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TECHNICAL PAPER

# Experimental investigation of combustion and exhaust emission values in a diesel engine using ethanol-butan-2-ol-diesel fuel blends

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**Abstract**
Ethanol and petroleum-based diesel fuel (PBDF) blends have been extensively tested on diesel engines as an alternative fuel without the use of any co-solvents. However, there is not enough information about the combustion noise and products of $CO_2$, $CH_4$ and $NH_3$ that are sourced from 2-butanol used as co-solvent to prevent phase separation between ethanol and PBDF. The present study aims to investigate the effect of ethanol-butan-2-ol-diesel fuel blends on combustion phenomenon and $CO_2$, $CH_4$ and $NH_3$ emission characteristics under different operating conditions of diesel engine. In this study, homogeneity of ethanol and PBDF fuel mixture was achieved by using butan-2-ol (2-butanol). First, engine tests were conducted with the base calibration strategy of injection timing. Later, the tests were conducted by advancing and retarding injection timing to observe the effects of the injection timing on the combustion and emission at different engine speeds. The test results showed an increase in combustion noise with the use of alcohol-PBDF blends. In the tests, the maximum combustion noise was monitored as 92.7 dB with the use of E10B2 at the engine speed of 1400 rpm. It was observed that the use of alcohol-PBDF fuels was much more effective in reducing $NH_3$ and CO emissions according to the change in fuel injection timing. In the tests, 50% or more reduction in CO and $NH_3$ emissions was observed with the use of alcohol-PBDF fuel blends. It was noted that engine speed was more effective than both alcohol-PBDF blends and injection timing on the formation of $CO_2$ emissions. It was observed that delaying the fuel injection time is more effective on $NO_x$ emissions than the use of ethanol-PBDF mixture. Moreover, it was observed that increasing ethanol ratio in fuel blend led to increase in in-cylinder gas pressure due to longer ignition delay that enables more time to fuel and air to mixture. The longest ignition delay was calculated as 9.2°CA in use of E15B3 fuel at 1400 rpm.

**Keywords** Ethanol-2-butanol-diesel blends · SOI timing · Diesel engine · Combustion · Combustion noise · Exhaust emissions

## 1 Introduction

Diesel engines are used in many areas in the industry due to their high efficiency and durability. However, because of the combustion of the fossil-based diesel fuels used as fuel in diesel engines, exhaust emissions, that cause global warming and are harmful to human health, are emitted. The development of cleaner diesel engines and combustion process have been worked more extensively since the introduction of diesel engine emission standards, which are getting more stringent in time. The use of biofuels (biodiesel, ethanol, butanol) is seen as one of the important methods of limiting the use of fossil-based diesel fuels and to reduce harmful effects of exhaust emissions. In this context, according to the estimations of the International Energy Agency (IEA), the demand for biofuels is expected to increase by 28% until 2026. Among the biofuels, ethanol is one of the fuels with high usability potential in the transportation sector [1–3].

Ethanol is an alcohol which is colorless transparent liquid, and it can be produced by fermentation of sugar with ingredients of yeast, corn, and vegetables [4]. One of the

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most important advantages of alcohol fuels is their cooling effect due to their lower boiling point and the other is their oxygen content [5]. As it is known that the cetane number has a dominant effect on the ignition delay while the volatility, aromatic and oxygen contents have only a minor influence on the ignition delay [6]. Compared to fossil fuels, higher flame burning velocity of ethanol allows a faster combustion rate [7, 8], and lower cetane number of ethanol enables high compression ratio in the cylinder [9].

One of the most important disadvantages of using ethanol by mixing it with petroleum-based diesel fuel (PBDF) is phase separation that is emerging in a very short time. The main causes of phase separation are low temperature, water, and high ethanol content in the mixture [10]. In the literature, it is recommended to add solvent such as butanol and its derivations to ethanol-PBDF blends to prevent phase separation [11, 12]. In this study, the solubility of ethanol and PBDF was observed with and without the additive of 2-butanol. It was seen that no significant phase change occurred with using 2-butanol as stabilizer in ethanol-PBDF blends up to 20% percent of ethanol in blend. It was observed that the addition of 2-butanol (or butan-2-ol) to fuel mixture has a slowing effect on phase separation between PBDF and ethanol.

The change in start of injection (SOI) timing is one of the most important parameters that has an impact on combustion, performance, and exhaust emissions in a compression ignition engine. When the fuel is injected at an angle far from the top dead center (TDC), the ignition delay (ID) time increases due to the low cylinder pressure and temperature [13], but the ID time decreases when the fuel is injected at an angle close to the TDC. As a result, with the change of SOI timing, the injection characteristics change and significantly affect engine performance and exhaust emissions [14, 15]. In another study [16], it was stated that the ignition delay for low heating value gas is essentially independent from the cylinder pressure. It was stated that the ignition delay is highly dependent on changes in the engine cylinder temperature. However, Rosa et al., [17] indicated that ethanol effectively increased ignition delay and reduced combustion duration.

In literature, some researchers showed effects of injection strategies and ethanol-PBDF blends on combustion parameters of compression ignition engines. If some of these studies are to be given as an example; He et al. [18] studied a compression ignition engine by adding solvent to diesel-ethanol fuel mixtures. At high engine loads, they saw a significant reduction in smoke emissions while using blended fuels, and a slight reduction in $CO_2$ and $NO_x$ emissions. They found a significant increase in acetaldehyde emissions in the use of blended fuels compared to PBDF fuel. Prabakaran et al. [19] investigated the effects on combustion by using ethanol-butanol-PBDF fuel mixtures in a compression ignition engine, and they found an increase in ID time meanwhile observing a decrease in cylinder gas pressure in blended fuel use. While they observed a decrease in $NO_x$ emission in the use of mixed fuel, they saw an increase in CO emissions. Murcak et al. [14] examined the effect of using ethanol-PBDF fuel mixtures and SOI timing on performance in a compression ignition engine. They achieved maximum engine power at 2400 rpm engine speed and 35°CA BTDC injection starting timing using E5 fuel. Liu et al. [20] investigated the effect of change of SOI time on combustion. At the same SOI time condition, as the ethanol ratio in the mixture increased, the cylinder gas pressure decreased while the heat release ratio went up. As the ethanol ratio in the mixtures increased, the combustion time decreased while the ID time raised. Li et al. [21] investigated the effects of change in SOI timing on combustion in a compression ignition engine where they used IBE (isopropanol-butanol-ethanol) fuel and blended fuels with PBDF fuels. In all fuel types, they found an increase in cylinder gas pressure and heat release rate by advancing the injection timing from the TDC. They noticed an increase in cylinder gas pressure as the proportion of IBE in blended fuels increased when compared to PBDF fuel at the SOI timing of 18°CA BTDC. They found that as the IBE ratio in the blend fuels increased, while the ID time raised, the combustion time decreased. Park et al. [22] investigated the effect of ethanol-PBDF use and SOI timing on combustion. They found that as the ethanol ratio in PBDF-ethanol blends increased, ID time and $NO_x$ emissions increased. They also achieved an increase in cylinder gas pressure and $NO_x$ emission as the injection timing was advanced. They found an increase in CO and HC emissions as the ethanol ratio in blended fuels went up, and a decrease in CO and HC emissions by approaching the SOI timing. Recently, Rosa et al. [23] examined the effect of change in SOI time and the water content in the ethanol on combustion and exhaust emissions in a reactive controlled compression ignition engine. They found that the increase in ethanol ratio under all test conditions prolonged the ignition delay. While the highest cylinder gas pressure value was achieved with the earliest SOI timing, it was observed that taking the SOI timing earlier reduced CO emission and increased $NO_x$ emission. Zhang et al. [24] investigated the effect of methanol or ethanol injected into the intake air of a compression ignition engine on exhaust emissions. As a result of the study, they stated that alcohol fumigation is a good method to reduce the concentration of $NO_x$, particulate mass concentration and particle number in the exhaust of a compression ignition engine. Emiroglu et al. [25] used 10% by volume methanol/diesel, ethanol/diesel, butanol/diesel mixtures as test fuel at different engine loads in a single-cylinder compression ignition engine. They stated that the use of alcohol-diesel fuel mixtures caused an increase in ignition delay, a significant decrease in CO emission, and a slight increase in

$NO_x$ emission compared to pure fossil diesel fuel. In another study, Satsangi et al. [26] stated that the use of butanol-diesel fuel mixture causes higher combustion and exhaust noise than pure fossil diesel fuel at high engine loads. Addition of butanol to diesel fuel caused an increase in ignition delay. Liang et al. [27] studied the effects of different engine load, injection timing effects on combustion and exhaust emissions by adding different solvents to the ethanol–diesel fuel mixtures against the solubility problem in a diesel engine. They observed an increase in ignition delay times and a decrease in combustion duration times with the use of fuel mixtures.

