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RESEARCH ARTICLE-MECHANICAL ENGINEERING

# Investigation of Effect of Dual Fuel Injection Stages on Combustion Parameters in a Diesel Engine Using Ethanol-Butan-2-ol-Fossil Diesel Blends

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## Abstract

The present study aims to evaluate on the effect of different injection strategies on the combustion and emission characteristics in a diesel engine with fueled ethanol-butan-2-ol-diesel fuel blends. The most critical problem in the use of fossil-based diesel fuel (FBDF) and ethanol mixtures as an alternative fuel is the phase separation that occurs in a very short time. Therefore, there is a need to use a stabilizer to maintain the homogeneity of the mixture for long-term test conditions. In this study, butan-2-ol (2-butanol) was added to the fuel mixtures as a co-solvent to prevent the phase separation in ethanol-FBDF mixtures. It was observed that the homogeneity of the ethanol and FBDF mixtures was ensured for a long time by adding butan-2-ol up to 20% of the ethanol in the mixture. In the experiments, three different states (main, 5% pilot and 10% pilot) were developed by changing the fuel injection rate shapes. Engine tests have been performed by spraying a total of 45 mg fuel for each fuel per cycle and by charging fresh air into the cylinder at constant temperature and pressure conditions. As the results of this study, the highest maximum cylinder gas pressure with FBDF was observed in a 10% pilot injection application. With the use of ethanol-butan-2-ol-FBDF blends, a decrease in CO, $CO_2$ and $NH_3$ emissions were detected, while $NO_x$ emission increased. The lowest combustion noise was seen with 15% ethanol + 3% butan-2-ol + 82% FBDF (E15B3) and 20% ethanol + 4% butan-2-ol + 76% FBDF (E20B4) fuels as 84 dB. A sharp increase in heat release rate was observed using E15B3 in 5% pilot injection compared to FBDF.

**Keywords** Ethanol · 2-butanol · Diesel engine · Dual injection · Combustion noise · Emission

## 1 Introduction

Nowadays, oil, natural gas, coal, biofuel, hydro, nuclear and renewable energy sources are used as basic energy sources in areas such as industry, transportation, housing and services. Among energy sources, fossil-based fuels are the most supplied energy source. According to the International Energy Agency (IEA) data, fossil-based fuels, which are the first in the world energy supply, constituted more than 81% of the world energy production in 2018. In 2018, the world energy production increased by approximately 3.2% compared to the previous year and became 14.421 Mtoe (Mega tons of oil equivalent). In 2018, while fossil fuels (natural gas, coal, and oil) increased by more than 370 Mtoe in energy production, production with renewable energy sources increased by 60 Mtoe and nuclear energy increased by 19 Mtoe. In addition, the world's total energy supply was 5.519 Mtoe in 1971, but by 2018 it increased by more than 2.5 times to 14.282 Mtoe. In addition, the total amount of $CO_2$ emissions in the world increased by 60% from 1990 to 2017, reaching 32.840 Mt (Mega ton). Despite the rapidly increasing use of biofuels in the transportation sector, which has seen a significant increase in energy use after 2000, OECD (Organization for Economic Co-operation and Development) member countries have obtained 92% of their energy needs from petroleum. While gas is used the most with 38% in the residential area, electricity has been the most preferred energy source with 54% in the services area [1, 2].

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Alcohol fuels, which can be produced from local resources and it can be used in addition to fossil-based fuels in vehicles as a sustainable energy source, stand out in more environmentally friendly energy resources research to reduce dependence on fossil-based fuels [3]. European Union (EU) supported use of biofuel in contrast limited use of fossil-based fuel with the regulation that was published in December 2018. To determine the effects of biofuels on greenhouse, guidelines released in June 2010 were revised in terms of examine of land carbon stocks rate. It is expected share of biofuel and biogas obtained from feedstock to be used in transport sector shall be at least 0.2%, 1% and 3.5% (2022, 2025 and 2030, respectively) [4].

Biofuels means liquid fuel for transport produced from biomass. Ethanol is one of the biofuel types which is a simple alcohol that can dissolve in water and contains 52% of carbon, 35% of oxygen and 13% of hydrogen. One of the major problems of ethanol to mix with fossil-based diesel fuel is phase separation. It is known, ethanol is more hygroscopic than fossil diesel fuel, ethanol absorbs water from ambient humidity and excessive water concentration in mixtures causes phase separation. Having a high level of water content leads ethanol-FBDF to phase separation at low mixing ratio. Ethanol contains some polar molecules that can form hydrogen bonds. Having hydrogen bonds lead to a limiting blend ratio of alcohols with low carbon numbers to FBDF without adding a co-solvent [5, 6]. It was reported by [7, 8] that when the ethanol ratio in ethanol–diesel fuel blends increases, phase separation time is shortened. Iso-propanol, n-propanol, n-butanol and iso-butanol are advised to be used as stabilizers in ethanol-FBDF and to be used for preventing alcohol-diesel fuel of fossil origin from phase separation [9, 10].

Generally, engine researchers [11, 12] showed the combustion behaviors in a gasoline engine fueled with gasoline-ethanol blends without stabilizer. However, the scientists [13–16] conducted on diesel engine experiments with *n*-butanol, etc., as a stabilizer. Huang et al. [17] investigated the performance and emission values in a diesel engine by adding 5% n-butanol of the mixture to ethanol–diesel fuel mixtures and using 100% diesel fuel. They reported an increase in the brake specific fuel consumption (BSFC) values with an increase in the amount of ethanol in the mixture. In that study, it was observed that the CO values obtained with blended fuels are lower when the engine was operated with a load more than 50%. Also, a reduction of up to 87% in smoke emissions has been detected with the use of blended fuels. Huang et al. stated that the phase separation did not occur up to 11 days at the earliest with the addition of n-butanol to ethanol–diesel fuel blends. Like the study of Huang et al., in our study, 20% of ethanol in ethanol-FBDF blends, 2-butanol was used as co-solvent to make blends fuel more stable. It was seen that the addition of 2-butanol (or butan-2-ol) to fuel mixture has a slowing effect on phase separation timing between FBDF and ethanol.

