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RESEARCH ARTICLE-MECHANICAL ENGINEERING

Effects of Supercharge Pressure on Combustion Characteristics of a Diesel Engine Fueled with Alcohol–Diesel Blends

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Abstract

The recent increase in diesel prices is the most crucial factor that maintains alternative fuel research on the agenda in diesel engines. This study aims to analyze the combustion characteristics of ethanol–butanol–diesel triple-fuel mixtures and to investigate the effects of the boost pressure in a single-cylinder diesel engine. In the engine test, while the boost pressure at 1600 rpm was fixed at 240 mbar, the intake air pressure gauge was increased to 264, 228, and 312 mbar. As a result of the study, the most prolonged combustion duration in all test conditions was obtained using pure fossil diesel fuel. More than a 10% increase in ignition delay times has been calculated for blends. In addition, significant increases were observed in the heat release rate as the alcohol content in the blends increased. While considerable reductions in CH_4, CO, and CO_2 emissions were monitored by using the alcohol–diesel mixtures with the increased boost pressure, the stable formation in NO_x emissions was not observed. Moreover, there was a significant increase in combustion noise with alcohol–diesel blends.

Keywords Boost pressure · Alcohol fuels · Diesel engine · Combustion · Exhaust emission

1 Introduction

Since the beginning of the nineteenth century, diesel engines have been used in many areas, especially in industry, agriculture, and transportation, due to their high efficiency and power density. However, the exhaust emissions from diesel engines fueled with petroleum-based diesel fuel harm the environment and human health. Reducing fuel consumption and exhaust emissions is one of the main problems that researchers and vehicle manufacturers have to overcome as a result of studies conducted in recent years on the problem of energy resources and environmental protection. In this context, studies on increasing the use of oxygen-containing alcohol fuels, developing injection strategies, and providing higher intake air pressure applications have gained momentum to reduce fossil fuel use and increase energy efficiency [1–3].

When ethanol is used as a fuel in internal combustion engines, it is considered a renewable biofuel with cleaner combustion properties [4, 5]. Due to its spray and physical fuel properties [6] close to fossil diesel fuel, it is suitable for internal combustion engines without any significant changes. Oxygen content is one of the most important properties of ethanol that improves combustion efficiency [7]. On the other hand, one of the significant problems in ethanol–diesel fuel mixtures, especially the blends containing high ethanol, decreases the phase separation period [8–10].

In some studies, adding various co-solvents to the mixtures was suggested to prevent the phase separation in ethanol–diesel blends [11, 12]. Because of the chemical structures of fossil diesel fuel and ethanol and some of their different properties, it is necessary to add a good solvent to

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create a homogeneous blend. Suggested solvents (such as *n*-butanol) may increase the molecular interaction between diesel and alcohols, prolonging the stability of the mixture [13].

Butanol, which has a four-carbon structure, has a higher chain than ethanol. It was recommended to use *iso*-propanol, *n*-propanol, *n*-butanol, *iso*-butanol, and *sec*-butanol to prevent phase separation and stabilize the mixture by adding them to ethanol–diesel fuel mixtures [14, 15]. Recently, Jin et al. [16] studied the effects of phase separation in alcohol fuels; they stated that butanol fuel has properties close to the water and therefore shows good water-holder properties against phase separation. In this study, 2-butanol was added to the ethanol–diesel fuel mixtures as a solvent at 20% of the mixture's ethanol ratio to extend the phase separation period.

In the literature, recent studies [17–20] have been carried out to examine the effect of engine parameters on combustion in diesel engines fueled with the alcohol–diesel blends. It was seen [21] a decrease in ignition delay time, combustion time, and brake-specific fuel consumption values with the increased boost pressure. While they monitored an increase in NO_x emissions with the increased boost pressure, they observed a decrease in soot emissions. And it revealed [22] that reducing the boost pressure increases the ignition delay time, ensuring the NO_x emission is low. Some researchers [23, 24] presented the effect of using ethanol as a fuel in a single-cylinder diesel engine on engine performance and exhaust emissions. As a result of the increase in intake air temperature, higher cylinder gas pressure and heat release ratio were observed, resulting in an increase in thermal efficiency. It was determined that the energy conversion time of the fuel was shortened with the increase in the intake air temperature. Zhao et al. [25] investigated the effect of low boost pressure on combustion and emissions in a diesel engine using diesel–alcohol mixtures. They observed an increase in the ID time and a shortening in the combustion time with the use of blends. As a result of the decreasing boost pressure, they observed a decrease in the cylinder gas pressure and a slight increase in the heat release rate. They have seen an increase in NO_x emissions because of the use of blended fuels. In another study, Yan et al. [26] investigated the effect on combustion and emissions under low boost pressure by using alcohol–diesel mixtures in a diesel engine. As a result of using alcohol fuels added to diesel, they found an increase in the ignition delay time and a shortening of the combustion duration. They obtained a rise in cylinder gas pressure with the use of fuel mixtures, but they also found a decrease in cylinder gas pressure because of the decrease in intake air pressure.

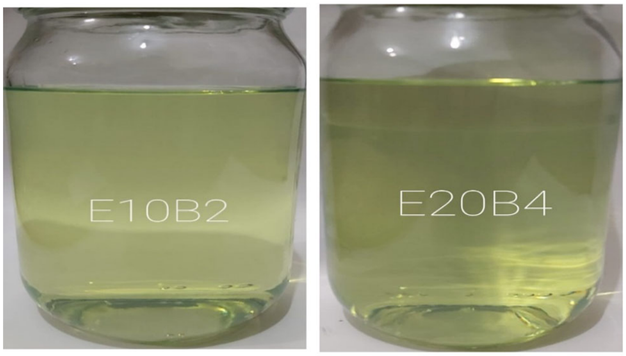
In recent studies, Chaurasiya et al. [27] investigated the effect of injection timing and blended fuels in a single-cylinder diesel engine on performance, combustion, and exhaust emissions. It was stated that using diethyl ether–diesel blends increased the cylinder gas pressure, which caused a reduction in ignition delay. In addition, they reported that the advanced injection timing negatively affected NO_x emission. Dasore et al. [19] studied the effect of ethanol–diesel fuels at different compression ratios on combustion and exhaust emission characteristics. They saw that for all fuel types, an increase in compression ratio caused a significant improvement in the cylinder gas pressure. On the other hand, they stated that a slight decrease was seen in ignition delay with a higher compression ratio. Rajak et al. [28] determined the effect of biodiesel–diesel, ethanol–diesel, and methanol–diesel blend fuels on engine performance and exhaust emission in a compression ignition engine. They reported that there was a significant decrease in the brake-specific fuel consumption when the engine load was increased from 25 to 50% engine load for all test fuels. Moreover, increased alternative fuel amount in the blend caused a reduction in NO_x emission.

