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Investigation of performance, combustion and emission characteristics in a diesel engine fueled with methanol/ethanol/nHeptane/diesel blends

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ABSTRACT

One of the important reasons of exhaust emissions harmful to the environment and human health is the use of fossil fuels in internal combustion engines as energy resources. In this study, in order to research for cleaner fuel resources and to reduce dependence on fossil fuels, 20% methanol, ethanol and n-heptane fuels added by volume to fossil-based diesel fuel. The effects on engine performance, combustion and exhaust emission characteristics were investigated in a diesel engine with a 4-cylinder common rail injection system, at different engine loads (40 Nm and 80 Nm) and different engine speeds (1500 rpm, 1600 rpm, 1700 rpm and 1800 rpm). The maximum brake thermal efficiency (BTE) value was obtained as 43% with diesel-methanol (M20) mixed fuel at 1800 rpm at 80 Nm engine load. Brake specific fuel consumption (BSFC) values improved in all fuel types with the increase in engine load. In all test conditions, the highest maximum cylinder gas pressure (CP_{max}) value was obtained with M20 fuel as 114.3 bar, while the highest cumulative heat release (CHR_{max}) value was determined as 811.7 J with diesel-nheptane (H20) fuel. Compared to diesel fuel (D100), the use of alcohol-diesel fuel mixtures resulted in longer ignition delay (ID) and shortened combustion duration (CD). In general, a significant reduction in carbon dioxide (CO_2) emissions has been observed with the use of blended fuels. As a result of the increase in engine the load, a decrease in HC emissions was observed for all test fuels. When compared to D100 fuel, oxygen (O_2) and nitrogen oxide (NO) emissions were increased with the use of diesel-methanol (M20) and diesel-ethanol (E20) fuels, while O_2 and NO emissions were decreased with the use of diesel-nheptane fuel.

1. Introduction

Due to the rapid population growth in the world, environmental pollution is increasing as a result of the increase in energy supply. A significant part of the exhaust gases that cause environmental pollution is caused by fossil fuels used as an energy source in internal combustion engines. Researchers are constantly working on replacing fossil-based fuels with sustainable and cleaner energy sources that can be used in internal combustion engines. In this context, fuels with chemical properties close to fossil-based diesel fuel and oxygen-containing alcohol fuels come to the fore [1–5].

Some of the methods that can be used to reduce exhaust emissions in diesel engines can be listed as follows; fuel injection strategies, advanced combustion strategies, use of oxygenated fuel, use of exhaust after-treatment systems and use of alternative fuel mixtures. The most important and cost-effective alternative among these methods is the use of mixed fuels [6,7]. In addition, some of the emissions can be controlled as a result of coating the combustion chamber elements with chrome carbide [8,9]. It is seen that alcohol-diesel fuel mixtures significantly reduce exhaust emissions depending on the mixture ratio and significantly improve combustion efficiency [10,11]. It is reported that methanol and ethanol, which have low carbon numbers among alcohol fuels, can be used by preparing a mixture with fossil fuels due to their high cooling effect and high oxygen content [12] and it is known that it is possible to use them in the internal combustion engine without any changes. In addition, since methanol and ethanol can be produced locally from organic products, oil imports can be reduced by declining the use of fossil fuels [13]. The use of alcohol-diesel fuel

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mixtures can significantly control particulate matter (PM) emissions, which are harmful to human health [14]. Despite its advantages, one of the most important factors preventing the mixing of methanol and ethanol with diesel fuel is the limited solubility caused by the amount of water in the alcohol fuels and the phase separation seen accordingly [13,15–17]. Three different methods can be used to prevent phase separation. These methods are: spraying alcohol fuel to the intake manifold, adding co-solvent to diesel-alcohol fuel mixtures, or continuous mixing of diesel-alcohol fuel mixtures in the fuel tank with the help of a mixer [18–21]. In this study, prevention of a phase separation in methanol-diesel and ethanol-diesel mixed fuels was achieved by mixing the fuels in the fuel tank by means of a mixer.

N-heptane fuel, which has a straight chemical chain shown with the chemical formula C_7H_{16}, is used as a reference fuel because it has an octane number of zero. Due to its chemical properties being close to fossil-based diesel fuel, it has a high potential for use in diesel engines. It has properties close to fossil-based diesel fuel in terms of ignition characteristics, combustion characteristics and exhaust emission formation. It was stated that n-heptane fuel showed better combustion performance especially under low operating conditions [22–24].