There are some studies in the literature about the effect of injection strategies on combustion phenomena in a diesel engine fueled with FBDF-ethanol blends. Kaiadi et al. [18] investigated the effect of single and double injection strategies on combustion and emission characteristics by using 100% ethanol fuel in a diesel engine with common-rail fuel injection system. As a result of that study, it was reported that the results obtained with the double injection system provided a more controllable combustion range compared to a single injection application and a lower combustion noise was obtained. Sayin and Uslu [19] investigated the effect of change of injection start time on engine performance and exhaust emissions in diesel engine using ethanol–diesel fuel. As the ethanol ratio in the mixture fuels increased and the injection start time was advanced from the Top Dead Center (TDC) compared to the original injection time, an increase was observed in nitrogen oxides ($NO_x$) and carbon dioxide ($CO_2$) emissions, while a decrease in carbon monoxide (CO) emissions was detected. BSFC values increased in all fuel types by advancing the spraying start time from TDC according to the original injection start time. Li et al. [20] investigated the performance, combustion and emission characteristics of a diesel engine using isopropanol-butanol-ethanol (IBE)/diesel blends. In that study, the pilot (25, 30, 35, 40 and 45 °CA Before Top Dead Center (BTDC)) and main (6, 9, 12, 15 and 18 °CA BTDC) injection times were changed, and the tests were done. In all test fuels, CO and HC emissions decreased as the main injection start time was advanced from 6 to 15 °CA angle, while CO and unburned HC emissions increased at 18 °CA angle. Compared to diesel fuel, it was observed that as the IBE ratio in the mixture increased at all main injection times, the ignition delay time increased. The maximum combustion time for each fuel was obtained at the starting time of the injection 6 °CA BTDC. It was observed that the pressure in the cylinder increased as the injection start time for each fuel was advanced from 6 to 18 °CA earlier. Ramachander et al. [21] investigated the effects of different injection strategies on combustion and exhaust emissions using diesel–biodiesel-pentanol fuel mixtures. They found a significant increase in combustion characteristics with increasing injection pressure. They have seen an increase in BSFC values with the use of mixed fuels. Compared to diesel fuel, they achieved a reduction in CO and HC emissions with the use of the blended fuels. Rasool et al. [22] investigated the effect on the performance and emissions of a diesel engine by changing the injection parameters. As a result of advancing the injection timing and increasing the injection pressure, they found a decrease in CO emissions, while they found an increase in $NO_x$ and $CO_2$ emissions. In another study, Chen et al. [23] investigated the effects of fuel injection strategies on combustion using iso-propanol and n-pentanol in diesel with 20% ratio by volume under low-temperature reactive condition. As a result of that study, the longest ignition delay period and the shortest combustion duration were obtained with the use of alcohol-diesel fuel mixtures. They stated that when the same injection timing is applied, the low-temperature reactive iso-propanol-diesel mixture has lower heat release and combustion temperature than the n-pentanol-diesel mixture, which means that the iso-propanol-diesel mixture is more suitable for modern diesel engine with two-injection strategies.

As mentioned above, injection strategies and alternative fuels containing oxygen (biofuels) have a significant impact on the combustion characteristics of diesel engines. In addition, the key problem with ethanol-FBDF mixtures is that the phase separation takes place in a short time. Therefore, a stabilizer should be used to maintain the homogeneity of the mixture for long-term test conditions. Various co-solvents have been used in the literature to improve the stability of ethanol-FBDF. One of the aims of this study is to experimentally examine the effect of butan-2-ol, which is used as a co-solvent, on the phase separation to keep the ethanol-FBDF mixtures homogeneous for a long time. In this study, it was seen that the significant improvements in phase separation time between ethanol and FBDF took place by adding butan-2-ol up to 20% volume of ethanol fuel into fuel mixtures. Another aim of this study is to experimentally examine the effect of the obtained fuel mixtures (ethanol-butan-2-ol-FBDF) on the combustion characteristics of today's diesel engines. Therefore, in this study, the combustion and exhaust emission values were investigated by applying two-stage injection strategies (pilot and main injection rates) in a direct injection diesel engine with equipped common-rail fuel injection system. The engine tests were carried out at 1600 rpm constant engine speed and 50% engine load by changing the fuel injection rates of a single cylinder diesel engine using FBDF and volumetric ethanol-butan-2-ol-FBDF blends (named as E5B1, E15B3 and E20B4). Fuels were first injected into the cylinder with main injection before the top dead point, which is the standard injection strategy of the engine for 1600 rpm specified by the engine manufacturer, and then by changing the main injection quantity 5%–95% and 10%–90% in accordance with the fuel injection rates. The received data were interpreted with reference to injection timing and neat FBDF and main injection strategy.

## 2 Material and Methods

### 2.1 Preparation of Test Fuels

In the study, four fuel blends were used, and these blends are named according to the ratio of ethanol and butanol they contain. The content of the fuel called FBDF is 100% diesel fuel. In E5B1 fuel contains 5% ethanol + 1% 2-butanol + 94% FBDF, E15B3 fuel contains 15% ethanol + 3% 2-butanol + 82% FBDF and E20B4 fuel contains 20% ethanol + 4% 2-butanol + %76 FBDF.

**Table 1** Fuel properties of FBDF, Ethanol and 2-Butanol

| Properties | FBDF | Ethanol ($C_2H_6O$) | 2-Butanol ($C_4H_{10}O$) |
|---|---|---|---|
| Purity | – | $\geq 0.99$ | $\geq 0.99$ |
| Density (kg/m$^3$) | 820–845 | 790 | 805 |
| Viscosity (mm$^2$/sec, 40 °C) | 2.0–4.5 | 1.13 | 3.1 |
| Lower Heating Value (MJ/kg) | 42.6 | 26.7 | 34.4 |
| Boiling Point (°C) | > 160 | 78 | 102 |
| Melting Point (°C) | – | −114.5 | −115 |
| Flash Point (°C) | $\geq 55$ | 12 | 20.5 |
| Water Content (%) | 0.020 | $\leq 0.2$ | $\leq 0.2$ |
| Cetane Number | $\geq 51$ | – | – |
| Auto-ignition Temperature (°C) | ≈ 210 | 361 | 405 |

FBDF was supplied from the national fuel station in Turkey, and J.T. Baker brand with 99.9% purity ethanol was used. In addition, to prevent phase separation in ethanol-FBDF blends, 20% of the ethanol ratio in mixture fuels Merck brand 2-butanol with 99.9% purity was added to blends. The properties of the FBDF and alcohols used in tests are given in Table 1.