As mentioned above, the alternative fuel research and product development processes for internal combustion engines continue rapidly. Engine researchers intensively focused on the investigation of crucial engine parameters such as fuel injection timing, pilot injection, injection pressure, compression ratio, and boost pressure for various diesel alternative fuels. But there is still a need to show the change in exhaust emissions to achieve future Euro emission standards. We need more data to show the effect of alcohol fuel blends on combustion noise, and the effect of boost pressure on methane emission, which is shown as the primary greenhouse emission, so we can quickly catch up with future emissions standards. This study investigates the engine performance and emission characteristics in a diesel engine operating with alcohol–diesel mixtures under the increased boost pressure. For this purpose, the effects of intake manifold pressure of a diesel engine using different alcohol–diesel mixtures on cylinder gas pressure, heat release rate (HRR), combustion noise (CN), ignition delay (ID), the duration of combustion (DOC), carbon monoxide (CO), carbon dioxide (CO_2), methane (NH_3), nitrogen oxide (NO_x) emission, and the exhaust gas temperature (EGT) were discussed and compared with the literature.

2 Methodology

2.1 Preparation of Test Fuels

In this study, neat petroleum-based diesel (D100) was purchased from a national fuel station in Turkey. The ethanol produced by J.T. Baker with 95% purity and the 2-butanol produced by Merck with 95% purity which was used as a co-solvent to prevent phase separation were used in



a) E10B2 b) E20B4

Fig. 1 Prepared blend fuels

ethanol–diesel mixtures. The blends were prepared in volumetric (v/v) proportions. E10B2 is containing 10% ethanol + 2% butan-2-ol + 88% D100, and E20B4 is containing 20% ethanol + 4% butan-2-ol + 76% D100. The pictures of the prepared mixtures are given in Fig. 1. The fundamental properties of prepared blends are given in Table 1. Lower heating value (LHV) measurements using ASTM D4809 were used to calculate the LHV for two alcohol–diesel mixtures. The cetane index of the fuel mixtures was calculated using the ASTM D4737 method.

This study aimed to increase the stability of the mixtures by adding 2-butanol to the blends to prevent phase separation in the ethanol–diesel mixtures. The fuels used in the experiments were kept in glass jars with special lids, and observations were made. As a result of the observations, phase separation was determined for all fuels within approximately 20 min in ethanol–diesel fuel mixtures prepared without adding 2-butanol to the fuels. However, the fuels were kept homogeneous for a longer time as a result of the addition of 2-butanol, 20% of the ethanol ratio, to the fuel mixtures. While phase separation was observed after 24 h for E20B4, no visible phase separation was observed for more than 10 days in the observations made for E10B2. In future studies, it is recommended to use 2-butanol as an option for researchers to prevent phase separation in ethanol–diesel fuel mixtures and to keep the mixtures more homogeneous. The basic properties of the main test fuels are given in Table 2.

2.2 Engine Test Setup and Conditions

The engine tests were conducted on a single-cylinder diesel engine with John Deere branded. The general features of the diesel engine are given in Table 3. AVL branded 515X supercharger device was used to control the boost air pressure. The properties of the 515X supercharger system are shown in Table 4.

The engine tests were performed as described in EN ISO 14396 standard concerning the test environment and

Table 1 Blend contents of test fuels

Test Fuels	D100 vol. %	Ethanol vol. %	2-Butanol vol. %	LHV (MJ/kg)	Cetane Index	C % m/m	H% m/m	O% m/m	Formula of Fuels
D100	100	–	–	42.6	51	85.71	14.29	–	$C_{12}H_{24}$
E10B2	88	10	2	40.83	45.98	84.51	14.24	1.25	$C_{10.84}H_{21.92}O_{0.12}$
E20B4	76	20	4	39.08	40.96	83.06	14.19	2.75	$C_{9.68}H_{19.84}O_{0.24}$