Number of studies have been conducted in the literature on the use of methanol, ethanol, n-heptane fuels in diesel engines. Saravanan et al. [25] stated the effects of exhaust gas recirculation (EGR) on performance, combustion and emission characteristics in a diesel engine using an ethanol-diesel fuel mixture. Ethanol-diesel fuel mixtures increased the maximum heat release rates. Ignition delay increased with increasing engine speed. Emiroğlu et al. [26] investigated the effects of mixing different alcohols with 10% diesel fuel on combustion, performance and exhaust emissions in a diesel engine. They determined that the maximum heat release rate and maximum cylinder gas pressure values obtained with the use of mixture fuels are higher than the heat release rate and cylinder gas pressure values obtained as a result of the use of D100 fuel. In addition, the use of methanol-diesel blend fuel, the highest BSFC values were observed at all engine loads. Sayin et al. [27] reported the effects of using ethanol-diesel fuel mixtures on performance and emissions at different engine loads and injection times. Compared to D100 fuel, a slight increase was detected in nitrogen oxides (NO_x) emissions as a result of the use of ethanol-diesel fuel mixtures, while they achieved a significant reduction in CO and HC emissions. For all fuel types, BTE increased while BSFC decreased due to increasing engine load at the same engine speed.

Chen et al. [28] investigated the effect of using different rates of diesel-pentanol-methanol fuel mixtures on combustion and emission characteristics in a common rail diesel engine. Compared to D100 fuel, the combustion duration is reduced and the ignition delay is prolonged with the use of fuel mixtures. As a result of the use of blended fuels, NO_x emissions worsened, while soot emissions improved. Han et al. [29] reported the effects of n-butanol/n-heptane/diesel fuels on combustion and emissions in a diesel engine. Ignition delay decreased in all experiments as a result of increasing injection pressure. Cylinder gas pressure and heat release rate increased as a result of using n-butanol/n-heptane fuels at the same EGR rate. Zhang et al. [30] stated the effects of n-heptane and n-butanol fuel mixtures on combustion characteristics. It has been observed that the use of n-heptane improves the combustion of n-butanol and increases the combustion efficiency. In addition, HC and CO emissions decreased with the use of n-heptane. Guo et al. [31] investigated the feasibility of different rates of n-heptane/monohydrocarbylbenzene fuels in a diesel engine. Compared to D100 fuel, the use of n-heptane/monohydrocarbylbenzene blends significantly reduced soot emissions, while CO and HC emissions increased. In addition, some of the recent experimental and numerical studies on the use of n-heptane fuel by Refs. [32–36].

The target of this study is improving the performance and emissions of diesel engine running with alternative fuel blends and determining the optimum blend type. The tests were conducted at different loads and engine speed that are most common daily working conditions of a diesel engine. The idle speed of the engine is 865 rpm and the maximum speed is 6000 rpm. For this reason, the range of 1500 rpm–1800 rpm, which is the speed range in which the engine is used in general, was chosen. In addition, since the usability of these fuels, which are used by mixing with fossil fuels up to 10% by volume, may increase in the near future due to the increase in incentives for the use of alternative fuels and the tighter exhaust emission regulations, the alternative fuel ratio in the blended fuels has been deemed appropriate as 20% by volume [37]. In engines working with direct injection, it is recommended that the proportions of alcohol in the diesel-alcohol fuel mixture should not be more than 20% by volume, due to low miscibility and high self-ignition [38–40]. Moreover, as stated in the studies [41,42], the proportion of alternative fuels in the blended fuels was taken as 20% by volume, since mixing alcohol fuels with diesel fuel at high rates would cause phase separation. In the experiments, methanol and ethanol, which are widely studied and can be used in diesel engines [43], were taken. In the studies in the literature, n-heptane fuel was chosen because of its simple molecular structure and its ignition properties close to diesel fuel, since it was determined that n-heptane fuel was used by mixing it directly with diesel fuel and that it was deficient in its experimental examination. In addition, during the engine tests, the test conditions of 40 Nm and 80 Nm, which are 25% and 50% of the engine torque, respectively, were chosen as a constant engine load. This study examined the use of 20% methanol-diesel, ethanol-diesel and nheptane-diesel fuels in an internal combustion diesel engine to contribute it to the literature. As a result of the study, BSFC, cylinder gas pressure, cumulative heat release, pressure rise rate, CO_2 emission, HC emission, O_2 emission and NO emission results obtained with M20, E20 and H20 fuels were compared with the results obtained with D100 fuel, which is the reference fuel.

2. Experimental apparatus and procedure

The experiments were carried out in a turbocharged, four-stroke, four-cylinder diesel engine with Common Rail fuel injection system. The test bench consists of test engine, eddy current dynamometer, combustion analysis system, exhaust emission measurement system and Bosch fault detection systems as the main parts of the test setup. The schematic view of the experimental setup is given in Fig. 1.