In the literature, the co-solvents such as iso-propanol, n-propanol, n-butanol and iso-butanol are recommended as stabilizers to prevent phase separation in ethanol-FBDF fuel mixtures [24, 25]. By using propanol and butanol derivatives as stabilizers, ethanol-FBDF fuel mixtures become more stable, and the amount of ethanol used in ethanol-FBDF fuel mixtures increases. The stability of fuel mixtures is achieved by alcohols with higher carbon numbers becoming less polar with longer non-polar hydrocarbon chains [5]. In this study, the fuel mixtures were prepared in such a way that they would not come into contact with air, and phase separations in the mixtures were observed. As a result of the experimental observations, phase separation took place in about 20 min in all ethanol-FBDF mixtures prepared without adding butan-2-ol. However, in the experimental observations made with the addition of butan-2-ol at the rate of 20% of the ethanol content in the fuel mixtures, it is ensured that the fuel mixtures remain homogeneous for a longer period. With the addition

**Table 2** Engine specification

| | |
|---|---|
| Engine type | Single cylinder—4 stroke |
| Fuel system | Common Rail Direct Injection—1800 bar |
| Cylinder volume | 1120 $cm^3$ |
| Valves | 3 (2 intake—1 exhaust)—(OHV) |
| Max. cylinder pressure | 190 bar |
| Max. engine speed | 2500 rpm |
| Max. power | 50 kW |
| Max. torque | 160 Nm |
| Bore | 106.5 mm |
| Stroke | 127 mm |
| Compression ratio | 16.14 |

of butan-2-ol to the fuel mixtures, phase separation was first observed after 24 h in E20B4 fuel and after 21 days in E15B3 fuel, while no significant phase separation was observed in E5B1 fuel.

## 2.2 Engine Test System

In this study, engine tests were carried out a single cylinder with 4-stroke and common-rail fuel injection system under 50% load and at a constant engine speed of 1600 rpm. The schematic view of the test cell, eddy-current dynamometer and test engine is given in Fig. 1. The specification of the diesel engine used in the experimental setup is given in Table 2.

Eddy-current dynamometer, diesel engine, fuel injection system and engine control unit (ECU) are the main parts of the experimental setup. In addition to the main parts, oil and coolant pump are integrated into the experimental setup. The oil and cooling water conditions are stabilized by the engine test bench. Hence parasitic loads, other than fuel pump, are eliminated. In the experiments, AVL fuel mass flow meter and AVL Flowsonix air flow meter are used. In addition, AVL-FTIR emission measuring devices have been integrated into the test bench for the measurement of $CO_2$, CO, ammonia ($NH_3$) and $NO_x$ emissions. Information on the sensitivity of the exhaust emission device is given in Table 3. Work of the engine system fully integrated with the dynamometer and the equipment of the test cell, and all systems are operated with a controller.

The change of the injection quantity of fuel has been controlled by the driver system that allows map change by connecting to ECU. Since the engine ECU is open to the user, the main injection amount and the rail pressure map can be controlled, so its effects on the engine can be observed instantly. The engine is equipped with an AVL-GU22C in-cylinder pressure measuring device for combustion characteristics and cylinder gas pressure calculations.

**Table 3** The sensitivity of the exhaust emission device

| Measuring values | Measurement method | Range | Accuracy |
|---|---|---|---|
| CO | Non-dispersive infra-red (NDIR) | 0–3000 ppm | $\leq \pm 0.5\%$ |
| $CO_2$ | Non-dispersive infra-red (NDIR) | 0–20% | $\leq \pm 0.5\%$ |
| $NH_3$ | Laser diode spectrometer (LDS) | 0–15 ppm | $\leq \pm 3\%$ |
| $NO_x$ | Chemiluminescence Detector (CLD) | 0–10,000 ppm | $\leq \pm 1\%$ |

**Table 4** Fixed engine input parameters during the experiments

| Input parameters | Unit | Value |
|---|---|---|
| Engine speed | rpm | 1600 |
| Fuel blends | Vol | FBDF, E5B1, E15B3, E20B4 |
| Engine load | % | 50 |
| Engine coolant temperature | °C | 70 |
| Engine oil temperature | °C | 90 |
| Air intake pressure | kPa | 24 |
| Air temperature | °C | 25 |
| Total injected fuel quantity | mg/stroke | 45 mg at compression stroke |
| 5% Pilot injection fuel quantity | mg/stroke | 2.25 mg at pilot inj. / 42.25 mg at main inj |
| 10% Pilot injection fuel quantity | mg/stroke | 4.5 mg at pilot inj. / 40.5 mg at main inj |
| Standard main injection | °CA BTDC | 10.4 |

## 2.3 Test Conditions

While performing these tests, the power measurement was made as described in ISO 14396, additional requirements for exhaust emission tests in accordance with ISO 8178. In the experiments, first, the engine was operated with FBDF to allow the engine to come to stable condition until the temperature oil of the test engine was 90 °C. During the engine tests, air intake pressure and temperature was adjusted at 240 mbar and 25 °C, respectively. In addition, fuel and cooling water temperature was adjusted at 20 °C and 70 °C, respectively. Table 4 has shown fixed engine input parameters during the experiments.

Currently, the most important factors limiting the amount of fuel to be injected in the pilot injection stage in modern internal combustion engines are the engine knock tendency and the risk of maximum cylinder gas pressure before TDC. Therefore, the amount of pilot injection is limited in today's