In the literature, there are many studies relevant to the effects of fuel injection strategies and the use of ethanol-fossil-based diesel fuel mixtures on combustion parameters and exhaust emissions. However, there is still a deficiency in the literature studies on the examination of combustion noise, methane emission, ammonia emission and phase separation in fossil diesel-ethanol fuel blends. As mentioned above, one of the important problems affecting the use of ethanol–diesel fuel mixtures in commercial diesel engines is that the phase separation between ethanol-fossil diesel fuel takes place in a much shorter time, especially as the ethanol ratio in the fuel mixture increases. In order for ethanol to be easily used as an alternative fuel in commercial diesel engines, there should be no phase separation in the fuel tank for a long time. Studies for this purpose, various solvent added to the fuel mixtures in order to keep the fuels homogeneous for a longer time. These studies showed that n-butanol, isopropanol, esters, etc., liquids can be used as co-solvents in ethanol-fossil diesel mixtures. In particular, in the study by Jin et al. [28] indicated that butanol and its isomers as a good water-holder due to their near-water properties.

In this study, 2-butanol was used to prevent phase separation between ethanol and PBDF mixtures, the effect of SOI and fuel blends on engine power, cylinder gas pressure, heat release rate, CO, $CO_2$, $NO_x$ were examined at 1000 rpm, 1200 rpm and 1400 rpm engine speeds and 50% constant engine load in a compression ignition engine. Moreover, combustion noise, $NH_3$ and $CH_4$ (are as known unregulated emissions) are not considered in most studies that were investigated in this study. The obtained data were compared with the results obtained with conventional engine applications and neat PBDF.

**Table 2** Prepared fuel mixtures and their some fuel properties

| Test fuels and properties | PBDF | E10B2 | E15B3 | E20B4 |
|---|---|---|---|---|
| PBDF, vol. % | 100 | 88 | 82 | 76 |
| Ethanol, vol. % | – | 10 | 15 | 20 |
| 2-Butanol, vol. % | – | 2 | 3 | 4 |
| Density (kg/m$^3$, 15 °C) | 834 | 830 | 827 | 825 |
| Kinematic viscosity (mm$^2$/s, 40 °C) | 3.5 | 3.03 | 2.72 | 2.67 |
| Lower heating value (MJ/kg) | 42.6 | 40.83 | 39.96 | 39.08 |
| C% (v/v) | 85.71 | 84.51 | 83.82 | 83.06 |
| H% (v/v) | 14.29 | 14.24 | 14.22 | 14.19 |
| O% (v/v) | – | 1.25 | 1.96 | 2.75 |

## 2 Materials and methods

### 2.1 The properties of test fuels

In the study, PBDF was taken from a national fuel station in Turkey, and J.T. Baker brand of ethanol with 95% and above purity ethanol was used to prepare the fuel blends. In addition, to prevent phase separation in ethanol-PBDF blends, the 2-butanol was added 20% of the ethanol ratio in mixture. The properties of the main fuels used in tests are given in Table 1. The properties of the fuels are taken from the manufacturer database.

In the study, three different fuel mixtures were prepared. Fuels were named according to the ratio of ethanol and 2-butanol in the mixture. The content of the fuel called PBDF is neat petroleum-based diesel fuel. Used fuels content and some properties are given in Table 2.

**Table 1** Properties of PBDF, ethanol, and 2-butanol fuels

| Properties | PBDF (≈$C_{12}H_{24}$) | Ethanol ($C_2H_5OH$) | 2-Butanol ($C_4H_9OH$) |
|---|---|---|---|
| Purity | – | ≥ 0.99 | ≥ 0.99 |
| Molecular weight (g/mol) | ≈170 | 46.07 | 74.12 |
| Latent heat of vaporization (kJ/kg) [29] | ≈256 | 920 | 670 |
| Boiling point (°C) | 160 | 78 | 102 |
| Melting point (°C) | − 9.6 | − 114.5 | − 115 |
| Flash point (°C) | ≥ 55 | 12 | 20.5 |
| Water content (%) | 0,020 | ≤ 0.2 | ≤ 0.2 |
| Cetane number [30] | ≥ 51 | 8 | 15 |
| Auto-ignition temperature (°C) | ≈210 | 361 | 405 |

In this study, the phase separation was observed within 30 min in the blends prepared before butan-2-ol was added to the ethanol–diesel fuel mixtures. In the phase separation observations made after the addition of butan-2-ol to ethanol–diesel fuel mixtures, the fuels were separated from each other in 2 layers within 24 h for E20B4 fuel. In the phase separation observations made for E15B3 fuel, phase separation was observed by the fuels as 2 layers after 72 h. In the phase separation observations made for E10B2 fuel, while a slight phase separation was detected on the 17th day, it was not observed that the fuels were clearly separated from each other in a layered manner. Observation of phase separation of 17 days is given in Fig. 1. In a similar study, Huang et al. [31] stated that adding n-butanol to diesel-ethanol fuel blends prevented blends fuel from phase separation up to 11 days. Jin et al. [28] reported that n-butanol is useful to the mutual solubility of diesel and water, and it also supports the water tolerance of the IBE (isopropanol-butanol-ethanol).

## 2.2 Engine test setup and conditions

In this study, engine tests were performed with a single cylinder, 4-stroke, and common rail fuel injection system under at different engine speeds (1000 rpm, 1200 rpm and 1400 rpm) with constant engine pedal position (50%). The engine specifications are given in Table 3.

The main components of the engine test bench are an eddy current dynamometer, fuel injection system (including common rail, injector, fuel filter and fuel pump) and engine control unit (ECU). The engine oil and cooling fluid conditioning are stabilized by the engine test bench. Therefore, parasitic loads, other than fuel pump, are eliminated. The schematic view of the test cell and test engine is given in Fig. 2.

Engine test cell works fully integrated with the dynamometer and parts of test bench, and all systems are operated with a controller. Injection start time can be controlled by the driver system that allows map change by connecting to ECU. Since the engine ECU is open to the user, the start of injection, the main injection amount and the rail pressure map can be altered real time, 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. Used devices in the experimental setup are given in Table 4 with their accuracy.

**Table 3** 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 | 16.14:1 |

During the operation of tests, the power measurement was made as described in ISO 14396, by considering the test environment and measurement methods defined in ISO 8178. Firstly, PBDF was used in engine to have stable conditions until the engine oil temperature was 90 °C. In the experiments, air intake temperature and pressure, fuel temperature, cooling water temperature were fixed at 25 °C and 240 mbar, 20 °C, 70 °C, respectively.