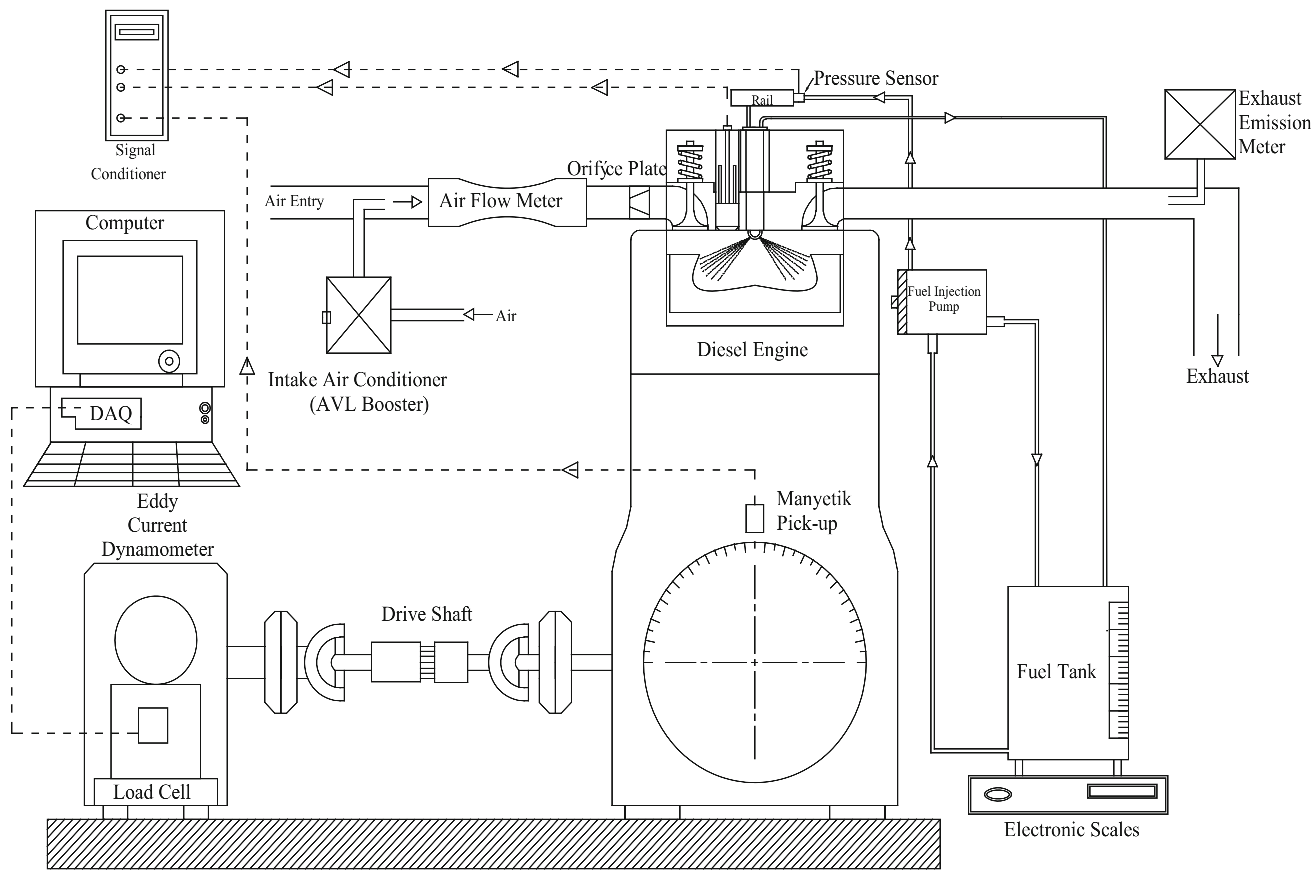


Fig. 2 Schematic view of the experiment setup

Table 2 Properties of D100, ethanol, and 2-butanol

Properties	D100 ($C_{12}H_{24}$)	Ethanol (C_2H_6O)	2-Butanol ($C_4H_{10}O$)
Purity	–	≥ 0.99	≥ 0.99
Density (kg/m^3)	~ 845	790	805
Viscosity (mm^2/sec, 40 °C)	~ 3.5	1.13	3.1
Lower Heating Value (MJ/kg)	42.6	26.7	34.4
Boiling Point (^{o}C)	> 160	78	102
Flash Point (^{o}C)	≥ 55	12	20.5
Water Content (%)	0.020	≤ 0.2	≤ 0.2
Cetane Number	≥ 51	8	15
Auto-ignition Temperature (^{o}C)	≈ 210	361	405

measurement methods defined in EN ISO 8178 standard. Engine tests were carried out the increased boost pressure by 10%, 20%, and 30% at 1600 rpm (± 2 rpm) constant

Table 3 Engine specification

Engine Type	Single Cylinder – 4 strokes
Fuel System	Common Rail Direct Injection – 1800 bar
Cylinder Volume	1205 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

Table 4 Supercharger specification

Temperature control range	– 30 / 130 °C
Temperature accuracy	± 5 °C
Pressure control range	200 / 400 mbar
Pressure accuracy	± 10 mbar
Maximum air volume	Depends on design

Table 5 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 (0–250 bar)	≤ 0.01 bar
Engine Coolant & Oil Conditioning	AVL-577	± 1 K
Fuel Consumption	AVL-735	< 0.15%
NO_x	AVL AMA i60 (Chemiluminescence Detector)	≤ ± 1%
CO, CO_2	AVL AMA i60 (NDIR)	≤ ± 0.5%
Temperature Sensors	PT100 (K Type)	≤ ± 1 K

engine speed and 50% (± 1 Nm) engine load. The boost pressure gauge of 240 mbar is the original boost pressure defined by the engine manufacturer. The boost pressure gauge was increased in each test by 10%, 20%, and 30%. The obtained data were interpreted concerning 240 mbar and pure D100. The schematic view of the test setup is shown in Fig. 2.

An AVL-GU22C model cylinder gas pressure transducer was used to be measured the change in cylinder gas pressure values at each crank angle. The AVL brand eddy current dynamometer was used in the tests to keep the engine load and speed constant. The accuracy values of the devices used in the experiments are given in Table 5. The instantaneous cylinder gas pressure and the instantaneous displacement volume in the cylinder were observed with 0.1 °CA crank angle resolution, then taking the average of 50 cycles, and the mean data were obtained. The experiments were repeated at least three times, and then, the average values were used.

Before starting the tests, using D100, the engine was operated until the oil temperature reached 90 °C and the engine was stabilized. In the tests, the intake air temperature was kept constant at 25 °C, the fuel temperature at 20 °C, and the cooling water temperature at 70 °C. The engine test conditions are given in Table 6.

In the engine test, the CO, CO_2, CH_4, and NO_x emissions were measured using the AVL-FTIR exhaust emission device. The uncertainties in the exhaust emission values are given in Table 7.

2.3 Calculation of Heat Release Rate (HRR)

The heat release rate was calculated based on the first law of thermodynamics. The heat release rates were obtained from Eqs. 1 and 2 according to the change in crank angle (°CA).

$$\mathrm{d}Q_{\mathrm{net}} = \mathrm{d}W + \mathrm{d}U \tag{1}$$

Table 6 Fixed test conditions in the experiments

Input Parameters	Unit	Value
Engine speed	rpm	1600
Engine load	%	50
Engine coolant temperature	°C	70
Engine oil temperature	°C	90
Main intake air pressure gauge	mbar	240
10% increased boost pressure gauge	mbar	264
20% increased boost pressure gauge	mbar	288
30% increased boost pressure gauge	mbar	312
Air temperature	°C	25
Total injected fuel quantity	mg/stroke	45
Standard main injection at 1600 rpm	°CA BTDC	10.4

Table 7 Uncertainties of measured values

Measuring Value	Uncertainties
CO	2.8%
CO_2	1.1%
CH_4	3.1%
NO_x	2.1%

$$\mathrm{d}Q_{\mathrm{net}} = \left[\left(\frac{k}{k-1}\right)P\mathrm{d}V + \left(\frac{1}{k-1}\right)V\mathrm{d}P\right] \tag{2}$$

where $\mathrm{d}Q_{\mathrm{net}}$ represents the net rate of heat release (J/°CA), V is the volume of the cylinder (m^3), P represents pressure (Pa) in cylinder, dW is boundary work due to piston displacement (Nm), dU is change in sensible internal energy (J), and k is the ratio of specifics heats.