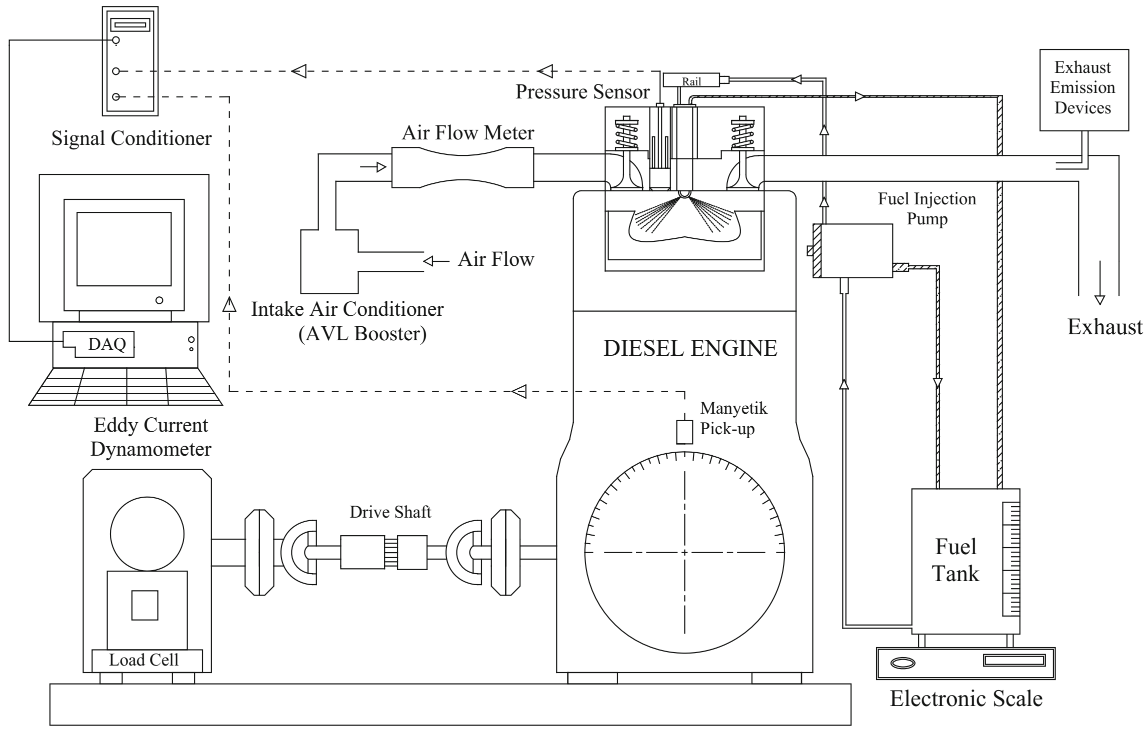


**Fig. 1** Schematic view of the experiment setup

internal combustion engine applications [26]. Considering these risk factors for this study, the amount of fuel injected during the pilot injection stage was chosen so that it does not exceed 10% of the total amount of fuel to be injected. For each fuel used in the tests, the fuels were injected at the standard main injection quantity and then the injection stages were changed as 5% and 10% pilot injections. The data was collected at each 0.1°CA, and the number of the total cycle was brought to average. As a result of the study, in-cylinder gas pressure, heat release rate, combustion noise, CO, $CO_2$, $NH_3$ and $NO_x$ emissions were compared based on test data injection quantity and fuel type, based on conventional injection quantity and neat FBDF.

## 2.4 Calculation of Combustion Phases

In this study, the start of combustion (SOC), the end of combustion (EOC), and the duration of combustion (DOC) is determined according to the crank angle of the heat release rate diagram where it crosses zero near the end of the expansion stroke. In the literature, several methods [27] were developed to estimate for SOC, EOC and DOC. Generally, when the fraction of combustion reaches 90%, it is accepted as EOC. As the third zone, it is accepted as the zone after the point where 90% of the fuel is burned. The durations stated here are shown on the combustion phases diagram in Fig. 2.

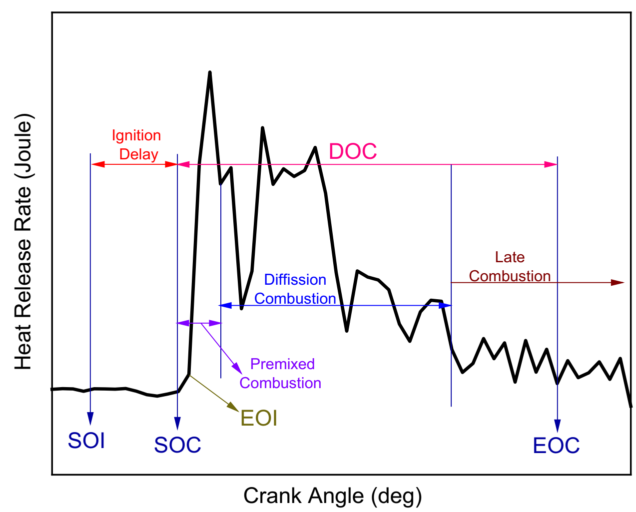


**Fig. 2** Phases of combustion

As can be seen in many studies [28, 29], the combustion phenomenon in the cylinder can be divided into three parts. The region expressed as the first part is the angle of ignition delay ($\theta_{ID}$), which is the difference between the start of injection angle ($\theta_{SOI}$) time and the start of combustion angle ($\theta_{SOC}$), ($\theta_{ID} = \theta_{SOI} - \theta_{SOC}$). The second part is the duration of combustion angle ($\theta_{DOC}$), which is the difference between the end of combustion angle ($\theta_{EOC}$) and the start of combustion angle, ($\theta_{DOC} = \theta_{EOC} - \theta_{SOC}$).

In this study, the Wiebe function is given in Eq. 1 used to predict the development of combustion phases. This formulation is calibrated by two factors: $a$ and $m$ are dimensionless numbers, $m$ indicates the formation factor, $a$ indicates the yield factor. Also in this formula, $\theta$ is the crank angle [$^oCA$], $x_b$ stands for the mass fraction of the burned, $m_b$ stands for the mass of fuel burned and $m_f$ represents the amount of fuel in the cylinder.

$$x_b = \frac{m_b}{m_f} = 1 - \exp\left[-a\left(\frac{\theta - \theta_{\mathrm{SOC}}}{\Delta\theta_{\mathrm{DOC}}}\right)^{m+1}\right] \tag{1}$$

### 2.5 Calculation of Heat Release Rate (HRR)

For a situation where there is no air flow into the cylinder between closing points of the intake and the exhaust valves, the first law equation of thermodynamics can be written as follows:

$$\mathrm{d}Q_n = \mathrm{d}W + \mathrm{d}U + \mathrm{d}Q_w \tag{2}$$

$$\mathrm{d}W = p\mathrm{d}V(Nm) \tag{3}$$

$$\mathrm{d}U = mc_v\mathrm{d}T(\mathrm{kJ/kg}) \tag{4}$$

$$\gamma = c_p/c_v \tag{5}$$

where: $\mathrm{d}Q_n$ is the net rate of heat release, $\mathrm{d}W$ is boundary work due to piston displacement (Nm), $\mathrm{d}U$ is change in sensible internal energy, $\mathrm{d}Q_w$ is heat loss to cylinder wall which can be expressed as Eq. 6., $P$ is the cylinder gas pressure ($Pa$) which was measured using a pressure transducer installed in the cylinder head, $\mathrm{d}V$ is change of cylinder volume ($m^3$), $m$ is the compressed mass, $\gamma$ is specific heat ratio, $c_p$and$c_v$ is the mean specific heat at constant pressure and volume, respectively.

$$T_g = PV/(nR_u)(K) \tag{6}$$

$$Q_w = h_c A_w\left(T_g - T_w\right) \tag{7}$$

where: $n$ is the number of moles of the working gas (mol), $R_u$ is the universal gas constant (J/(mol-K), $h_c$ is heat exchange coefficient, $A_w$ is cylinder wall surface area ($m^2$), $T_g$ is the mass-averaged gas temperature in the cylinder ($K$), $T_w$ is the wall surface temperature ($K$). The wall temperature is accepted as constant and uniform. Eichelberg's correlation [30] has been used to predict the heat transfer coefficient in Eq. 6., where $C_m$, is the mean-piston-speed, m/sec.

$$h_g = 7.67 \times 10^{-6}(C_m)^{\frac{1}{3}}(PT_g)^{\frac{1}{2}}, \left(\frac{\mathrm{kW}}{\mathrm{m^2K}}\right) \tag{8}$$