Moreover, data were collected at each 0.1°CA, and an average of 50 cycles were considered during calculation of combustion characteristics. The coefficient

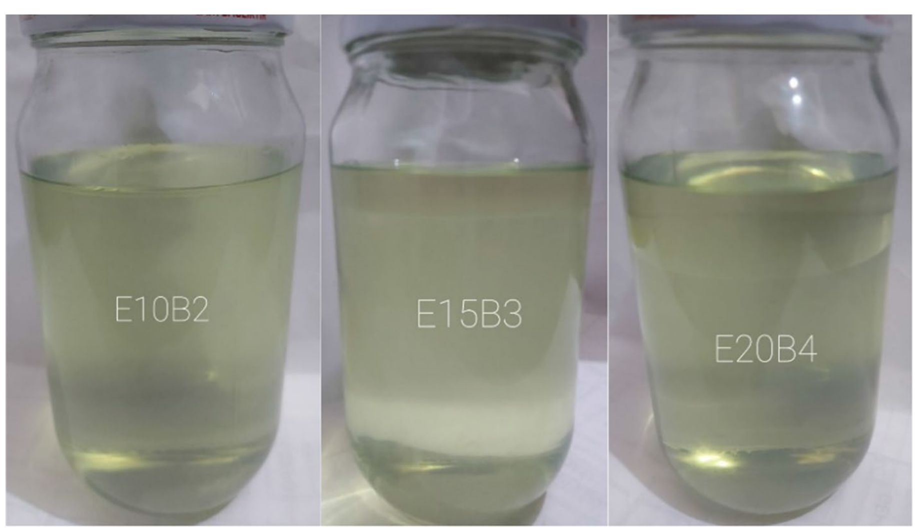


**Fig. 1** Observation of phase separation of blends with adding butan-2-ol (17th day)

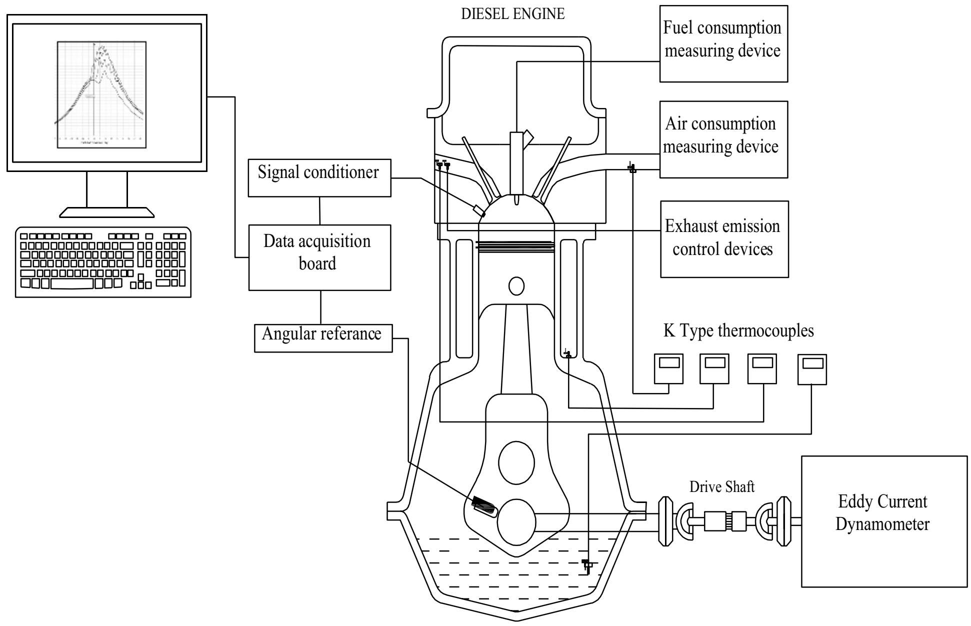


**Fig. 2** Schematic view of the experiment setup

**Table 4** Accuracy of used devices

| Measurement | Device | Accuracy |
|---|---|---|
| Torque | HBM torque flange | ±0.1% |
| Engine speed | AVL Encoder | ≤±0.1°CA |
| Test cell humidity and temperature | Vaisala – HMT 330 | ±1% RH, ±0.2 °C |
| In-cylinder pressure | AVL-GU22C | ±0.3 bar |
| Injection timing | Angle Encoder | ±0.1°CA |
| Engine coolant and oil conditioning | AVL-577 | ±1 K |
| Fuel consumption | AVL-735 | <0.15% |
| CO, $CO_2$, $NH_3$, $CH_4$, $NO_x$ | AVL AMA i60 | ≤±1% |
| Temperature sensors | PT100 (K Type) | ≤±1 K |

of variation (COV) of indicate mean effective pressure (IMEP), $COV_{IMEP}$ values for all engine operation conditions was calculated according to Eq. (1) [32] to check stability of combustion in each cycle. In the literature, $COV_{IMEP}$ values below 3% are accepted as very good combustion stability for low engine speed condition [33]. The $STD_{IMEP}$ represent the standard deviation of the IMEP value of the 50 cycles for each the in-cylinder pressure measurement. In this study, since the air/fuel ratio, engine speed and the amount of fuel injected in each cycle were fixed, and the $COV_{IMEP}$ values were calculated to be quite low. The highest calculated $COV_{IMEP}$ value was calculated as 0.13% with E15B3 at 1000 rpm.

$$COV_{IMEP}(\%) = \frac{STD_{IMEP} \times 100}{IMEP} \tag{1}$$

In this study, AVL IndiCom and Concerto software applications were used to calculate the combustion parameters; such that start of combustion, combustion duration and combustion noise. The AVL IndiCom software enables the calculation of combustion noise from the in-cylinder pressure data. The combustion noise was calculated by a power spectrum in the cylinder pressure signal, where the reference level is the human hearing threshold of 20 μPa [34]. The normal Fourier transformation is followed by a conversion to a third-octave spectrum. The spectral lines then define the mean signal level in dB within each third-octave band. The root-mean-square (RMS) value of the filtered pressure, $P_{RMS}$, was calculated, and the final noise level was obtained by comparing $P_{RMS}$ to a reference sound level [35].

$$\text{Combustionnoise (dB)} = 20 * \log_{10}\left(\frac{P_{RMS}}{20\mu Pa}\right) \tag{2}$$

In this study, 45 mg fuel was injected into cylinder for each cycle. In tests, first, fuels were injected at main

**Table 5** Fixed engine input parameters during the experiments

| Input parameters | Unit | Value |
|---|---|---|
| Engine speeds | rpm | 1000, 1200, 1400 |
| Fuel blends | (v/v) % | E10B2 E15B3, E20B4 |
| Engine load | % | 50 |
| Engine coolant temperature | °C | 70 |
| Engine oil temperature | °C | 90 |
| Intake air pressure | kPa | 24 |
| Intake air temperature | °C | 25 |
| Total injected fuel amount | mg/cycle | 45 |
| Start of main injection time at 1000 rpm | °CA BTDC | 5.8 |
| Start of main injection time at 1200 rpm | °CA BTDC | 7.7 |
| Start of main injection time at 1400 rpm | °CA BTDC | 9.1 |
| Change of injection time | °CA | ±2 |

injection start time according to engine speed then injection start time was changed ± 2°CA for each engine speed. Test conditions and test parameters are summarized in Table 5. In conclusion, result of experimental study, in-cylinder gas pressure, heat release rate, power, combustion noise, CO, $CO_2$, $CH_4$, $NH_3$ and $NO_x$ emissions were compared to base on test data SOI timing and fuel type, according to conventional SOI time and PBDF.