The experiments were carried out at two different engine loads (40 Nm and 80 Nm) and at four different constant engine speeds (1500 rpm, 1600 rpm, 1700 rpm and 1800 rpm). The effects of different fuel mixtures (D100, M20, E20 and H20) on the performance, combustion and exhaust emissions of a diesel engine with a four-stroke common rail fuel injection system were investigated. The characteristics of the test engine used in the experiments are given in Table 1. The maximum power of the dynamometer used for loading the engine shown in the experiments is 160 kW, its maximum torque is 475 Nm and its maximum speed is 8000 rpm. The cooling process of the engine was carried out with the heat exchanger of the dynamometer. In addition, with the sensors connected to the control system of the dynamometer, engine coolant inlet and outlet temperatures, intake air and exhaust gas temperatures, and fuel temperatures can be measured and recorded. Load measurement can be made depending on the position of the accelerator pedal by entering the interface of the dynamometer

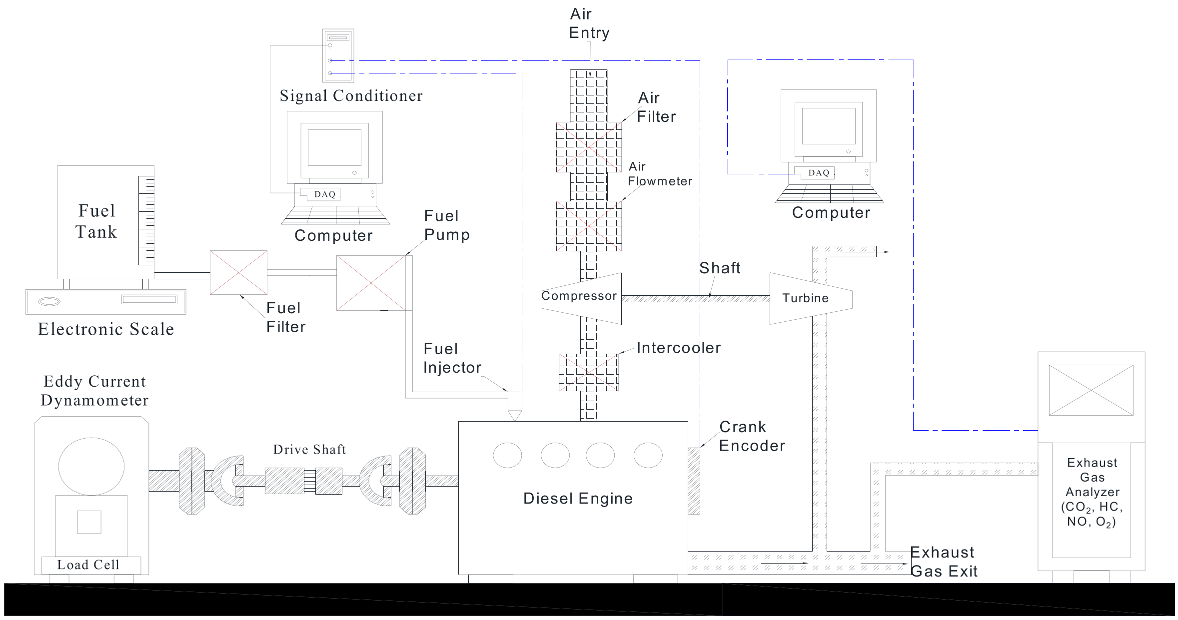

Fig. 1. Schematic view of the experimental setup.

Table 1
Specifications of the test engine.

Engine Type	In-Line
Cylinder Volume	1461 cm^3
Bore	76 mm
Stroke	80.5 mm
Connecting Rod Length	133.75 mm
Cylinder Number	4
Valves	8
Compression Rate	18.25:1
Maximum Power (4000 rpm)	48 kW (65 hp)
Maximum Torque (1750 rpm)	160 Nm
Fuel System	Common-Rail
Maximum Fuel Injection Pressure	1400 bar

according to the engine speed.

For in-cylinder gas pressure measurement, Oprant brand cylinder gas pressure sensor (sensitivity ±1%) is placed in the glow plug socket of the first cylinder of the test engine. In addition, a Kistler brand 4067C2000S model (sensitivity ±0.3333) fuel line pressure sensor is placed at the closest point of the injector connection on the high-pressure line. To determine the position of the crankshaft, Kübler brand, an encoder (accuracy ±0.1%) which produces 360 signals (pulses) per revolution of crank angle is placed on the crank pulley of the test engine. In the tests, Bosch brand BEA 460 model emission device was used for the measurement of exhaust emissions. The technical specifications of the exhaust emission device are given in Table 2.

Four different fuels were used in the experiments. The proportions of the fuels in the fuel mixtures were determined by volume and named according to the ratio of methanol, ethanol and n-heptane they contain. The first fuel type, D100 fuel, is 100% fossil-based diesel fuel obtained from a national fuel station. The second fuel type is M20 fuel containing 20% methanol +80% D100 fuel. The third fuel type used in the experiments is E20 fuel, which contains 20% ethanol +80% D100 fuel. The fourth type used is H20 fuel, which contains 20% n-heptane + 80% D100 fuel. After the preparation of the fuel mixtures, phase separation was observed. In the made observations, visible phase separation occurred in M20 and E20 fuels in a short time. In order to prevent phase separation and keep the fuels homogeneous, the mixed fuels in the fuel tank were mixed continuously. However, as a result of long observations, no visible phase separation was detected in H20 fuel. The properties of the fuels used in the experiments are given in Table 3. Moreover, properties of blended fuels are given in Table 4.