Although many types of heat transfer analysis are applied with computer modeling for today's internal combustion engines, this correlation continues to give good estimates for calculating heat transfer. When Eq. (3–7) is substituted into Eq. 2, the traditional single-zone formula in Eq. 9 can be obtained according to the first law of thermodynamics.

$$\mathrm{d}Q_n = \frac{\gamma}{\gamma - 1}P\mathrm{d}V + \frac{1}{\gamma - 1}V\mathrm{d}P + h_g A_W\left(T_g - T_w\right) \tag{9}$$

$$Q_{\mathrm{chr}} = \sum\nolimits_{i=\mathrm{SOI}}^{\mathrm{EOC}} \mathrm{d}Q_{n,i} \tag{10}$$

The cumulative heat release, ($Q_{\mathrm{chr}}$), can be calculated by using Eq. 9 over the crank angle range from SOC to EOC with adding together the heat release energy from each result of calculation [31]. This calculation process is given in Eq. 10. The calculated values with use of the equations are given in Table 5.

## 3 Results and Discussions

### 3.1 Combustion Results

In this study, the start of combustion is determined from the change in the slope of the rate of heat dissipation. The effect of change in the amount of fuel injected on combustion phases for all fuel types is given in Table 5. It was seen that the ignition delay (ID), maximum cylinder gas pressure ($\mathrm{CP_{max}}$) and maximum heat release rate ($\mathrm{HRR_{max}}$) changed significantly with different fuel injection stages. The earliest $\mathrm{CP_{max}}$ values in all fuel types were obtained in 4.8 $^o$CA After Top Dead Center (ATDC) with E20B4 fuel and 5% pilot injection. In addition, the earliest combustion started in 1 $^o$CA BTDC with 5% pilot type E20B4 fuel.

#### 3.1.1 Ignition Delay (ID)

The important fuel properties affecting ignition delay can be listed as cetane number, viscosity and oxygen content of the fuel [32, 33]. The effect of using different fuel mixtures under different injection stages on ignition delay is given in Table 5. Figure 3 shows the effect of pilot injection strategies on ignition delay.

Many studies [34, 35] on the use of ethanol-FBDF blends clearly show that the auto-ignition properties of blends deteriorate with increasing ethanol content in the mixture, as a result of the low auto-ignition tendency of ethanol. Since the cetane number of butanol is twice that of ethanol, it can slightly improve the cetane number of the mixture, but this does not contribute enough to the mixture auto-ignition properties. It was also seen in this study that ignition delay times with use of alcohol-FBDF blends were longer than those of

**Table 5** Injection and combustion characteristics of the test fuels for different injection stages

| Fuels | Injection stages | $ACP_{max}$ [°CA ATDC] | $CP_{max}$ [bar] | $HRR_{max}$ [J/°CA] | SOI [°CA BTDC] | SOC [°CA BTDC] | EOC [°CA ATDC] | ID [°CA] | DOC [°CA] |
|---|---|---|---|---|---|---|---|---|---|
| FBDF | Main | 9.8 | 79.9 | 100.6 | 10.4 | 2.4 | 38.0 | 8.0 | 40.4 |
| | 5% Pilot | 9.8 | 78.1 | 131.5 | 10.4 | 2.6 | 38.3 | 7.8 | 40.9 |
| | 10% Pilot | 8.7 | 85.6 | 76.2 | 18.8 | 6.5 | 38.1 | 12.3 | 44.6 |
| E5B1 | Main | 9.0 | 81.0 | 171.7 | 10.4 | 1.2 | 34.9 | 9.2 | 36.1 |
| | 5% Pilot | 9.1 | 79.5 | 132.5 | 10.4 | 1.5 | 34.8 | 8.9 | 36.3 |
| | 10% Pilot | 8.4 | 82.4 | 85.7 | 18.8 | 7.2 | 35.3 | 11.6 | 42.5 |
| E15B3 | Main | 8.3 | 85.3 | 217.5 | 10.4 | 0.7 | 33.6 | 9.7 | 34.3 |
| | 5% Pilot | 7.7 | 82.4 | 240.9 | 10.4 | 0.9 | 34.2 | 9.5 | 35.1 |
| | 10% Pilot | 8.0 | 81.8 | 78.8 | 18.8 | 9.3 | 33.9 | 9.5 | 43.2 |
| E20B4 | Main | 9.8 | 79.6 | 192.9 | 10.4 | 1.1 | 34.7 | 9.3 | 35.8 |
| | 5% Pilot | 4.8 | 81.7 | 207.2 | 10.4 | 1.0 | 33.6 | 9.4 | 34.6 |
| | 10% Pilot | 8.7 | 81.5 | 70.3 | 18.8 | 8.8 | 34.4 | 10.0 | 43.2 |

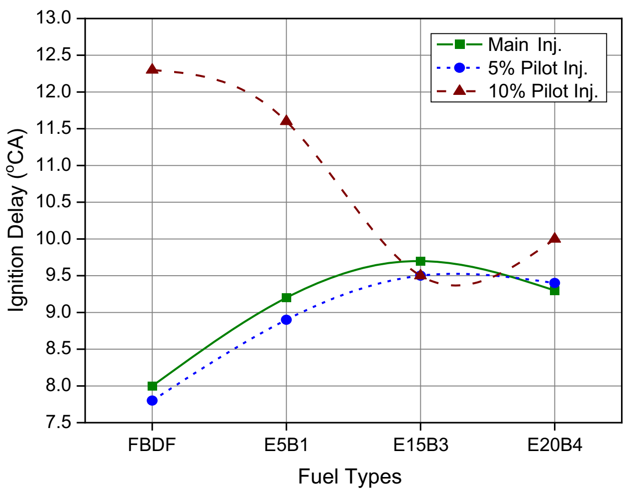


**Fig. 3** Effect of pilot injection strategies on ignition delay

FBDF fuel at main spraying type and 5% pilot spraying type, while in 10% pilot injection stages, and the longest ignition delay was detected with FBDF. While the longest ignition delay between blend fuels in the 10% pilot injection stage was observed in E5B1 fuel use, in the main injection application and in 5% the pilot injection stage the longest ignition delay time was determined in the use of E15B3 fuel. The addition of ethanol to FBDF, the cetane number decreases, but the oxygen content increases, so it is thought that the ignition delay times are prolonged, and the combustion duration shortened in the use of fuel mixtures. Similar ignition delay trends have been reported in the literature by some researchers adding various additives to ethanol-FBDF mixtures [36, 37]. However, when compared to use of FBDF, it was seen that the total combustion duration was shortened with use of blend fuels that similar result was stated by Sarkar et al. [38].