Although the net HRR is calculated in the calculation made according to Eq. 2, it is necessary to calculate the heat transfer in the cylinder wall to obtain precise results. In this context, the results obtained by considering the heat transfer on the cylinder wall (Q_{ht}) were named as the gross HRR (Q_{g} or HRR). In this study, the gross HRR was calculated using Eq. 3, and the heat transfer rate on the cylinder wall was computed using Eichelberg's convective heat correlation [29] (Eq. 4–6). The wall temperature of the cylinder is accepted as constant and uniform (400 K) [30].

$$\mathrm{d}Q_{\mathrm{g}} = \mathrm{d}Q_{\mathrm{net}} + \mathrm{d}Q_{\mathrm{ht}} \tag{3}$$

$$T_{\mathrm{g}} = \frac{\mathrm{PV}}{(nR_u)(K)} \tag{4}$$

$$h_g = 767 \times 10^{-8}(V_p)^{1/3}(\mathrm{PT_g})^{1/2},\ (\mathrm{kW/m2K}) \tag{5}$$

$$Q_{\mathrm{ht}} = h_{\mathrm{g}} A_{\mathrm{w}} (T_{\mathrm{g}} - T_{\mathrm{w}}) \tag{6}$$

where n is the number of moles of the working gas (mol), R_u is the universal gas constant (J/mol-K), h_g is convective heat transfer coefficient (W/m^2 K), $A_w$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), and V_p is the mean piston speed, m/sec.

2.4 Uncertainty analysis

Within the scope of this study, uncertainty analysis was performed to express the stability of each measurement value [31]. In the exhaust emission, the N times measurement of an A variable was made to express error values. Equation 7 was used to describe the mean value A_m, the standard deviation, which is an expression of the distribution of the A values.

$$A_m = \frac{1}{N} \sum_{i=1}^{N} A_i \tag{7}$$

In this study, Eq. 8 was used to calculate the standard deviation (S_{sd}). It estimates the effects on the measured variable, A, of the error sources that change during the measurement. Calculated standard deviation values are given in Table 8.

$$S_{sd} = \left[\frac{1}{N-1} \sum_{i=1}^{N} (A_i - A_m)^2 \right]^{\frac{1}{2}} \tag{8}$$

3 Results and discussion

3.1 Comparison of cylinder gas pressure and HRR

Figure 3 shows the effect of different intake air pressures on cylinder gas pressure and HRR. The maximum cylinder gas pressure was detected with increasing boost pressure for all test fuels.

With the use of D100, the highest cylinder gas pressure was obtained as 82 bar by 30% increased the boost pressure. In the tests carried out under the original boost pressure, it was observed that the maximum cylinder gas pressure values obtained for D100, E10B2, and E20B4 were close to each other. In 10%, 20%, and 30% increased boost pressure applications, when compared to the use of D100, an increase in cylinder gas pressure values was seen with the use of blended fuels. In E20B4 use, the lowest cylinder gas pressure was observed as 79.6 bar in the original boost pressure application. Compared to D100, although the fuel mixtures have lower heating values, it is thought that the increase in cylinder gas pressure as a result of the use of fuel mixtures occurs because the oxygen amount of the mixed fuels improves the combustion in the cylinder and causes better atomization in the cylinder due to the low viscosity of the fuel mixtures. It can be explained by the increase in the cylinder gas pressure due to the improvement of the combustion in general due to the increase in the boost pressure and the intake of more oxygen into the cylinder.

Pires de Oliveira et al. [12] observed an increase in combustion performance due to the increase in combustion speed with the intake of more oxygen into the cylinder. Kim et al. [17] confirmed an increase in cylinder gas pressures with the addition of ethanol to diesel. They stated that it was due to the improved fuel atomization due to ethanol's lower density and viscosity. The maximum cylinder pressure (CP_{max}) values and their locations (ACP_{max}), maximum HRR (HRR_{max}), and other combustion phases are given in Table 9.

Under all test conditions, the maximum HRR was obtained as 218 J using E10B2, while the minimum HRR was obtained as 100.6 J using D100 in the main boost pressure application. Compared to the main boost pressure application for E10B2, a decrease in the maximum HRR was observed with the increase in the boost pressure ratio. With the use of E20B4, the maximum HRR in 20% and 30% boost pressure applications were determined to be close to 207 J and 209 J, respectively, while the minimum HRR for the same fuel type was observed as 170 J at 10% boost pressure application. Compared to D100, a significant increase was detected in the maximum HRR values with the use of blends. In applying 10% boost pressure, the highest HRR was observed using E10B2, while the minimum HRR was found using D100. The increase in the HRR values as a result of the use of fuel mixtures can be explained by the fact that more fuels are burned simultaneously due to the low cetane number of alcohol fuels and the increased ignition delay caused by the high latent heat of vaporization [32, 33].

3.2 Comparison of ignition delay (ID) values

In this study, the ID time was computed as the time from the start of the injection to the start of combustion. Figure 4 shows the effect of the boost pressures and blend fuels on ID and DOC. For all fuels, a slight reduction in ID times was found with an increase in boost pressure compared to the main boost pressure. This situation can be explained by the fact that more oxygen is taken into the cylinder with the application of high boost pressure. In all boost pressure applications, it was observed that the ID increased due to the use of blends compared to D100. These results can be explained by the low cetane number and high latent heat of vaporization of alcohol fuels. The ID times increase because of the low cetane number, which reveals that the fuel will

Table 8 Standard deviations of measured values

Application	Fuels	CN	CO	CO_2	NO_x	CH_4	EGT
Original Boost	D100	0.595	3.912	0.0230	3.810	0.0673	0.267
	E10B2	0.540	3.952	0.0238	5.7	0.0987	0.783
	E20B4	0.406	4.403	0.0271	6.252	0.0449	0.592
10% Inc. Boost Pressure	D100	0.433	4.918	0.0202	5.223	0.0333	0.414
	E10B2	0.567	4.399	0.0230	5.045	0.0718	0.739
	E20B4	0.543	4.8	0.0466	9.239	0.0757	1.409
20% Inc. Boost Pressure	D100	0.465	4.625	0.0210	5.251	0.0476	0.472
	E10B2	0.621	4.164	0.0232	5.056	0.0713	0.574
	E20B4	0.395	3.592	0.0319	5.727	0.0564	1.003
30% Inc. Boost pressure	D100	0.593	4.815	0.0186	3.678	0.0608	0.284
	E10B2	0.465	4.161	0.0257	5.344	0.0498	0.519
	E20B4	0.370	3.490	0.0219	4.972	0.0425	0.685