### 2.3 Heat release rate (HRR) analysis

In the single zone combustion model, the heat release rate calculated according to the first law of thermodynamics (Eq. 3), it is accepted as an open system [36]. In this study, the energy output in the cylinder at each crank angle was determined by using the instantaneous in-cylinder gas pressure and volume which was used in the heat release calculation as shown in Eq. 4.

$$\mathrm{d}Q_{\mathrm{net}} = \mathrm{d}W + \mathrm{d}U \tag{3}$$

$$\mathrm{d}Q_{\mathrm{net}} = \left[\left(\frac{\gamma}{\gamma-1}\right)P\mathrm{d}V + \left(\frac{1}{\gamma-1}\right)V\mathrm{d}P\right] \tag{4}$$

where; $\mathrm{d}Q_{\mathrm{net}}$ is the net rate of heat release, d$W$ is boundary work due to piston displacement (Nm), d$U$ is change in sensible internal energy (Joule), $\gamma$ is the ratio of specific heat, $P$ is cylinder gas pressure (Pa), $V$ is cylinder volume ($m^3$). Another factor to consider here is the losses through the cylinder wall. To determine the gross heat release rate ($Q_{\mathrm{gross}}$ or HRR) (Eq. 5), it is also necessary to calculate the heat transfer rate in the cylinder wall ($Q_{\mathrm{losses}}$). In this study, the gross heat release rate is calculated from Eq. 5.

$$\mathrm{d}Q_{\mathrm{gross}} = \mathrm{d}Q_{\mathrm{net}} + \mathrm{d}Q_{\mathrm{losses}} \tag{5}$$

In this study, the convective heat transfer coefficient is calculated using the Eichelberg's correlation [37] as shown in Eqs. (6) and (7).

$$T_{\mathrm{gas}} = PV/(nR_u)(K) \tag{6}$$

$$h_{\mathrm{gas}} = 7.67 \times 10^{-6}(C_{\mathrm{m}})^{\frac{1}{3}}(PT_{\mathrm{gas}})^{\frac{1}{2}}, \left(\frac{\mathrm{kW}}{\mathrm{m}^2\mathrm{K}}\right) \tag{7}$$

Eichelberg's correlation has been used to calculate the heat transfer ($Q_{\mathrm{heattransfer}}$) passing through cylinder walls by convection as shown in Eq. (8).

$$Q_{\mathrm{heattranfer}} = h_{\mathrm{gas}}A_{\mathrm{wall}}\left(T_{\mathrm{gas}} - T_{\mathrm{wall}}\right) \tag{8}$$

where: $n$ is the number of moles of the working gas (mol), $R_u$ is the universal gas constant (J/(mol-K), $h_{\mathrm{gas}}$ is convective heat transfer coefficient (W/$m^2$K), $C_{\mathrm{m}}$, is the mean-piston-speed (m/sec), $A_{\mathrm{wall}}$ is cylinder wall surface area ($m^2$), $T_{\mathrm{gas}}$ is the mass-averaged gas temperature in the cylinder ($K$), $T_{\mathrm{wall}}$ is the wall surface temperature ($K$). The wall temperature is accepted as constant (400 K) and uniform [38]. Heat transfer by radiation is neglected in this study.

### 2.4 Uncertainty analysis

In this study, the uncertainty analysis using Eq. (9) was performed to define the level of goodness of a measurement in engine tests. In an engine test considered to be stable, N consecutive measurements of an X variable (NO, $NH_4$, etc.) was used to define the uncertainty values [39]. Table 6 shows experimental measurement uncertainties. Also, the mean value $\overline{X}$ was calculated from Eq. (10) to describe the standard deviation (*STD*) which is an indicator of the width of the distribution of the $X$ values.

$$\overline{X} = \frac{1}{N}\sum_{i=1}^{N} X_i \tag{9}$$

**Table 6** The experimental uncertainties

| Measuring value | Uncertainties (%) |
|---|---|
| Engine speed | < 1 |
| Temperature | < 1 |
| CO | 2.84 |
| $CO_2$ | 1.1 |
| $CH_4$ | 3.1 |
| $NH_3$ | 4.6 |
| $NO_x$ | 2.07 |

$$STD = \left( \frac{1}{N-1} \sum_{i=1}^{N} (X_i - \overline{X})^2 \right)^{1/2} \quad (10)$$

# 3 Result and discussion

## 3.1 The effects of blends on combustion characteristics

As can be seen in Fig. 3, when compared with neat PBDF under all test conditions, an increase in ID times was detected in the use of blended fuels. The maximum ID time was seen in the use of E15B3 fuel at 1400 rpm at the 11.1°CA BTDC SOI time. The minimum ID time was observed at 1000 rpm engine speed at the 3.8°CA BTDC SOI time in PBDF use. When the mixed fuels are compared among themselves, the minimum ID time was observed in the use of E15B3 fuel at 1000 rpm and 1200 rpm engine speed, while the minimum ID time at 1400 rpm was detected in the use of E20B4 fuel.

Compared to neat PBDF values, the increase in ignition delays due to the use of fuel blends can be explained by the low cetane number and high latent heat of vaporization properties of alcohol fuels. While the low cetane number causes the fuel to ignite more difficult, the high latent heat of vaporization causes the fuel to absorb more heat from the environment as it evaporates, so the temperature in the cylinder decreases and the ignition delay time increases. It is clearly seen that the ignition delay increases as the engine speed increases, since the injection start time occurs earlier and the temperature in the cylinder is lower.

Total combustion durations calculated for each fuel under all test conditions are shown in Fig. 4. Maximum combustion duration at all engine speeds was obtained with use of neat PBDF. For PBDF, an increase in combustion duration was detected at all engine speeds by removing the SOI timing from the TDC. When compared to neat PBDF, a decrease in combustion duration was observed in the use of blends. Minimum combustion duration at all SOI times and engine speeds were determined when using E15B3 fuel. When the blended fuels are compared with each other, the maximum combustion duration was obtained with E20B4 fuel at 1200 and 1400 rpm engine speed, while the maximum combustion duration at 1000 rpm engine speed was obtained with the use of E10B2 fuel.

In general, for all test fuels, there was an increase in combustion duration as engine speed increased since at high engine speed SOI is earlier than low engine speed. As a result of this, temperature in cylinder is lower so it takes longer time to burn fuel. It is thought that due to the oxygen content of the mixture fuels, it accelerates the combustion and thus the combustion in the cylinder is shorter than PBDF for this reason. It is expected for higher oxygen content led the ignition of fuel easier. Kaya et al. [40] reported that the start of combustion occurs earlier with oxygenated fuels and hence shorter combustion duration.

### 3.1.1 Cylinder gas pressure and brake power

In Fig. 5, the effect of ethanol-butan-2-ol-PBDF use on cylinder gas pressure at different engine speed and injection start times are given. When all test conditions are compared, the highest maximum cylinder gas pressure was obtained as 85.4 bar at 1000 rpm engine speed at 3.8°CA BTDC SOI