The results and combustion characteristics obtained in the experiments were calculated using the below equations.

$$\dot{m}f = \dot{V}f \ . \ \rho \tag{1}$$

where;

$\dot{m}f$: mass flow of fuel (kg/s)
$\dot{V}f$: volumetric flow rate of fuel (m^3/s)
ρ : fuel density (kg/m^3)

Table 2
Exhaust emission device specifications.

	Range	Sensibility
Carbon dioxide (CO_2)	0–18% vol.	0.01% vol.
Hydrocarbon (HC)	0–9999 ppm	1 ppm
Oxygen (O_2)	0–22% vol.	0.01% vol.
Nitrogen oxide (NO)	0–5000 ppm	1 ppm

Tablo 3
Properties of the fuels used in the experiments [30,44].

	Diesel	n-Heptane	Methanol	Ethanol
Chemical formula	$C_{12}H_{24}$	C_7H_{16}	CH_3OH	C_2H_5OH
Cetane number	40–55	56	3	8
Oxygen content (% weight)	–	–	50	34.8
Density (g/cm^3 @20 °C)	0.82–0.86	0.684	0.796	0.790
Auto-ignition temperature (°C)	200–220	–	470	434
Lower heating value (MJ/kg)	42.5	44.57	19.9	26.8
Boiling point (°C)	180–370	–	64.5	78.4
Latent heat (kJ/kg@25 °C)	270	317	1109	904
Viscosity (mm^2/s@40 °C)	1.9–4.1	–	0.59	1.08

Table 4
Properties of the blended fuels used in the experiments.

	D100	H20	M20	E20
Chemical formula	$C_{12}H_{24}$	$C_{11}H_{22.4}$	$C_{9\cdot8}H_{20}O_{0.2}$	$C_{10}H_{21\cdot2}O_{0.2}$
LHV (MJ/kg)	42.5	42.9	37.·98	39.36
Cetane Index	55	55.2	44.6	45.6
C (%)	85.7	85.5	83.5	83.1
H (%)	14.3	14.5	14.2	14.7
O (%)	–	–	2.3	2.2
Diesel (%)	100	80	80	80
n-Heptane (%)	–	20	–	–
Methanol (%)	–	–	20	–
Ethanol (%)	–		–	20

The results in Eq. (1) are replaced in Eq. (2) and the BSFC value is found.

$$BSFC = \frac{\dot{mf}}{Ne}.3600.\,10^3 \tag{2}$$

where;

BSFC: Brake specific fuel consumption (g/kWh)
Ne: Effective engine power (kW)

The *BTE* value is obtained by substituting the BSFC value obtained from Eq. (2) in Eq. (3).

$$BTE = \frac{Ne}{\dot{mf}\ .\ LHV}\ x\ 100 \tag{3}$$

where;

BTE: Brake thermal efficiency (%)
LHV: Lower heating value (kJ/kg)

Eq. (4) was used to calculate the pressure rise rate.

$$PRR = \frac{dP}{d\theta} \tag{4}$$

where;

PRR: Pressure rise rate (bar/^{o}CA)
P: Cylinder pressure (bar)
θ : Crank angle (^{o}CA)

Eq. (5) is used to calculate the ignition delay.

$$ID = \theta_{soc} - \theta_{soı} \tag{5}$$

where;

ID: Ignition delay (^{o}CA)
θ_{soc} : Start of combustion (^{o}CA)
$\theta_{soı}$: Start of injection (^{o}CA)

The heat release rate was obtained from the variation of the pressure in the cylinder according to the crank angle, and then the cumulative heat release was obtained by using the heat release rate values. The following equations are used for these calculations. The heat losses in the cylinder walls are not taken into account in the calculation of the heat release rate. The specific heat ratio is taken as 1.35 as a constant [45].

$$\frac{dQ}{d\theta} = \frac{k}{k-1}P\frac{dV}{d\theta} + \frac{1}{k-1}V\frac{dP}{d\theta} \tag{6}$$

where:

$\frac{dQ}{d\theta}$: Heat release rate (J/^{o}CA)
V: Cylinder volume (m^3)
k: Specific heat rate

$$Q_{cum} = \int_{\theta_{soc}}^{\theta_{eoc}} \frac{dQ}{d\theta}d\theta \tag{7}$$

where;

Q_{cum} : Cumulative heat release (J)
θ_{eoc} : End of combustion (^{o}CA)

Exhaust emission values were measured in ppm (particule per million) volumetrically, but in order to compare the results with similar studies in the literature, conversions to g/kWh units were performed using the following Eq. (8) [46].

$$Ep_i = Ev_i\frac{M_i}{M_{ex}}.\ \frac{m_{ex}}{Ne} \tag{8}$$

where;

Ep_i : Pollutant mass of the i component (g/kWh).
Ev_i : Exhaust emission value of components on dry basis, i, as volume share (ppm).
M_i : Molecular mass of the components (g/mol).
M_{ex} : Molecular mass of the exhaust gases (g/mol)
m_{ex} : Exhaust mass flow (kg/h)

Before the start of the engine tests, the engine coolant temperature is expected to reach 80–90 °C to ensure that the test engine operates steadily. The tests for each fuel type were repeated 3 times. For combustion analysis, 200 cycles were taken during each experiment. As a result of the tests, the cylinder gas pressure, cumulative heat release, pressure rise rate, BTE, BSFC, CO_2, HC, O_2 and NO values are measured by the results obtained with reference fuel.