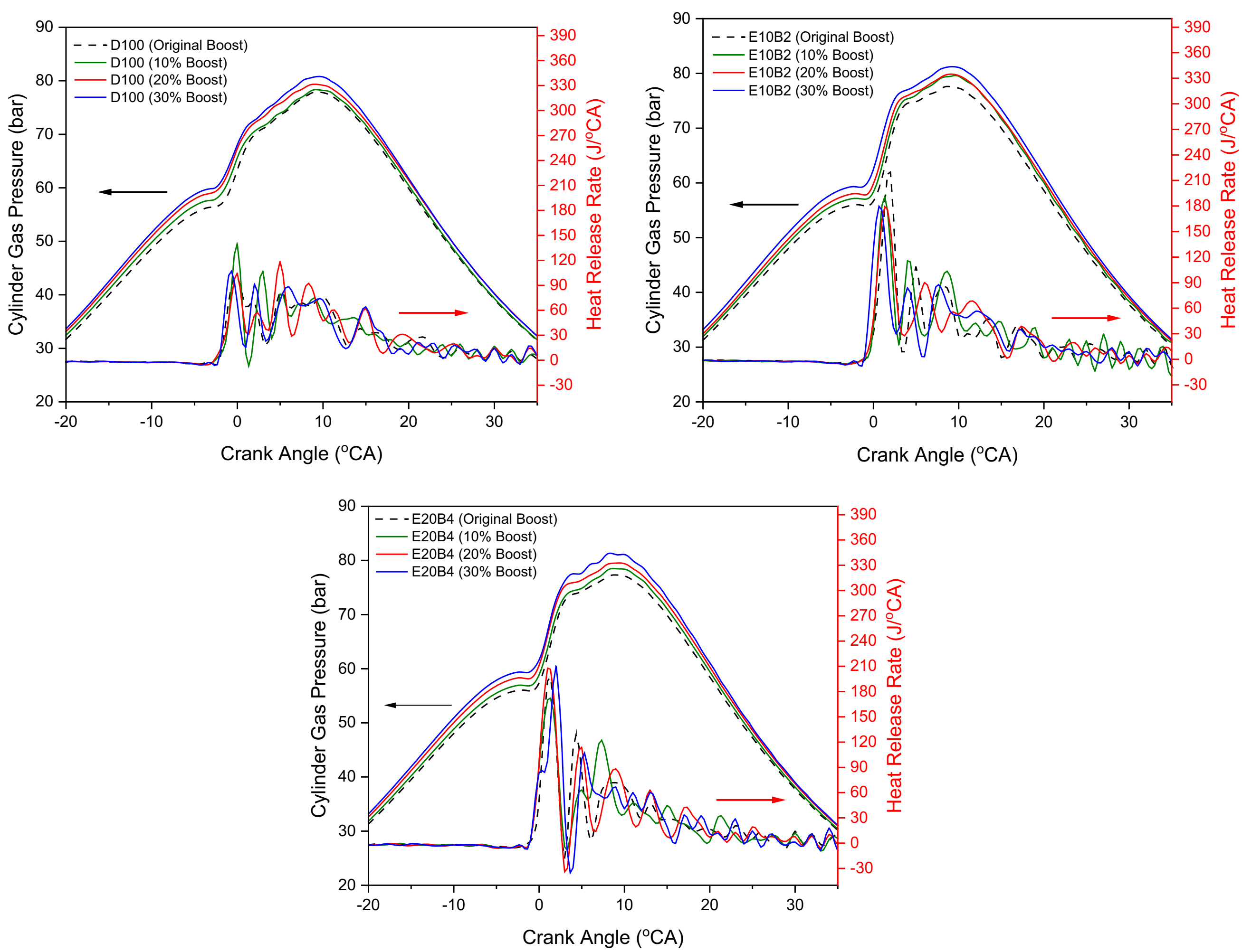
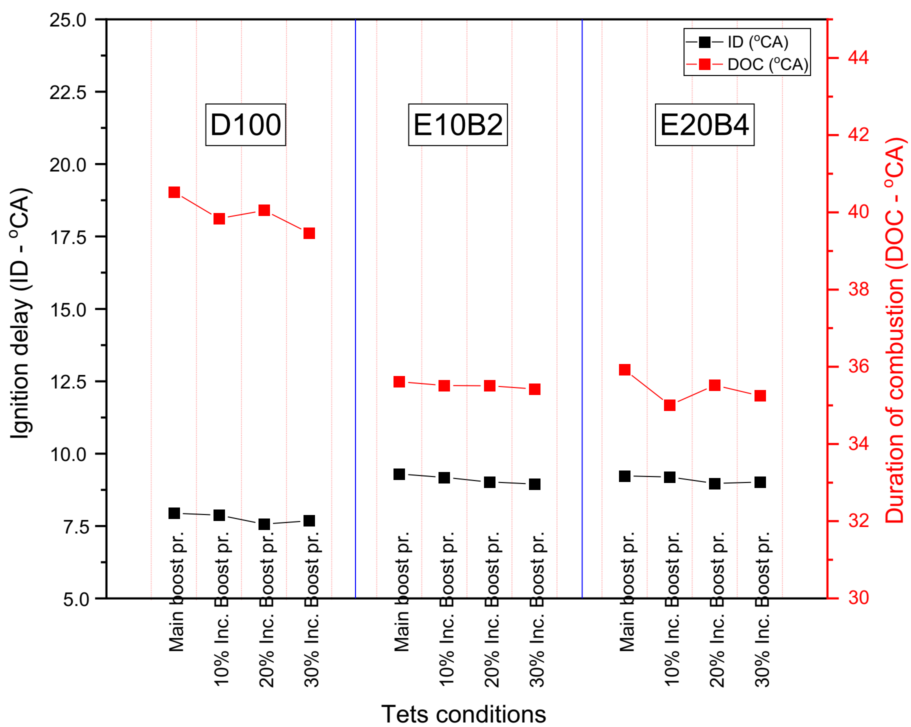


Fig. 3 The effect of the boost pressure on cylinder gas pressure and HRR

Table 9 Combustion phases

	Fuel Types	CP_{max} [bar]	ACP_{max} [°C ATDC]	HRR_{max} [J/°CA]	MFB10 [°C ATDC]	MFB50 [°C ATDC]	MFB90 [°C ATDC]
Main Boost Pr	D100	79.9	9.8	100.6	0.9	10.8	38
	E10B2	80.6	9	218	1.6	9.2	34.5
	E20B4	79.6	9.8	192	1.5	9.3	34.7
10% Inc. Boost Pr	D100	80.4	9.6	138	0.98	10.6	37.3
	E10B2	84.7	9.1	180	1.4	9.5	34.3
	E20B4	82.7	7.5	170	1.4	9.1	33.8
20% Inc. Boost Pr	D100	81.2	9.8	118	1	10.6	37.2
	E10B2	82.2	9.7	164	1.3	9.3	34.1
	E20B4	82	3.1	207	1.3	9.3	34.1
30% Inc. Boost Pr	D100	82	10.9	101	0.99	10.5	36.7
	E10B2	84.6	9.3	173	1.2	9.3	33.9
	E20B4	86.1	2.9	209	1.3	9.2	33.8

Fig. 4 The effect of the boost pressures on ID and DOC



show more resistance to ignition. Also, the fuel with the high latent heat of vaporization will absorb a high amount of heat from the environment while evaporating, causing the temperature of the cylinder to decrease so ID increases. In all test conditions, the shortest ID time was determined as 7.5 °CA at a 20% boost pressure increase, while an increase in ID times of more than 1.4 °CA was observed using fuel mixtures. In 10% boost pressure application, the longest ID time was obtained with E20B4, E10B2, and D100, respectively, 9.2 °CA, 9.1°CA, and 7.8 °CA. Emiroglu et al. [33] reported an increase in the ID times due to the decrease in the cetane number with the addition of alcohol fuels to diesel. Ning et al. [34] reported an increase in the ID time due to the addition of alcohol fuels, which have high latent heat of vaporization, to diesel fuel, causing a cooling effect in the cylinder.