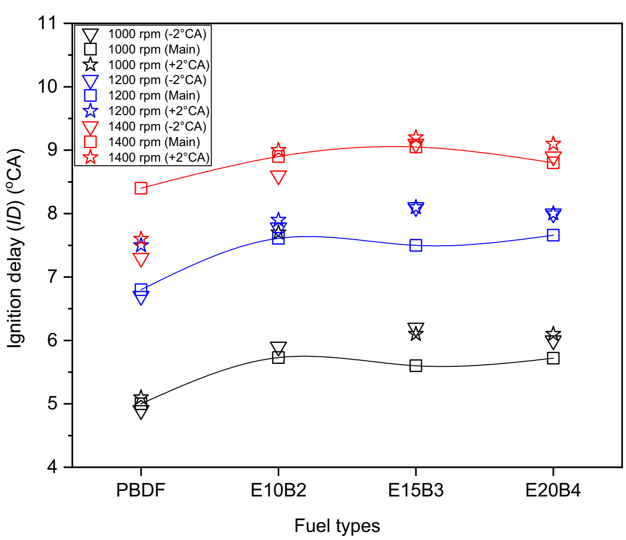


**Fig. 3** Effect of fuel blends and SOI time on the ignition delay (ID) values

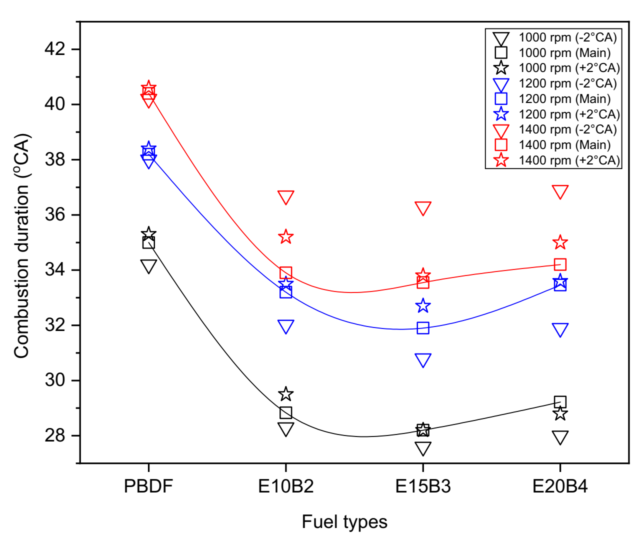


**Fig. 4** Effect of fuel blends and SOI time on the total combustion duration

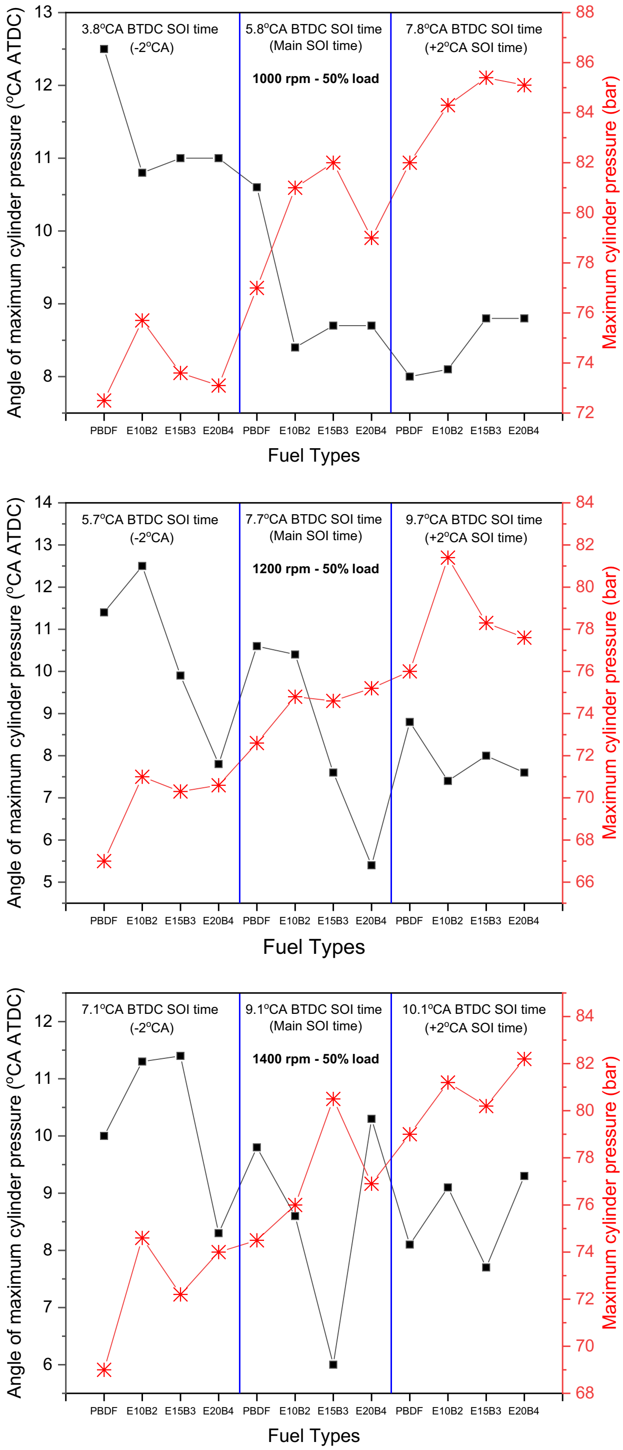


**Fig. 5** Effect of fuel blends and SOI time on cylinder gas pressure

time with E15B3 fuel use. While the increase in cylinder gas pressures was observed by advancing the SOI time for all fuels, a decrease in cylinder gas pressures was detected by retarding the SOI time. Compared to PBDF fuel, an increase in the maximum cylinder gas pressures obtained with blended fuels at the same engine speed and SOI times.

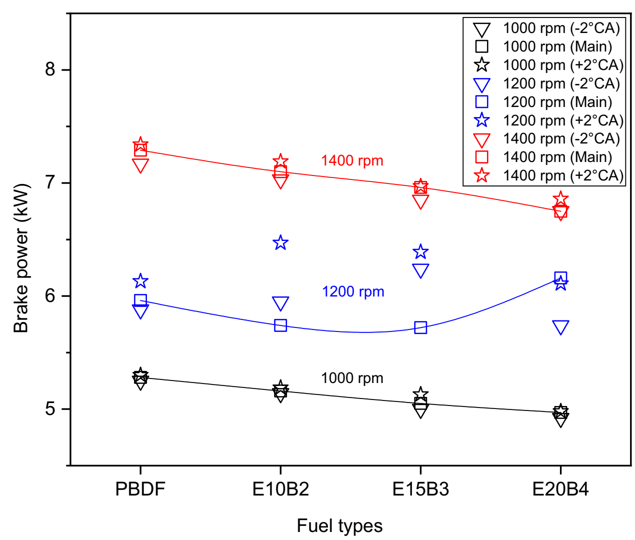


**Fig. 6** Effect of fuel blends and SOI time on power

It is thought that this increase is seen due to the fact that the oxygen molecules contained in the mixture fuels make the fuel burn better in the cylinder.

Considering all test conditions, the lowest maximum cylinder gas pressure was obtained at 1400 rpm engine speed as 69 bar at 7.1°CA BTDC SOI time. Comparing the cylinder gas pressure values obtained at 1200 rpm engine speed and 9.7°CA BTDC SOI time, the highest maximum cylinder gas pressure value was obtained as 81.4 bar in E10B2 fuel use while it was obtained as 78.3 bar for E15B3 fuel, 77.6 bar for E20B4 fuel and 76 bar for PBDF fuel use. In the tests performed at 1400 rpm engine speed, the highest maximum cylinder gas pressure value was determined as 82.2 bar at 11.1°CA BTDC SOI time in E20B4 fuel use. The reason why the injection start times are delayed at the same engine speed and the increase in cylinder gas pressures can be explained by the fact that there is more time for the fuel–air mixture to mix homogeneously and the maximum pressure is obtained around the TDC.

In Fig. 6, the effect of ethanol-butan-2-ol-PBDF use on engine power at different engine speed and SOI times is given. As stated earlier, 45 mg/stroke fuel was injected into the cylinder in each cycle at engine tests. There was a change in the amount of energy released as a result of the combustion of 45 mg of fuel with the addition of alcohol fuel to the fuel mixtures. Consequently, slight variations in engine power occurred for each test. As expected, while the maximum engine power was obtained at 1400 rpm for all fuel types and SOI times, the minimum engine power was obtained at 1000 rpm.