3. Result and discussion

3.1. Combustion characteristics

In this study, the ignition delay was calculated from the point of the pilot injection started to at the point of the start of combustion. It is taken that the ignition delay from the start of the injection to the point where the heat release rate is positive. Also, the point at which the cumulative heat release is maximum is accepted as the end of combustion. Cetane number, latent heat of evaporation, oxygen content and viscosity of the fuel are the factors that directly affect the ignition delay and combustion duration [47]. Combustion characteristics such as CP_{max}, CHR_{max}, ID and CD are given in Table 5. In Fig. 2, the ignition delay changes of the fuel mixtures at 40 Nm and 80 Nm at different engine speeds are given.

In both engine loads, an increase in ignition delay was observed because of the engine speed increase. In the tests performed at the same engine load and engine speed, an increase in ignition delay was determined as a result of the use of M20 and E20 fuels compared to D100 and H20 fuels, and the maximum ignition delay was 18^{o}CA at 80 Nm engine load and 1800 rpm engine speed as a

Table 5
Combustion characteristics values.

Engine Load	Engine Speed	Fuel Types	CP_{max}	Point of CP_{max}	CHR_{max}	CD	ID
Nm	**rpm**		**bar**	**^oCA**	**J**	**^oCA**	**^oCA**
40	1500	D100	78.7	361	411.6	59	7
		M20	79.6	361	414.2	57	7
		E20	79.2	360	391.2	56	10
		H20	78.7	360	399.5	70	4
	1600	D100	81.2	360	450.4	70	8
		M20	82.4	361	464.3	57	9
		E20	80.6	373	460.5	54	13
		H20	81.3	362	576.4	61	8
	1700	D100	82.8	360	465	66	11
		M20	84.3	361	481.2	60	10
		E20	82	360	424.1	56	15
		H20	83.7	360	490.8	66	9
	1800	D100	86.2	361	486.8	65	11
		M20	87.1	360	453.1	59	11
		E20	86.2	360	449	56	15
		H20	88.1	360	512	67	8
80	1500	D100	110.8	373	790.2	83	13
		M20	112.4	372	698.3	75	13
		E20	111.9	372	701.5	70	15
		H20	108.5	373	803.4	83	13
	1600	D100	110.3	372	726.9	85	13
		M20	113.1	373	732.2	74	14
		E20	110.8	373	719.8	63	16
		H20	108.1	373	792.6	82	10
	1700	D100	114.1	372	722.7	85	12
		M20	113.6	373	775.9	68	13
		E20	110.7	373	759	62	16
		H20	110.8	374	893.8	71	10
	1800	D100	112.8	372	738.9	80	14
		M20	114.3	373	759	68	14
		E20	112.2	373	755.9	60	18
		H20	110	373	793.1	76	13

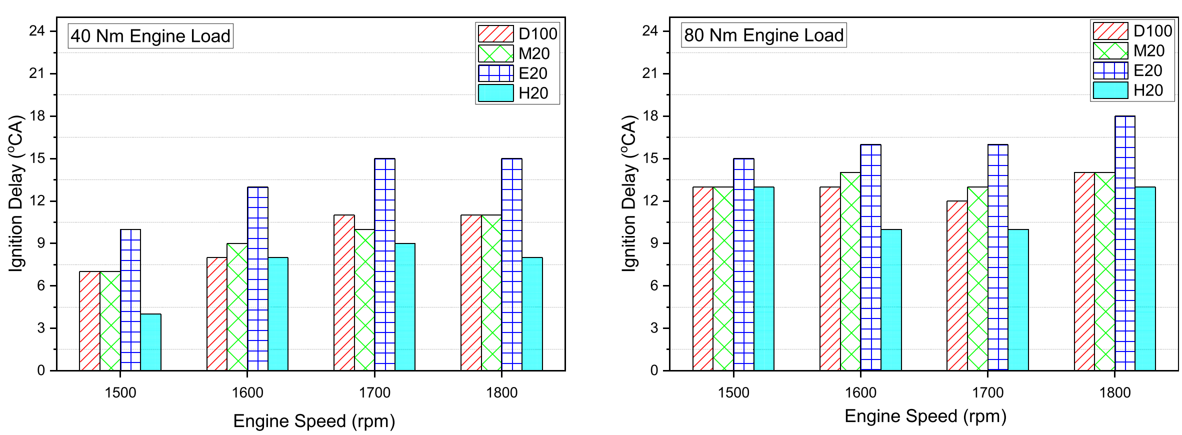

Fig. 2. Changes in ignition delay at different engine loads and speeds.