In this study, the duration of combustion (DOC) was taken as the time between the start of combustion (SOC) and the point of 90% mass fraction burned (MFB90) of fuel. As shown in Fig. 4, the maximum DOC was obtained using D100 under all test conditions. It has been observed that the use of mixtures shortens the DOC up to about 5°CA. While the shortest DOC was obtained with the use of E20B4 in 10% increased boost pressure application, the DOC was 35 °CA. In contrast, the longest DOC was determined as 40.5 °CA with the use of D100 in the main boost pressure application. As a result of increasing the boost pressure by 30% compared to the main boost pressure application, approximately 1°CA reduction in DOC was obtained for the D100. In the use of E10B2, the longest DOC was observed in the main suction pressure application, while the shortest DOC for E10B2 was observed in the 30% increased boost pressure application. In the 20% increased boost pressure application, the longest DOC was observed in the use of D100 as 40 °CA, while it was determined that the DOC was shortened in the use of blends. In general, a reduction in DOC as a result of the use of E10B2 and E20B4 compared to D100 can be explained by the fact that the oxygen they have in the fuel mixtures improves and accelerates the combustion in the cylinder. In addition, it is thought that the decrease in DOC seen in all fuel types as a result of the increased boost pressure is due to the improvement of the combustion of the excess amount of oxygen taken into the cylinder. Liang et al. [35] stated that with the addition of alcohol fuels to diesel, the oxygen in the mixture fuels accelerates the combustion in the cylinder. Han et al. [36] also stated that adding alcohol fuels to diesel increases the ID time, resulting in a shorter DOC as more fuel is burned together.

3.3 Comparison of combustion noise

One of the types of noise, which is one of the environmental problems sourced from diesel engines, is the combustion noise that occurs due to the sudden increase in the cylinder gas pressure [37]. Figure 5 shows the effect of the boost pressures on combustion noise. In general, it has been determined that the combustion noise measured in the use of D100 in all test conditions is lower than the combustion noise measured with blends. The maximum combustion noise was calculated as 93.44 dB in the main boost pressure application using E10B2. With the increase in the boost pressure, a slight decrease in combustion noise was monitored in all fuel types. The minimum combustion noise was calculated as 88.5 dB with D100, and a 30% boost pressure increase in all test conditions. It is thought that with the increase in the boost pressure, more oxygen is taken into the cylinder, and the combustion becomes more controlled and better, decreasing the combustion noise.

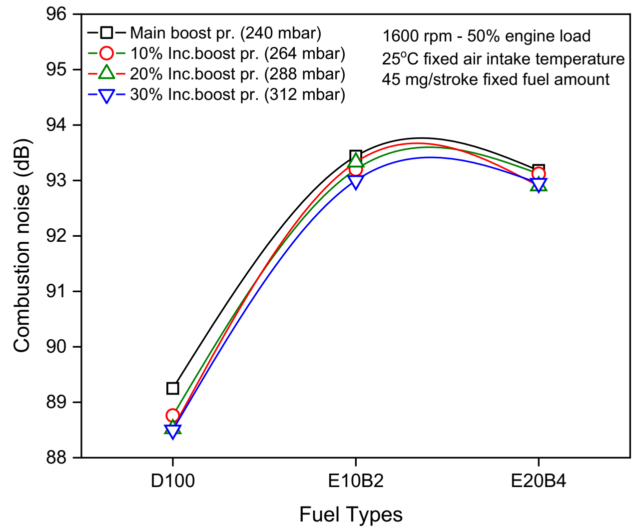


Fig. 5 The effect of the boost pressures on combustion noise

In all boost pressure changings, the maximum combustion noise was measured using E10B2. In the maximum boost pressure application, the highest combustion noise was measured in the use of E10B2, and then, the highest combustion noise was measured in the use of E20B4, while it was determined in the use of D100 in the same boost pressure application (88.5, 93, and 92.95 dB, respectively). It was seen that combustion noise increased in the use of E10B2 and E20B4 compared to pure D100 in 20% increased boost pressure application. It is thought that the increase in combustion noise as a result of the use of E10B2 and E20B4 is due to the increase in the ID time due to the low cetane number of the fuel mixtures and the high pressure increase as a result of the sudden combustion [38, 39].

3.4 Comparison of carbon monoxide (CO) emission values

CO emission is a toxic exhaust emission that occurs as a result of complete combustion due to low temperatures in the cylinder [40]. Figure 6 shows the effect of ethanol–diesel mixtures and boost pressure change on CO emission. Under all test conditions, the maximum CO emission was determined as 301 ppm in the use of D100 in the original boost pressure application. In comparison, the minimum CO emission was observed as 206 ppm in the use of E20B4 at a 30% increased boost pressure application.