Maximum engine power was obtained as 7.34 kW (11.1°CA BTDC) by advancing the SOI timing by 2°CA from TDC according to main SOI time at 1400 rpm in PBDF fuel use. While the increase in engine power was observed by advancing the SOI time from the TDC for each fuel at 1400 rpm, a decrease in engine power was detected as the ethanol-butan-2-ol ratio in the fuels increased.

In the tests performed at 1200 rpm, the maximum brake power was obtained as 6.47 kW in E10B2 fuel use by advancing the SOI time from the TDC, while the minimum brake power was obtained as 5.72 kW with E15B3 fuel at the 7.7°CA BTDC SOI time. The lowest brake power was observed as 4.92 kW at 1000 rpm by approaching the SOI time to TDC (3.8°CA BTDC). At 1000 rpm, as the ethanol-butan-2-ol ratio in fuels increased, a decrease in engine power was observed during the entire SOI time. As it is well known the heating content of the ethanol is lower than that of PBDF, and this case leads to declination of the heating value of blends. As a result, engine power decreases with increase in ethanol content in mixtures. Some studies [41] in the literature showed that no significant difference in the engine power was observed with use of ethanol–diesel mixtures, but some studies [42] presented similar findings to the results of this study.

#### 3.1.2 Combustion noise

Combustion noise occurs with the sudden increase in cylinder gas pressure after sudden spontaneous combustion of the fuel in the cylinder, and the combustion noise is a main environmental concern, mainly in urban areas, influencing many people [33, 43]. The effect of ethanol-butan-2-ol-PBDF blends use on combustion noise at different engine speed and SOI time is shown in Fig. 7. When the combustion noise was examined, it was determined that the combustion noise increased for each fuel type as the engine speed increased. Maximum combustion noise at all fuel type and injection start time was determined as 92.7 dB at 1400 rpm in E15B3 fuel type by advancing the SOI time (11.1°CA BTDC) from the TDC. At 1000 rpm, it was observed that the combustion noises measured with PBDF at all injection start times were lower than the combustion noises obtained with mixed fuels.

Minimum combustion noise was obtained as 84.4 dB with PBDF at 1000 rpm at 5.8°CA BTDC SOI timing. At 1200 rpm, the combustion noise values measured with mixed fuels, the last ones were obtained very close to each other at all SOI times and higher than PBDF fuel. At 1200 rpm, the minimum combustion noise was measured as 86.8 dB with PBDF by approaching the SOI time 2°CA to TDC, according to main SOI time (7.7°CA BTDC). As a result of the experiments, it is clear that there is an increase in combustion noise values with the use of alcohol-PBDF fuel mixtures. The cetane number of the fuels decreases with the addition of alcohol fuels to PBDF, due to the low cetane number and high oxygen content of the mixture fuels, it causes a sudden pressure increasing in the cylinder and higher combustion noise occurs [44].

### 3.2 The effects of fuel blends on exhaust emission values

#### 3.2.1 Carbon monoxide (CO) emission

Even diesel engines work under lean operation conditions, excessive oxygen in the combustion chamber cannot burn all carbon atoms in the fuel chain [33]. CO emission emerges as an incomplete combustion product due to oxygen deficiency at the time of combustion. As it is known, it can kill people by replacing oxygen in the blood vessels [33]. In Fig. 8, the

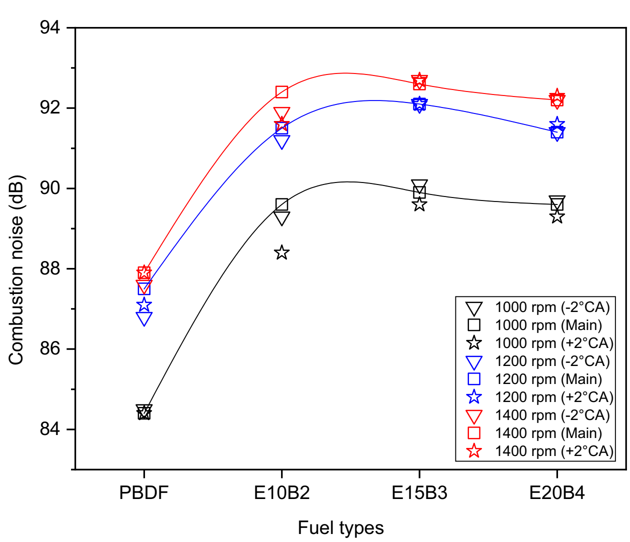


**Fig. 7** Effect of fuel blends and SOI time on combustion noise

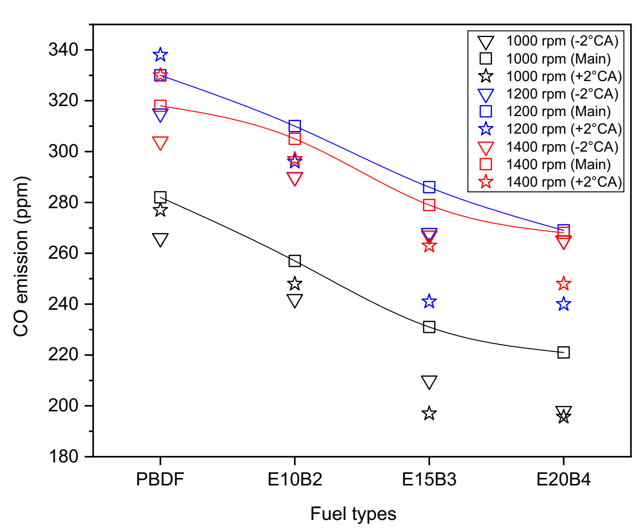


**Fig. 8** Effect of fuel blends and SOI time on CO emission

effect of ethanol-butan-2-ol-PBDF use on carbon monoxide emission at different engine speed and SOI time was demonstrated. Under all test conditions, it was determined that the CO emission values measured with PBDF fuel are higher than the CO emission obtained with blending fuels. This situation can be explained by the fact that the oxygen molecules of the mixed fuels provide better oxidation of the fuel in the cylinder and a better combustion event occurs as a result. Addition of ethanol to the diesel fuel increases oxygen content in the combustion, and this stimulates more oxygen to react with fuel in combustion chamber.

In this study, the maximum CO emission was measured as 338 ppm at the SOI time of 9.7°CA BTDC at 1200 rpm with PBDF. Minimum CO emissions for all fuel types are determined at 1000 rpm engine speed. In tests performed at 1000 rpm, the minimum CO emission in PBDF (266 ppm) and E10B2 (242 ppm) fuel types was obtained at the SOI time of 3.8°CA BTDC, while minimum CO emission values in E15B3 (197 ppm) and E20B4 (196 ppm) fuel types were obtained at 7.8°CA BTDC SOI time. When the results obtained at 1400 rpm are compared, the maximum CO emission was seen with PBDF (338 ppm) fuel at the SOI time of 11.1°CA BTDC, while at the same engine speed, maximum CO emission release with E10B2, E15B3 and E20B4 (305 ppm, 279 ppm and 268 ppm, respectively) was obtained at 9. 1°CA BTDC SOI time. It is thought that CO emissions increase as a result of the incomplete combustion of a certain part of the fuel–air mixture, as there will be a shorter time for the fuel injected into the cylinder to burn as a result of the increase in engine speed.

### 3.2.2 *Carbon dioxide ($CO_2$) emission*

$CO_2$ emission, known as natural result of combustion of fuels, contain hydrocarbon. Even though $CO_2$ emission is not considered as a pollutant gas, it is one of the major emissions causing greenhouse gases [45]. In Fig. 9, the effect of ethanol-butan-2-ol-PBDF on carbon dioxide emission at different engine speed and SOI time is shown.