### 3.1.2 Cylinder Gas Pressure and Heat Release Rate

As known, cylinder gas pressure is highly related to combustion duration in the cylinder, lower heating value of used fuel and heat release rate [15]. The released energy in the engine increases because of increasing the fuel flow rate or the energy content of fuel [39]. Figure 4 shows the effect on the cylinder gas pressure values by using ethanol-FBDF blends and neat FBDF fuels under different injection applications. At the main injection application with use of FBDF, the $CP_{max}$ was determined as 80 bar, while it was measured as 78 bar with a slight decrease at 5% pilot injection application and it was measured as 85.6 bar with 5.7 bar increase in the $CP_{max}$ at 10% pilot injection type. $ACP_{max}$ (point of angle of maximum cylinder pressure) is given in Table 5 for each fuel.

In case of E5B1 fuel use, the highest $CP_{max}$ was realized as 82.4 bar with 10% pilot injection, while the lowest $CP_{max}$ was achieved as 79.5 bar with 5% pilot injection. Compared to the main injection applications, a decrease in the use of E15B3 fuel was observed with 10% pilot injection application. In E20B4 fuel use, when compared to the main injection application, the $CP_{max}$ value in the 5% pilot injection type increased by 2.1 bar and was determined to be 81.7 bar, while in the pilot 10% injection type, the increase in the cylinder gas pressure value was 81.5 bar with a slight decrease. In general, due to the high oxygen content of the blended fuels, the pressure values of the blended fuels and cylinder gas were found to be higher in the main and 5% pilot injection applications. Similar results were reported in some studies [16, 33, 40].

At the same time, Fig. 4 illustrates the effect of pilot injection stages on the heat release rate. With neat FBDF, the heat release rate in the main injection type was determined as 100.6 j, while it was achieved as 131.5 j with an increase of

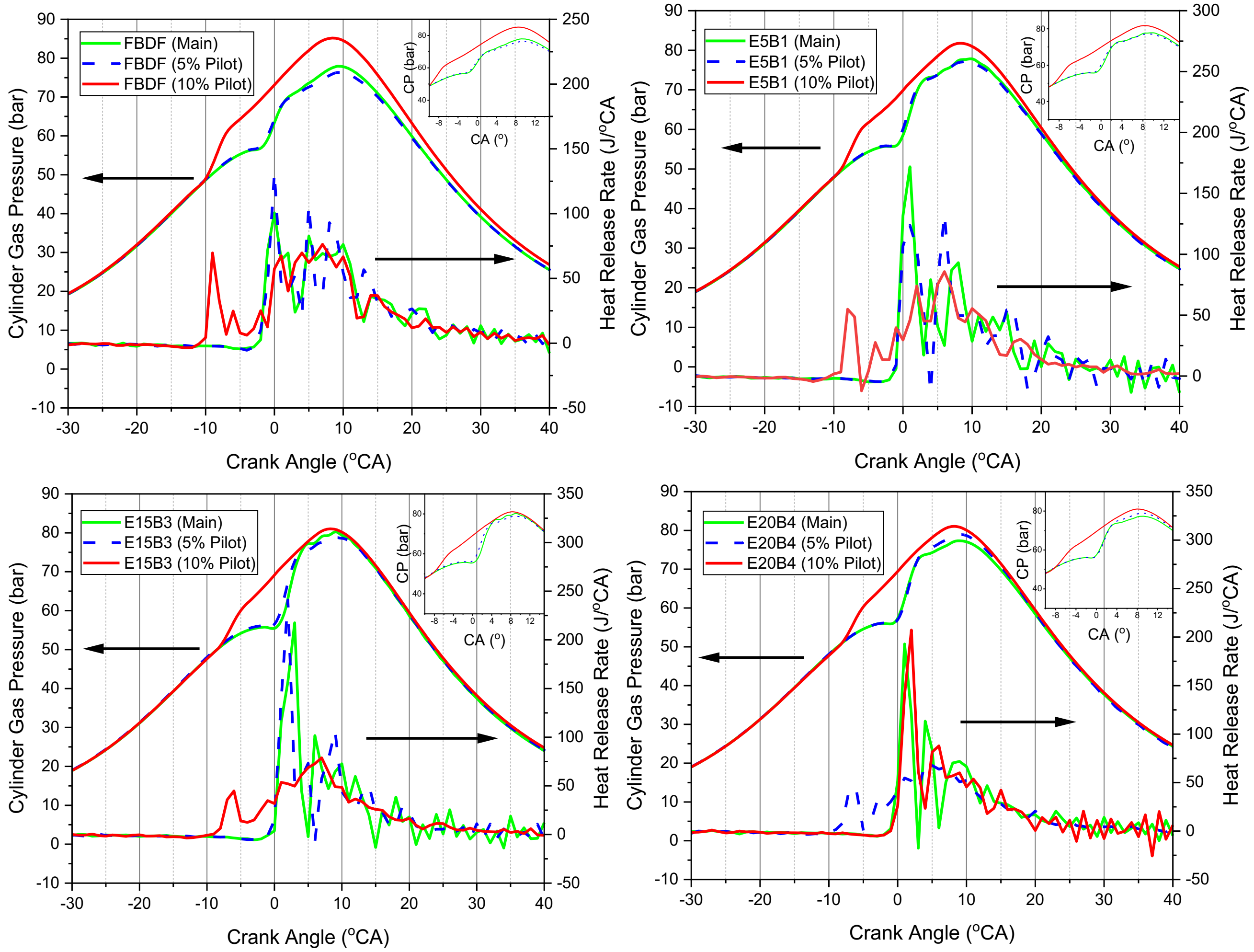


**Fig. 4** Effect of pilot injection stages on cylinder gas pressure and heat release rate

approximately 30% in the use of 5% pilot injection and 76.2 j with a decrease of approximately 24% with a 10% pilot injection application.

With E5B1 fuel, the highest $HRR_{max}$ values were determined in the main injection type, while a decrease was observed in 5% and 10% pilot injection types. With E15B3 and E20B4 fuels, maximum heat release rates were obtained as 240.9 j for E15B3 fuel and 207.2 j for E20B4 fuel at 5% pilot injection application, and the lowest $HRR_{max}$ was found to be 78.8 j for E15B3 fuel and 70.3 j for E20B4 fuel. While it was observed as 217.5 j in the main injection type with E15B3 fuel, it was found as 192.9 j in the main injection application in the use of E20B4 fuel. In general, the higher $HRR_{max}$ was obtained due to the oxygen content of blend fuels, lower cetane number and longer ignition delay.