Compared to neat D100 in all boost pressure applications, a significant reduction in CO emissions was observed with blends. With the addition of alcohol fuels to diesel, more oxygen will be in the cylinder, so more carbon atoms will react with oxygen. It is thought that the increased oxygen containing in blends improves combustion and causes less CO emission. In the use of E10B2, the CO emission values

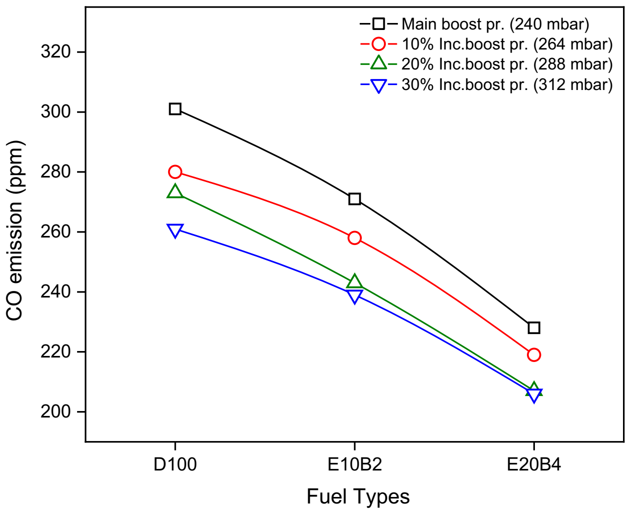


Fig. 6 The effect of the boost pressure on CO emissions

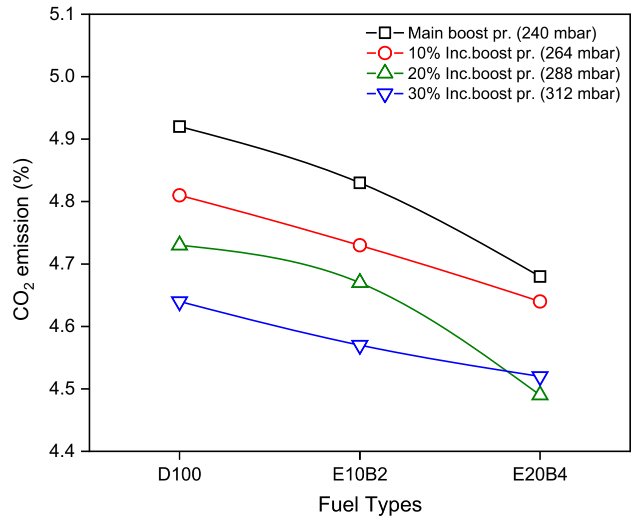


Fig. 7 The effect of the boost pressure on CO_2 emissions

released as a result of the original boost pressure application 10%, 20%, and 30% increased boost pressure applications were obtained 271, 258, 243, and 239 ppm, respectively. In D100 use, the minimum CO emission was observed as 261 ppm at 30% increased boost pressure application. In all boost pressure applications, minimum CO emission were seen in the use of E20B4. It is thought that as a result of the increase in the intake air pressure, the supply of sufficient oxygen to the cylinder via the intake manifold causes a decrease in the CO emission when the oxygen deficiency problem is solved. Baskar and Senthilkumar et al. [41] increased the intake air pressure; as a result, more oxygen was supplied to the cylinder, and they stated a decrease in CO emissions from oxygen enrichment. Patel et al. [42] noted that the reduction in rich fuel mixture with more air supply ensures proper combustion in the cylinder.

3.5 Comparison of carbon dioxide (CO_2) emission values

CO_2 emission is the exhaust emission resulting from the complete combustion of the fuel. Although it is not generally accepted as a polluting gas, it is the most critical gas that causes global warming. Global total CO_2 emissions in 2018 were determined as 33.5 billion tons. Total CO_2 emissions from the transportation sector, including cars, trucks, buses, trains, ships, airplanes, etc. have been estimated at 8 billion tons. This calculation shows that the transportation sector is responsible for 8 billion/33.5 billion = 24% of global CO_2 emissions [43]. Figure 7 shows the effect of boost pressures on CO_2 emission.

It was monitored that the CO_2 emission values obtained with the use of D100 under all boost pressure applications are higher than those of blended fuels. This can be explained by the fact that the C/H ratio of the D100 is higher than the C/H ratio of the fuel mixtures. Under all test conditions, the maximum CO_2 emission was achieved as 4.9% in the original boost pressure application in D100 use. In comparison, the lowest CO_2 emission was observed as 4.5% at a 20% increased boost pressure increase in E20B4 use. In applying 30% increased boost pressure, which is the maximum boost pressure increase, D100 released 4.64%, E10B2 4.57%, and E20B4 4.5% CO_2 emission. In the original boost pressure application, the lowest CO_2 emission was caused by E20B4 by 4.7%, while a slight increase in CO_2 emission was observed with the use of E10B2 and was 4.83%. By using the blended fuels, an increase in CO_2 formation can be expected due to more CO molecules finding the opportunity to react with O_2 in the cylinder. This study monitored that higher CO_2 was formed using D100 compared to the blended fuels. The lower carbon number of alcohol fuels compared to diesel fuel is considered effective in CO_2 trends.

3.6 Comparison of methane (CH_4) emission values

In order to reduce methane emissions, which are shown as the second most crucial emission causing global warming, new studies are planned to be carried out in the near future and limiting them with regulations [44]. Figure 8 shows the effect of the ethanol–diesel fuel mixtures and the boost pressure on CH_4 emission.

In the experiments, it has been observed that the CH_4 emissions from the use of D100 are higher than the CH_4 emissions from the use of fuel mixtures. As a result of the increase in boost pressure values for all fuel types, a decrease in CH_4 emissions was observed. Maximum CH_4 emission were observed as 3.64, 2.6, and 1.1 ppm for each fuel in the original boost pressure application, resulting from D100,

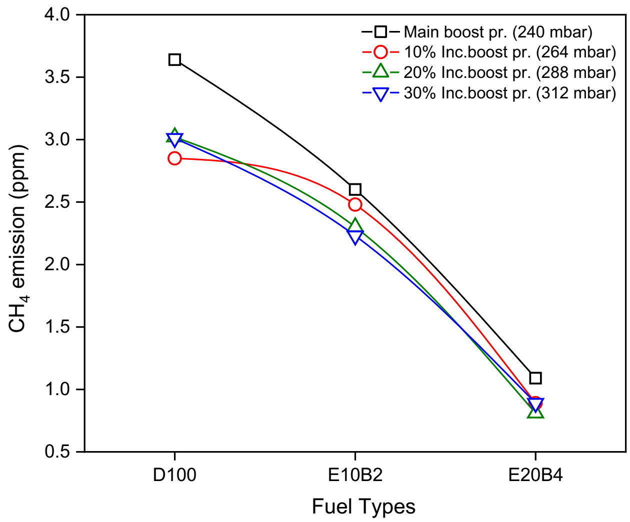


Fig. 8 The effect of the boost pressure on CH_4 emissions

E10B2, and E20B4, respectively. It was determined that the alcohol content in the mixture increased, and there was a decrease in the CH_4 emission. In all experimental conditions, minimum CH_4 emission was obtained as 0.8 ppm at 20% boost pressure in the use of E20B4. In the 30% boost pressure application, the maximum CH_4 emission was 3 ppm with the use of D100, while it was determined as 2.23 ppm in the use of E10B2 fuel and 0.9 ppm in the use of E20B4 fuel. With the 10% boost pressure application, the maximum CH_4 emission release was seen as 2.85 ppm because of the use of D100.