result of the use of E20 fuel. In the experiments carried out at 40 Nm engine load, at least 4^oCA increase in ignition delay was detected as a result of increasing the engine speed from 1500 rpm to 1800 rpm for all fuel types. In all the test conditions, a decrease in ignition delay was determined because of the use of H20 fuel, and the shortest ignition delay was obtained as 4^oCA in the tests performed at 40 Nm engine load and 1500 rpm engine speed in the use of H20 fuel. It is thought that the high latent heat of evaporation properties of alcohol fuels are effective as the reason for the increase in ignition delay as a result of diesel-alcohol fuel mixtures. While the fuel with a high latent heat of vaporization evaporates in the cylinder, it absorbs a lot of heat from the environment and a cooling effect occurs. As a result, it is thought that the ignition of the fuel-air mixture in the cylinder is delayed. Due to the high cetane number of H20 fuel, the ignition delay were reduced with the use of n-heptane-diesel fuel mixture. This is thought to be due to the high cetane number of n-heptane fuel. Although methanol fuel has a lower cetane number than ethanol, lower ignition delay was observed with M20 fuel, which can be explained by the low viscosity and high oxygen content of methanol.

The combustion duration changes of the fuel mixtures at 40 Nm and 80 Nm at different engine speeds are given in Fig. 3. When the combustion durations are examined, it has been observed that the combustion durations have decreased significantly as a result of the use of diesel-alcohol mixed fuels. For all the fuel types, it was determined that the combustion durations are prolonged because

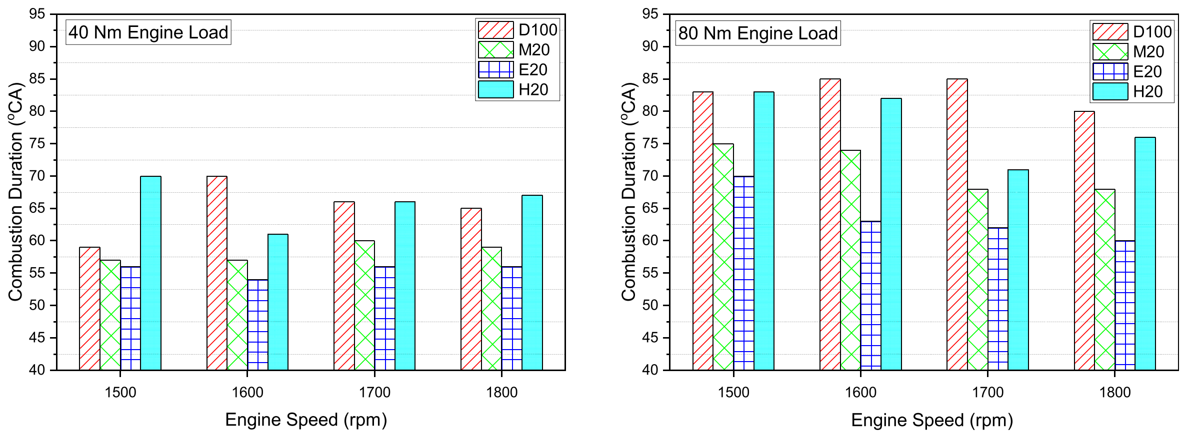

Fig. 3. Changes in combustion duration at different engine loads and speeds.

of the increase in engine load. It is thought that the reason for the increase in combustion duration due to the increase in engine load is the increase in the amount of the fuel sent into the cylinder. The maximum combustion period was obtained with the use of D100 fuel at 1600 rpm at 80 Nm engine load and 85°CA at 1700 rpm. In all test conditions, the shortest combustion durations were achieved by using E20 fuel. In the tests performed with E20 fuel at 40 Nm engine load, it was observed that the combustion durations were close to each other. In the experiments carried out at 80 Nm engine load and 1500 rpm, it was determined that the combustion duration reached the maximum times for M20, E20 and H20 fuel types (75°CA, 70°CA and 83°CA, respectively). In all the test conditions, the shortest combustion period was obtained as 54°CA at 40 Nm and 1600 rpm in the use of E20 fuel. It is thought that the reason why the obtained combustion durations are shorter than D100 and H20 fuels is because the oxygen they have in ethanol and methanol improves and accelerates the combustion in the cylinder. D100 and H20 fuels are heavy fuels due to their high carbon numbers and it is thought that their combustion durations are longer as a result of their use. When the ignition delay is extended, some of the fuel may be burned at this stage, so the combustion period may be shortened.