Minimum $CO_2$ emissions for all fuel types were achieved at 1000 rpm engine speed. In the tests, the maximum $CO_2$ emission was seen as 5.1% at the SOI time of 9.7°CA BTDC at 1200 rpm with PBDF fuel, while the minimum $CO_2$ emission was determined as 4.27% at the SOI time of 7.8°CA BTDC at 1000 rpm with E15B3 fuel use. It is thought that at 1200 rpm and 1400 rpm engine speeds, there will be more movement of the air in the cylinder, thus improving combustion at certain points and causing an increase in $CO_2$ emissions.

At 1200 rpm engine speed, the maximum $CO_2$ emission for E10B2 and E20B4 fuels was achieved as 5.1% and 4.94%, respectively, at 9.7°CA BTDC SOI time, while at the same engine speed for E15B3 fuel the maximum $CO_2$

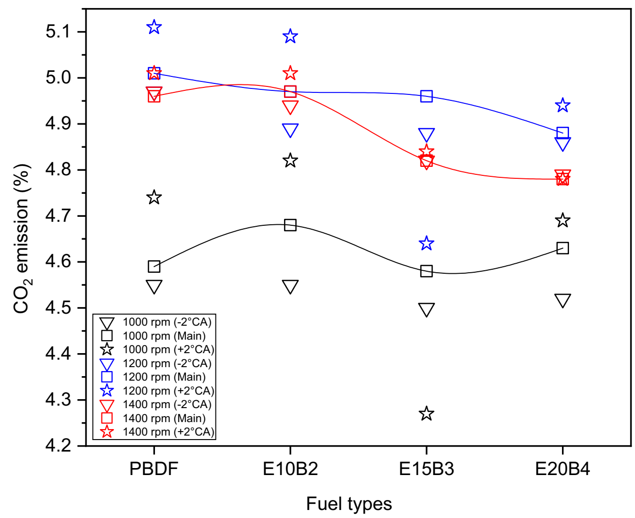


**Fig. 9** Effect of fuel blends and SOI time on $CO_2$ emission

emission was determined as 4.96% at the main (7.7°CA BTDC) SOI time. In the tests performed at 1400 rpm, at the SOI time of 11.1°CA BTDC, maximum $CO_2$ emission was observed as 5% with PBDF and E10B2 fuels, while a decrease in $CO_2$ emission was observed in the use of E15B3 and E20B4 fuels. Minimum $CO_2$ emission at 1400 rpm was achieved as 4.8% in the use of E20B4 fuel at the SOI time of 9.1°CA BTDC and 11.1°CA BTDC.

The molecular weight of PBDF fuel is a heavier fuel than those of alcohol fuels in terms of carbon and hydrogen molecules. Since the C/H ratio of PBDF fuel is slightly higher than that of blended fuels (see Table 2), it is generally thought to cause higher $CO_2$ emissions. On the other hand, more $CO_2$ emission was expected with ethanol–diesel blends, since CO is reacted with $O_2$ in the system and generates extra $CO_2$. However, this effect cannot be seen from the experiment results.

### 3.2.3 *Methane ($CH_4$) emission*

Methane emission is the second most important cause of global warming, after $CO_2$ emission. According to IEA [46] estimates, the oil and gas industry emitted approximately 82 Mt of methane in 2019. In Fig. 10, the effect of ethanol-butan-2-ol-PBDF on methane emission at different engine speed and SOI time is shown.

In engine tests performed at 1000 rpm engine speed, maximum methane emission was seen as 3.96 ppm with PBDF at the main SOI time (5.8°CA BTDC), and as the ethanol-butan-2-ol ratio in blended fuels increased, a decrease of up to 80% was seen in $CH_4$ emission. In all engine speeds by advancing the SOI time from TDC (+2°CA) minimum $CH_4$ emission was observed with use of E20B4 fuel as

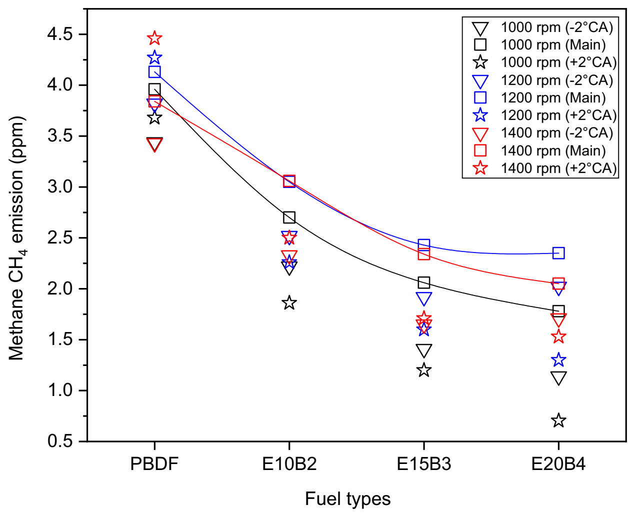


**Fig. 10** Effect of fuel blends and SOI time on $CH_4$ emission

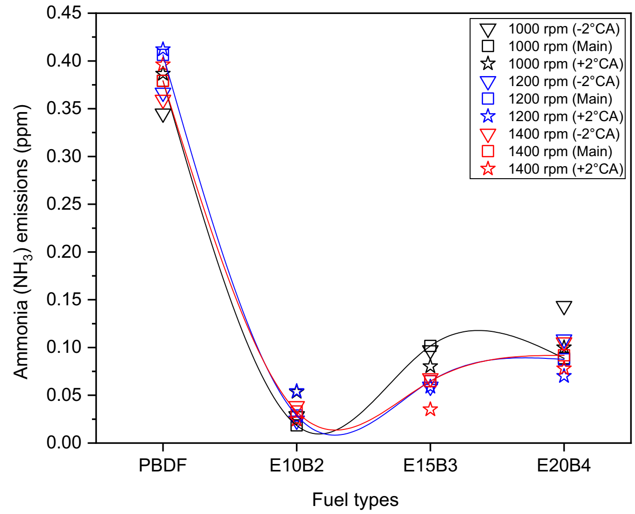


**Fig. 11** Effect of fuel blends and SOI time on $NH_3$

0.7 ppm, 1.3 ppm and 1.53 ppm at 1000 rpm, 1200 rpm and 1400 rpm, respectively.

In tests performed at 1200 rpm engine speed, maximum $CH_4$ emission was obtained as 4.27 ppm with PBDF at 9.7°CA BTDC SOI time. In addition, when comparing the SOI timing at 1200 rpm, for blend fuels the minimum $CH_4$ emission values were determined at the 9.7°CA BTDC SOI time. The maximum $CH_4$ emission at 1400 rpm was obtained as 4.46 ppm with PBDF at the SOI time of 11.1°CA BTDC, while at the same engine speed, the minimum $CH_4$ emission was determined as 1.53 ppm with E20B4 fuel at the 11.1°CA BTDC SOI time. When the injection start timings were compared at 1400 rpm, maximum $CH_4$ emission with blended fuels was seen at the 9.1°CA BTDC SOI time.

### 3.2.4 *Ammonia ($NH_3$) emission*

Ammonia, which has a colorless and pungent odor, is a caustic and dangerous gas [47]. The formation of $NH_3$ emission depends on the temperature level of the combustion chamber and the amount of hydrogen [48]. In Fig. 11, the effect of ethanol-butan-2-ol-PBDF on ammonia emission at different engine speed and SOI times is shown. It was observed that the $NH_3$ emissions obtained with PBDF at all SOI times and engine speeds are significantly higher than the $NH_3$ emissions obtained with blended fuels. Under all test conditions, the maximum $NH_3$ emission was determined as 0.42 ppm with PBDF fuel at 9.7°CA BTDC SOI time at 1200 rpm, while the minimum $NH_3$ emission was observed as 0.0185 ppm with E10B2 fuel at the 5.8°CA BTDC SOI time at 1000 rpm.

