as their concentration increases, the NO_x value increases and the smoke values decrease. In the results obtained with the same concentrated CAK fuel series at all engine loads, it was determined that NO_x emissions were lower than the results obtained compared to the CAN fuel series.

Abbreviations

BSFC:Brake specific fuel consumption; CA::Crank angle; CA10: Crank angle location of 10% accumulated HRR (°CA); CA50:Crank angle location of 50% accumulated HRR (°CA); CA90: Crank angle location of 90% accumulated HRR (°CA); CAK:Crambe abyssinica KOH catalyst; CAN:Crambe abyssinica NaOH catalyst; CI:Compression ignition; CO:Carbon monoxide; CO₂:Carbon dioxide; HC:Hydrocarbon; h_c :Heat convection coefficient; $\dot{m}_{ep}$:Indicated mean effective pressure; k :Polytropic index; KOH:Sodium hydroxide; n :Engine speed; NaOH:Potassium hydroxide; P :In-cylinder pressure; Q :Heat release rate; RI:Ringling intensity; SOC:Start of combustion; SOI:Start of injection; θ :Crank angle; T_g :In-cylinder temperature; T_w :Cylinder wall temperature; V :Cylinder volume.

Declarations

Ethics approval and consent to participate

Not applicable

Consent for publication

Not applicable

Availability of data and materials

The data that support the findings of this study are available from the corresponding author upon reasonable request.

Competing interests

The authors declare that they have no competing interests

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Authors' contributions

Uyaroğlu, Kocakulak and Aydoğan conceived and designed this study. Uyaroğlu performed the experiments. Kocakulak and Aydoğan analyzed the data. Kocakulak and Aydoğan wrote the manuscript. All authors read and approved the final manuscript.

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Figures

Figure 1

Schematic representation of the test setup

Figure 2

Simulink model visual

Figure 3

Heat release rate, in-cylinder pressure, 1st and 2nd derivative graphs

Figure 4

Standard diesel fuel combustion stages

Figure 5

Cylinder pressure and heat release rate graphs

Figure 6

Thermal efficiency and CA50 values

Figure 7

Comparison of imep values

Figure 8

Comparison of CA0-90 combustion duration

Figure 9

Comparison of ID values

Figure 10

Comparison of RI values

Figure 11

Effect of different biodiesel blended fuels on specific fuel consumption under different engine load conditions

Figure 12

Effect of different biodiesel blended fuels on emissions under different engine load conditions

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