3.2. Brake thermal efficiency (BTE)

BTE is an indicator of how efficiently the chemical energy is converted into mechanical energy as a result of the combustion of the fuel in the cylinder [48]. In Fig. 4, the BTE changes of the fuel mixtures at 40 Nm and 80 Nm at different engine speeds are given. In the experiments, it was determined that the BTE values increased for all fuel types as a result of increasing the engine load. It was observed that the use of fuel mixtures prepared with alcohol fuels is more efficient than D100 and H20 fuels. This is thought to be due to the fact that the amount of oxygen in alcohol fuels improves combustion in the cylinder. In all the test conditions, BTE was found to be higher than other fuel types due to the usage of M20 fuel, and the maximum BTE value was obtained as 43% at 80 Nm engine load and 1800 rpm engine speed. At 80 Nm engine load and 1700 rpm, the BTE value was 32.7% with the use of D100 fuel, approximately 10% increase in the BTE value with the use of H20 fuel, 15% with the use of E20 fuel and more than 20% with the use of M20 fuel. In all experimental conditions, the lowest BTE was obtained as a result of using D100 fuel as 25.1% at 40 Nm engine load and 1800 rpm engine speed, while it was observed that BTE increased by approximately 40% with the use of M20 fuel under the same test conditions. It can be said that the BTE values determined by the use of H20 fuel are generally close to each other. It has been observed that BTE has improved as a result of the use of alcohol-diesel fuel mixtures, which is thought to be due to the increased ignition delay due to the fuels used, resulting in faster combustion and prevention of heat losses [49,50].

3.3. Brake specific fuel consumption (BSFC)

BSFC expresses the amount of the fuel consumed to produce a kW of power per hour and can increase or decrease depending on the lower heating value of the fuel used and the amount of oxygen [26]. Fig. 5 shows the BSFC changes of the fuel mixtures at 40 Nm and 80 Nm at different engine speeds. In all the test conditions, the lowest BSFC value was obtained as 220 g/kWh at 1800 rpm engine speed and 80 Nm engine load in the use of M20 fuel, while the maximum BSFC value in all the test conditions was approximately 340 g/kWh at 1800 rpm engine speed and 40 Nm engine load in the use of D100 fuel.

In general, a decrease in BSFC values was determined for other fuels except H20 fuel in the tests performed at 1600 rpm when compared to 1500 rpm at both engine loads. In the tests performed at 80 Nm engine load, an increase in BSFC values was observed in the tests performed at 1800 rpm engine speed compared to 1700 rpm for all the test fuels. At 1600 rpm engine speed and 40 Nm engine load, the lowest BSFC value was determined as 232.2 g/kWh in the use of M20 fuel, while the highest BSFC value was obtained as 279 g/kWh in the use of E20 fuel under the same test conditions. As a result of the use of H20 fuel, the BSFC values increased by increasing the engine speed from 1500 rpm to 1800 rpm in the tests performed at 40 Nm engine load. Compared to D100 fuel, BSFC values were found to be lower as a result of using blended fuels at 80 Nm at 1700 rpm and 1800 rpm engine speeds. With use of H20 fuel, the lowest BSFC values were obtained at 1500 rpm and 1700 rpm engine speeds in the tests performed at 80 Nm engine load. Despite the lower heating value of ethanol fuel, in general, ethanol-diesel fuel mixtures were found to cause lower BSFC, which is thought to be due to the high oxygen content of ethanol [51]. It was determined that BSFC values for all test fuels decreased significantly as a result of increased engine load. The reason for this decrease in BSFC, which occurs as a result of increasing engine load, can be explained by the high engine power obtained.

3.4. Cylinder gas pressure

Cylinder gas pressure expresses the pressure formed as a result of the combustion of the air-fuel mixture in the cylinder at the end

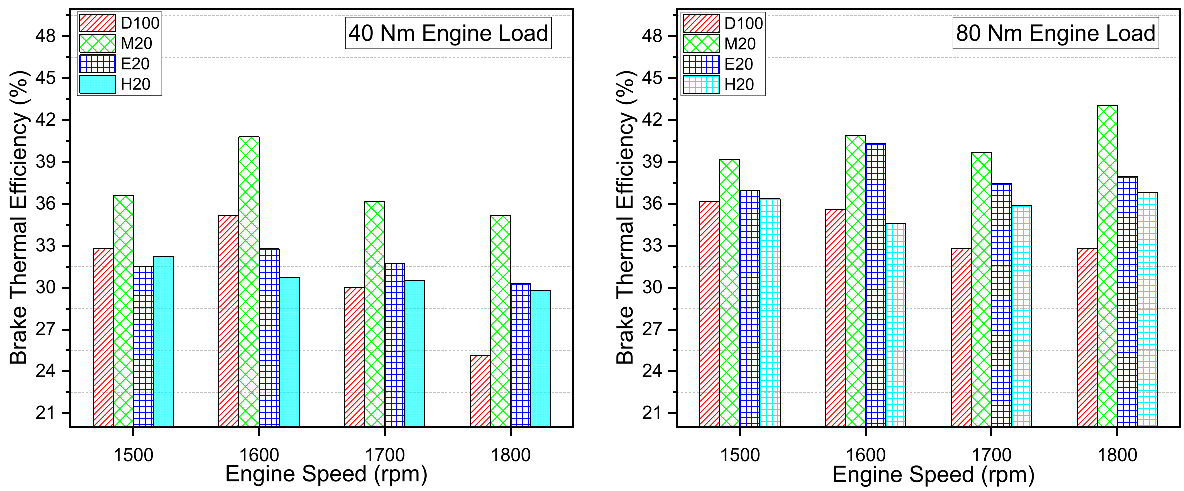

Fig. 4. Changes in brake thermal efficiency at different engine loads and speeds.