When the mixture fuels were compared among themselves, it was observed that, generally as the alcohol ratio in the mixture increased, the emission of $NH_3$ increased as well. The maximum $NH_3$ emission among the mixture fuels was measured as 0.144 ppm at the SOI time of 3.8°CA BTDC at 1000 rpm with E20B4 fuel. At 1400 rpm engine speed, as the SOI timing is advanced from the TDC (11.1°CA BTDC), the increase in the $NH_3$ emission seen with PBDF fuel was observed, while a decrease in the $NH_3$ emission was detected as the SOI timing was advanced from the TDC in the use of blended fuels at the same engine speed. Dhahad et al. [49] reported that $NH_3$ formation depends on the combustion temperature in the combustion chamber.

### 3.2.5 *Nitrogen oxides ($NO_x$)*

Formation of $NO_x$ emissions are depended on combustion temperature, oxygen concentration and cylinder gas pressure [45]. In Fig. 12, the effect of ethanol-butan-2-ol-PBDF use on nitrogen oxide emission at different engine speeds and SOI times is given. In the tests, the maximum $NO_x$ emission value for each fuel at all SOI times and engine speeds was determined at the SOI time of 7.8°CA BTDC at 1000 rpm. The maximum $NO_x$ emission value was obtained as 759 ppm with E10B2 fuel at the 7.8°CA BTDC SOI time at 1000 rpm. At all engine speeds, an increase in $NO_x$ emissions was observed by advancing the injection start timing from the TDC. The increase in $NO_x$ emission can be explained by the early injection of the fuel and the lower engine speed, as there is more time for the mixture to be more homogeneous, as a result the combustion improves and the temperature in the cylinder increases.

In the tests performed at 1200 rpm engine speed, the minimum $NO_x$ emission was determined as 444 ppm with PBDF fuel at the 5.7°CA BTDC injection start time, while the maximum $NO_x$ emission value was found as 679 ppm with E10B2 fuel at the 9.7°CA BTDC injection start time.

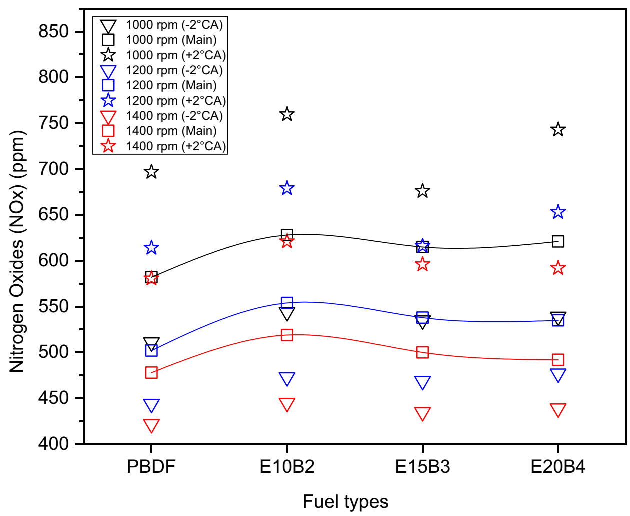


**Fig. 12** Effect of fuel blends and SOI time on $NO_x$

While at main injection start time (7.7°CA BTDC) at 1200 rpm engine speed, the minimum $NO_x$ emission was determined as 502 ppm with PBDF fuel, the $NO_x$ emission value increased to more than 530 ppm in the use of E15B3 and E20B4 fuels. Minimum $NO_x$ emission values at all engine speeds and injection start times were obtained by approaching the injection start time to TDC at 1400 rpm. In the tests performed at 1400 rpm, maximum $NO_x$ emission emissions were determined as 445 ppm, 519 ppm and 621 ppm with E10B2 fuel at all injection start times (7.1°CA BTDC, 9.1°CA BTDC, 11.1°CA BTDC, respectively). It is thought that the reason for the decrease in $NO_x$ emissions with the increase in engine speed is due to the fact that there is not enough time for the air–fuel mixture in the cylinder to mix and burn homogeneously. It can also be explained by the fact that, when compared to PBDF, fuel mixtures improve combustion in the cylinder due to their oxygen content and cause an increase in $NO_x$ emission.

## 4 Conclusion

In this study, the effects of injection timing changes on combustion and exhaust emissions at engine speeds of 1000, 1200, and 1400 rpm with %50 engine pedal position in a diesel engine with equipped a common rail fuel injection system using ethanol-butan-2-ol-PBDF fuel mixtures were experimentally investigated. In order to prevent phase separation in ethanol-PBDF mixtures and to keep the fuels homogeneous for a longer time, 2-butanol was added as co-solvent to the fuel mixtures. It was observed that addition of 2-butanol to the fuel blends extended the phase separation time. Hence, the drawbacks of phase separation were eliminated before the engine tests.

Compared to neat PBDF, at the same engine speed and injection start timing, up to 10% increase in ignition delay duration and more than 20% reduction in combustion duration were detected with the use of alcohol-PBDF fuel mixtures. The highest combustion duration and ignition delay times for all fuel types were observed at 1400 rpm engine speed. The highest maximum cylinder gas pressure was obtained as 85.4 bar at 1000 rpm engine speed and BTDC 7.8°CA with the use of E15B3 fuel. The combustion of ethanol-PBDF fuel blends led to increase in in-cylinder pressure up to 5% when compared with combustion of neat PBDF. Moreover, when SOI time was advanced during the use of ethanol-PBDF blends, it was observed that in-cylinder gas pressure was increased up to 10% when compared to those of neat PBDF.

It is a known phenomenon that increasing engine speed increases engine power. Both with neat PBDF and ethanol-PBDF fuel blends showed similar trends in view of engine speed and power. However, the increasing content of ethanol in fuel blend reduced the engine power. During the tests, the pedal position was kept the same, so the same amount of fuel was injected for all test points and for all fuel blends. It was concluded that the slightly decrease in engine power was related with energy content of ethanol and increase in ethanol ratio in the blend reduced the overall energy content of the fuel mixture. The observation of reduction in engine power is consistent with the presence of ethanol in the fuel.

It was seen that injection start timing has not a significant effect on combustion noise. The highest combustion noise was obtained as 92.7 dB with the use of E15B3 fuel at 1400 rpm. In the same test conditions, when compared to neat PBDF, combustion noise values increased by up to 5% as a result of using blended fuels. In addition, combustion noise increased for all test fuels with the increase in engine speed.

At the same SOI and engine speed, more than 20% reduction in CO emissions was detected as a result of using fuel mixtures. Compared to 1200 rpm and 1400 rpm, the lowest CO and $CO_2$ emissions of all fuel types showed up to 25% reduction at 1000 rpm. Even though a decrease in $CO_2$ emissions was observed with the use of E15B3 and E20B4 fuels, dramatic reduction in $CO_2$ emission as in CO emission was not observed for ethanol-PBDF fuel blends in general, and this is mainly related with the mixture's contents. The reduction in $CH_4$ emission could be reached up to 80% at 1000 rpm engine speed and advanced SOI time with ethanol-PBDF fuel blends. On the other hand, $NH_3$ emission measured with ethanol-PBDF fuel blends showed that almost 13 times reduction when compared with neat PBDF results. It was observed that SOI and engine speed did not have a significant effect. Compared to PBDF, up to 10% increase in $NO_x$ emissions were detected as a result of the use of fuel blends, resulting in an increase in $NO_x$ emissions of more than 15% in SOI advances for all fuel types.

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**Authors' contributions** MV was involved in investigation, methodology, data processing, writing—original draft. ANÖ was involved in conceptualization, visualization, supervision and project administration. MH was involved in engine testing. ANÖ, CS and MH were involved in writing—review and editing.

## Declarations

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

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