### 3.1.3 Combustion Noise

With the spontaneous combustion of the fuel in the cylinder, combustion noise occurs with the sudden increase in cylinder gas pressure [41]. Figure 5 shows the effects of alcohols-FBDF fuel blends and different injection applications on combustion noise. While the minimum combustion noise in main injection use was measured as 89.4 dB with FBDF, the maximum combustion noise was measured as 94 dB with an increase of approximately 5% in E15B3 fuel use. In the main injection application, when using E5B1 and E20B4 fuels, the combustion noise values are close to each other and higher than the use of FBDF fuel. Compared to the combustion noise measured as a result of the use of main injection with FBDF, a slight decrease in combustion noise was observed with 5% and 10% pilot injection use.

With the E5B1, the combustion noise was measured as a result of the application of the main injection strategy and the combustion noise obtained as a result of the 5% pilot injection

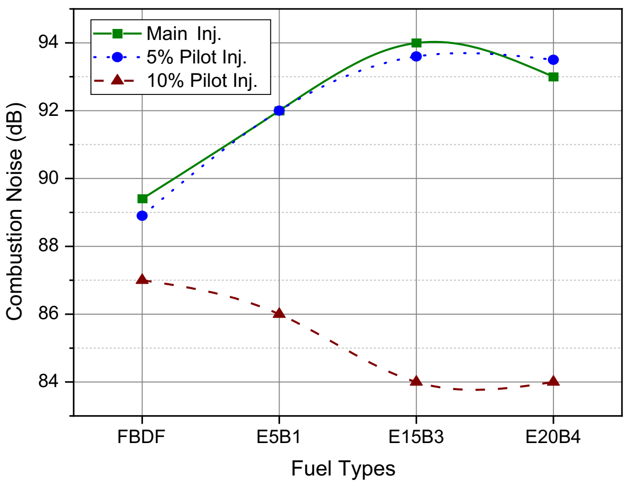


**Fig. 5** The effect of pilot injection stages on combustion noise

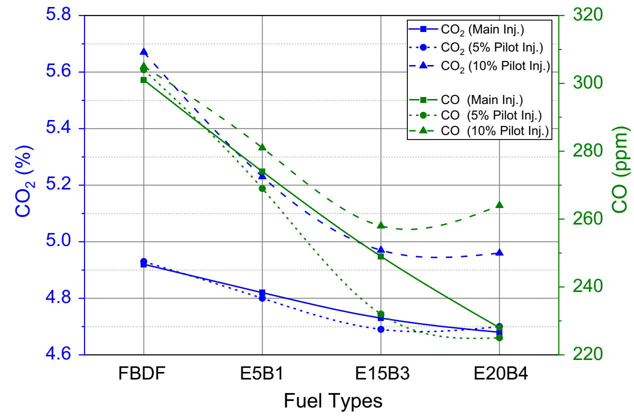


**Fig. 6** The effect of pilot injection stages on CO and $CO_2$ emission

application was measured equally, while it was measured as 86 dB by providing more than 5% reduction in the 10% pilot injection application. With E15B3 and E20B4 fuels, the values measured with 5% pilot injection were very close to each other and equal to 84 dB at 10% pilot application. Due to the low cetane number and high oxygen content of the mixture fuels, it causes a sudden increase in pressure and an improvement in cylinder combustion, which is reflected in an increase in cylinder gas pressure and heat release rate. In the literature, similar trends were showed by researchers [42, 43].

## 3.2 Exhaust Emission Results

### 3.2.1 Carbon Monoxide (CO) and Dioxide (CO2) Emissions

Figure 6 indicates its effect on CO and $CO_2$ emission characteristics by using alcohol fuel blends and FBDF fuel under different pilot injection strategies. Comparing all fuel types in the main injection application, the maximum CO emission was found to be 301 ppm with FBDF, while it was found to be 274 ppm, 249 ppm and 228 ppm (E5B1, E15B3 and E20B4, respectively). Considering the values in the main injection application, a decrease in CO emissions was detected as the alcohol fuel ratio in fuel mixtures increased.

Compared to the main injection application, there was an increase in CO emissions for FBDF in the pilot application of 5%, while a decrease in CO emissions for fuel blends was found in the 5% pilot application. When all injection applications are compared, maximum CO emission values for all fuel types are seen in the 10% pilot injection application. With 10% pilot injection, the maximum CO emission value was determined as 305 ppm with FBDF, while it was 281 ppm for E5B1 fuel, 258 ppm for E15B3 fuel and 264 ppm for E20B4 fuel. Due to the oxygen content of the blended fuels, it is thought that in-cylinder combustion has improved, and therefore a decrease in CO emissions was observed.

Compared to neat FBDF in the main injection application, there was a slight reduction in $CO_2$ emissions with the use of blend fuels. Compared to the main injection application, an increase in $CO_2$ emissions was observed in the 5% pilot injection application for FBDF and E20B4 fuel, while a decrease was observed in the 5% pilot injection application for the E5B1 and E15B3 fuel.

The maximum $CO_2$ emission value for each fuel type was seen in 10% pilot injection practice. In the 10% pilot injection application, $CO_2$ emission values were 56.742 ppm for FBDF, 52.380 ppm for E5B1, 49.709 ppm for E15B3 and 49.618 ppm for E20B4 fuel. Compared to FBDF, a decrease in $CO_2$ emissions was observed as the ethanol ratio in blend fuels increased for 10% pilot injection. It was determined that the values obtained are higher due to the C/H ratio of FBDF fuel compared to the blended fuels.

### 3.2.2 Ammonia (NH3) and Nitrogen Oxide (NOx) Emissions

In internal combustion engines, the formation of $NH_3$ emission depends on the temperature level in the combustion chamber and the amount of hydrogen [44]. Figure 7 shows the use of different fuel types and the effect of pilot injection applications on $NH_3$ and $NO_x$ emissions. For FBDF, when the main injection application and the 5% pilot and 10% pilot injection applications are compared, there is a decrease of approximately 5% in the $NH_3$ emission in the 5% pilot injection application compared to the main injection application, and an increase of approximately 5% in the $NH_3$ emission in the 10% pilot injection application.