3.7 Comparison of nitrogen oxide (NOx) emission values

In general, NO_x emission is known as exhaust emission that occurs at high combustion frequencies depending on the oxygen amount and equivalence ratio in the cylinder and causes acid rain. Total NO_x emissions in turbocharged diesel engines are approximately composed of NO_2 emission in the range of 5–15% and NO emission in the range of 85–95% [45]. Figure 9 shows the effect of ethanol–diesel fuel mixtures and boost pressure change on NO_x emission and exhaust gas temperature (EGT). With the increase of the intake manifold pressure, the NO_x emission level when using E10B2 is higher than those of D100 and E20B4. In the main boost pressure application, it was observed that NO_x emissions from the use of an alcohol–diesel mixture were higher than that of D100. Minimum NO_x emission for all test fuels was obtained as 534 ppm for D100 at 30% increased boost pressure application, 538 ppm for E10B2, and 516 ppm for E20B4. Under all test conditions, maximum NO_x emission was observed as 559 ppm in E10B2 use. In the 20% increased boost pressure type, the minimum NO_x emission was found to be 525 ppm in the use of E20B4, while an increase in NO_x emission was detected in the use of D100 and E10B2, 543 ppm and 548 ppm, respectively. The oxygen in the fuel containing 10% ethanol improves combustion, so NO_x emissions have increased as a result of reaching high temperatures in the cylinder. On the other hand, as a result of the use of fuel with a 20% ethanol ratio, the latent heat of vaporization of the mixture fuel increased and the energy content decreased; therefore, the lower temperature in the cylinder caused a decrease in NO_x emissions [17, 46, 47].

In the tests carried out, it was observed that the exhaust gas temperature obtained with D100 was higher than the exhaust gas temperature obtained because of the use of mixed fuel. Compared to blended fuels, it is thought that higher exhaust gas temperatures are measured due to the longer combustion times and higher energy content due to using D100. Yun et al. [48] stated that adding alcohol fuel to diesel fuel reduces the energy of the mixtures, resulting in a decrease in exhaust gas temperatures. The minimum exhaust gas temperature for each fuel type was seen as 310 °C, 304 °C, and 287°CA (D100, E10B2, and E20B4, respectively) because of the 30% increased boost pressure application. The decrease in exhaust gas temperatures in all fuel types as a result of the increase in intake air pressure can be explained by the shortening of the duration of combustion in all fuel types.

4 Conclusions

In this study, it is seen that the addition of 2-butanol to ethanol–diesel fuel mixtures extended the phase separation time in the blended fuels and allowed the fuels to remain more homogeneous. In future studies, the use of 2-butanol as a stabilizer may be a good option for alternative fuel research. In the same boost pressure application, when using E10B2 and E20B4 compared to D100, an increase in combustion noise and ignition delay, a decrease in combustion duration was detected.

In all test conditions, it was found that the increased boost pressure and the use of alcohol–diesel blends rise the cylinder gas pressure. And a significant increase in the maximum HRR was detected with the use of blends compared to D100. Compared to the main boost pressure application, it has been determined that there is an increase in the maximum HRR values in the use of D100 in proportion to the boost pressure increase.

A significant reduction in CH_4, CO_2, and CO emissions was monitored in all test fuels with the boost pressure increase compared to the main boost pressure. In addition, compared to D100, CO emissions were reduced by more than 20% with alcohol–diesel blends in all boost applications. Compared to the D100, a slight increase in NO_x emissions was observed in the use of E10B2 in general, while a slight decrease in the use of E20B4 was observed. Compared to

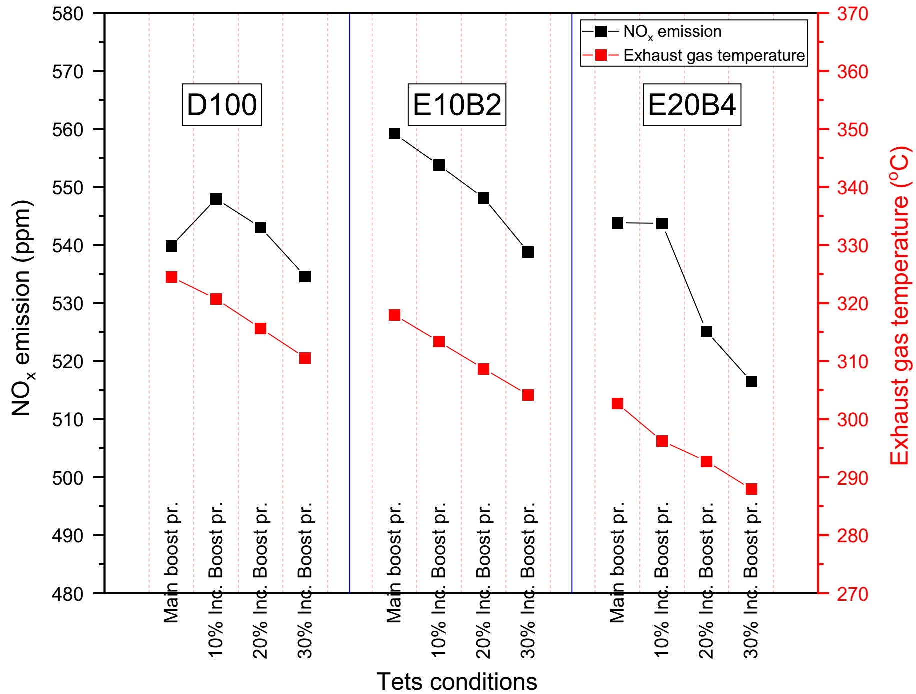


Fig. 9 The effect of the boost pressure on NO$_x$ emissions and exhaust gas temperatures

the main boost pressure, there was a significant reduction in NO$_x$ emission with a 30% increased boost pressure increase. According to the original boost pressure application, it has been determined that the exhaust gas temperature decreased by 5 °C on average for each 10% increase in boost pressure. At the same boost pressure, the exhaust gas temperature gradually decreased with an increase in the alcohol content in the mixture.

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Authors' contributions MV contributed to investigation, methodology, data processing, and writing—original draft. ANÖ was involved in writing—review and editing, conceptualization, visualization, and project administration. AT contributed to engine testing and writing—review and editing. CS was involved in writing—review and editing. İK contributed to writing—review and editing.

Data availability As shown in the below figure, AVL brand engine test system and emission measurement system are installed in our engine test room. The main methodologies used in the study are described in the methodology section of the article. The datasets used and/or analyzed during the current study are available from the corresponding author on reasonable request.

Declarations

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

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