In all fuel types and injection strategies, minimum $NH_3$ emission was 0.0516 ppm in 10% pilot injection application with E15B3 fuel. The maximum $NH_3$ emission for E5B1 fuel was found to be 0.09 ppm with 10% pilot injection, with an increase of more than 50% compared to main injection. While

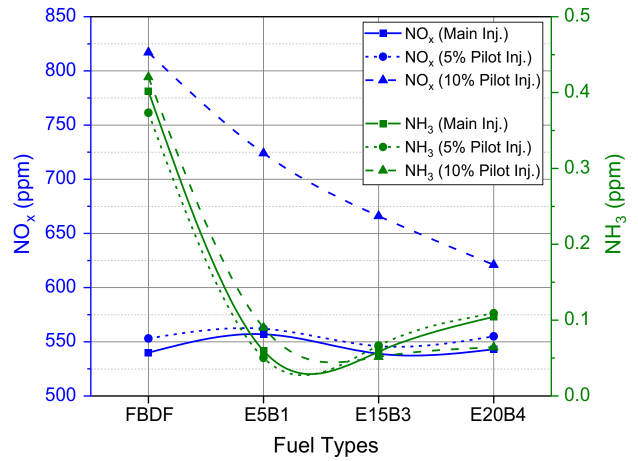


**Fig. 7** The effect of pilot injection stages on $NH_3$ emission and $NO_x$ emission

**Table 6** Exhaust temperatures and pressures fuels for different injection characteristics

| Fuels | Injection stages | Exhaust temperature (°C) | Exhaust pressure (mbar) |
|---|---|---|---|
| FBDF | Main | 324.5 | 11.8 |
| | 5% Pilot | 328.6 | 13.1 |
| | 10% Pilot | 360.1 | 14.8 |
| E5B1 | Main | 317.3 | 11.4 |
| | 5% Pilot | 316.2 | 11.4 |
| | 10% Pilot | 336.2 | 13.5 |
| E15B3 | Main | 307.2 | 9.9 |
| | 5% Pilot | 304.9 | 10.3 |
| | 10% Pilot | 321.6 | 11.5 |
| E20B4 | Main | 302.6 | 9.2 |
| | 5% Pilot | 299.2 | 10.8 |
| | 10% Pilot | 316 | 11.2 |

$NH_3$ emission value was determined as 0.0584 ppm with E15B3 fuel with the main injection application, an increase more than 10% was observed in the $NH_3$ emission in the 5% pilot injection application, but a decrease more than 10% was observed in the $NH_3$ emission in the 10% pilot injection application. While $NH_3$ emission values obtained by main injection and 5% pilot injection application with E20B4 fuel were found close to each other, a reduction of more than 30% was observed in 10% pilot injection application compared to the main injection application. Dhahad et al. [45] reported that $NH_3$ formation depends on the combustion temperature in the combustion chamber.

As known, formation of $NO_x$ emissions is dependent on combustion temperature, oxygen concentration and cylinder pressure [46]. With the main injection method, the $NO_x$ emission value in the use of E15B3 fuel was found to be very close to the $NO_x$ emission value released because of the use of neat FBDF fuel, while the use of E5B1 and E20B4 fuels were at higher $NO_x$ emission values than FBDF fuel. In a 5% pilot injection application, the maximum $NO_x$ emission was achieved as 562 ppm with E5B1 fuel, while the minimum $NO_x$ emission was achieved as 546 ppm with E15B3 fuel. In the 5% pilot injection application, it was observed that the $NO_x$ emission values emitted by FBDF and E20B4 fuels were very close to each other. In addition, exhaust temperature and exhaust pressure are given in Table 6.

Maximum $NO_x$ values for all fuels were found at 10% pilot injection. As a result of spraying fuels with 10% pilot injection, the maximum $NO_x$ emission was determined as 817 ppm with FBDF, while the increase in the ethanol ratio in the mixture fuels, the $NO_x$ emissions decreased, and the minimum $NO_x$ emission was 621 ppm with E20B4 fuel in the 10% pilot injection application. Higher $NO_x$ values were obtained with the blended fuels at the application of main injection since the oxygen content in the mixture fuels improves the in-cylinder combustion and increases the temperature inside the cylinder.

## 4 Conclusion

In this study, in a single cylinder diesel engine, the 5% and 10% pilot injection stages at compression stroke of a diesel engine using FBDF, E5B1, E15B3 and E20B4 were applied at 1600 rpm constant engine speed and 50% constant engine load.

- It was observed that butan-2-ol added ethanol-FBDF mixtures showed a good solvent property and extended the homogeneity time in all fuel mixture types.
- Compared to the use of neat FBDF, the ID increased while the DOC decreased with the increase in the amount of alcohol fuel in the blends. In addition, an increase in the ID and DOC were observed with the increase in the amount of pilot injection.
- In the main and 5% pilot injection applications, as the alcohol content in the mixture fuels increases, the combustion noise increases, while in the 10% pilot injection application, since more controlled combustion is provided, a reduction in combustion noise up to 10% is achieved.
- While $CP_{max}$ values were obtained earlier and higher than FBDF fuel with fuel mixtures, the highest $CP_{max}$ was obtained in the use of FBDF in 10% pilot injection application. As a result of the use of fuel blends, a significant increase was observed in the $HRR_{max}$ values in the Main and 5% pilot injection applications. However, in 10% pilot injection application, there was a decrease in $HRR_{max}$ values in all fuel types.

– Generally, a significant reduction in CO and $CO_2$ emissions were observed as the ethanol and butan-2-ol ratio in blended fuels increased, compared to FBDF fuel. The highest CO and $CO_2$ emissions in all fuel types were detected in 10% pilot injection applications
– In all injection applications, maximum $NH_3$ emission was obtained in the use of FBDF, while $NH_3$ emission reduction was observed up to 12 times in the use of fuel blends. When the blend fuels were compared among themselves, it was determined that the $NH_3$ emission increased as the alcohol content increased. While it was observed that the $NO_x$ emission values emitted in the Main and 5% pilot injection applications were close to each other, an increase in emissions up to 25% was observed in the 10% pilot injection application.

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**Authors Contributions** Mustafa Vargün contributed to investigation, methodology, data processing, writing—original draft. Ahmet Necati Özsezen contributed to writing—review and editing, conceptualization, visualization, supervision, project administration.

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## Declarations

**Conflict of interest** The authors declare that they have no competing interests.

**Ethical Approval** The results presented in the manuscript were obtained from our original research. We certify this paper has not been published, is not under consideration for publication elsewhere, nor will be made available for consideration for publication simultaneously elsewhere, and all co-authors have read and agreed to the submission of the manuscript.

**Consent to Participate** This study is not containing any individual person's data in any form.

**Consent to Publish** Our manuscript does not contain data from any individual person.

**Availability of data and materials** The engine test setup shown below was used in engine tests. The engine test system of AVL is installed in our engine test room. The engine tests and measurements made within the scope of this study were carried out by the authors.

We have explained this case with a sentence " While performing these tests, the power measurement was made as described in ISO 14396, additional requirements for exhaust emission tests in accordance with ISO 8178." to describe the main methodologies using in the study. In addition, we gave information about the equipment and measurements outside the standards in the material and method section. The datasets used and/or analyzed during the current study are available from the corresponding author on reasonable request. All data generated or analyzed during this study are included in this published article [and its supplementary information files].

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