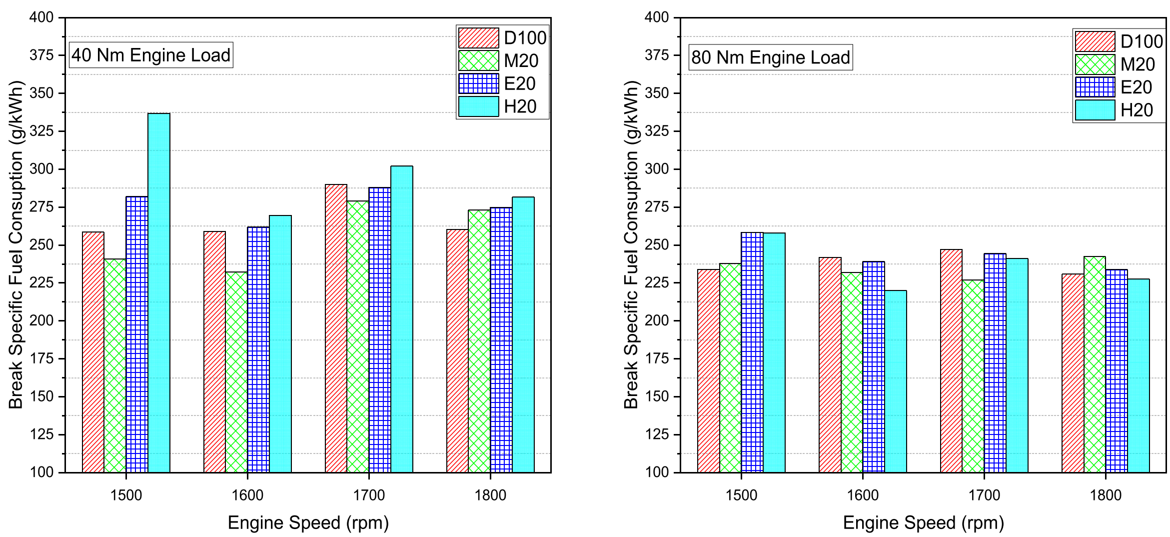

Fig. 5. Changes in brake specific fuel consumption at different engine loads and speeds.

of the compression time and the beginning of the expansion time. Since the gas pressure is directly dependent on the combustion of the injected fuel, the importance of the oxygen content, viscosity, latent heat of vaporization and cetane number of the fuel used becomes evident [26]. In Fig. 6, the effect of the mixed fuels used on the cylinder gas pressure at different engine speeds and engine loads is given. In all test conditions and fuel types, the highest maximum cylinder gas pressure value was obtained with the use of M20 fuel as 114.3 bar (at 373 ^{o}CA) at 80 Nm engine load and 1800 rpm engine speed. In general, in the tests performed at both engine loads for all the fuel types, an increase in cylinder gas pressure values was detected as a result of the increased engine speed. In the engine tests performed at 40 Nm constant engine load, the highest maximum gas pressure values for all the fuel types were obtained as 86.2 bar, 88.1 bar, 86.2 bar and 87.1 bar (D100, H20, E20 and M20, respectively) at 1800 rpm constant engine speed. As a result of increasing the engine load from 40 Nm to 80 Nm, the maximum cylinder gas pressure points were obtained more than 10 ^{o}CA late for all the fuel types. In the experiments performed at 80 Nm engine load and 1700 rpm engine speed, the highest maximum cylinder gas pressure value was determined as 114.1 bar at 372 ^{o}CA when using D100 fuel, while the lowest maximum cylinder gas pressure value was 110.7 bar at 373 ^{o}CA using E20 fuel under the same test conditions. In the experiments carried out at 40 Nm engine load, it was observed that the maximum cylinder gas pressure values obtained as a result of the use of H20 fuel mixture were higher than the cylinder gas pressure values obtained as a result of the use of D100 fuel. In the engine tests performed at 40 Nm engine load and 1800 rpm engine speed, the maximum cylinder gas pressure was obtained as 88.1 bar at 360 ^{o}CA with the use of H20 fuel, while the maximum cylinder gas pressure was obtained at 361 ^{o}CA as 86.2 bar with the use of D100 fuel. It was seen in the experiments carried out at 40 Nm engine load that, due to the low amount of injection, two peak points were formed for the cylinder gas pressure and it was determined that the first peak point was higher in general. In the tests carried out at 80 Nm engine load, single peak points were formed in the maximum cylinder gas pressure graphs for all fuel types and it was observed that these single peak points were higher than the peak points obtained as a result of the tests performed at 40 Nm engine load. It is thought that the increase in the maximum cylinder gas pressure values in the use of M20 and E20 fuel compared to D100 fuel is due

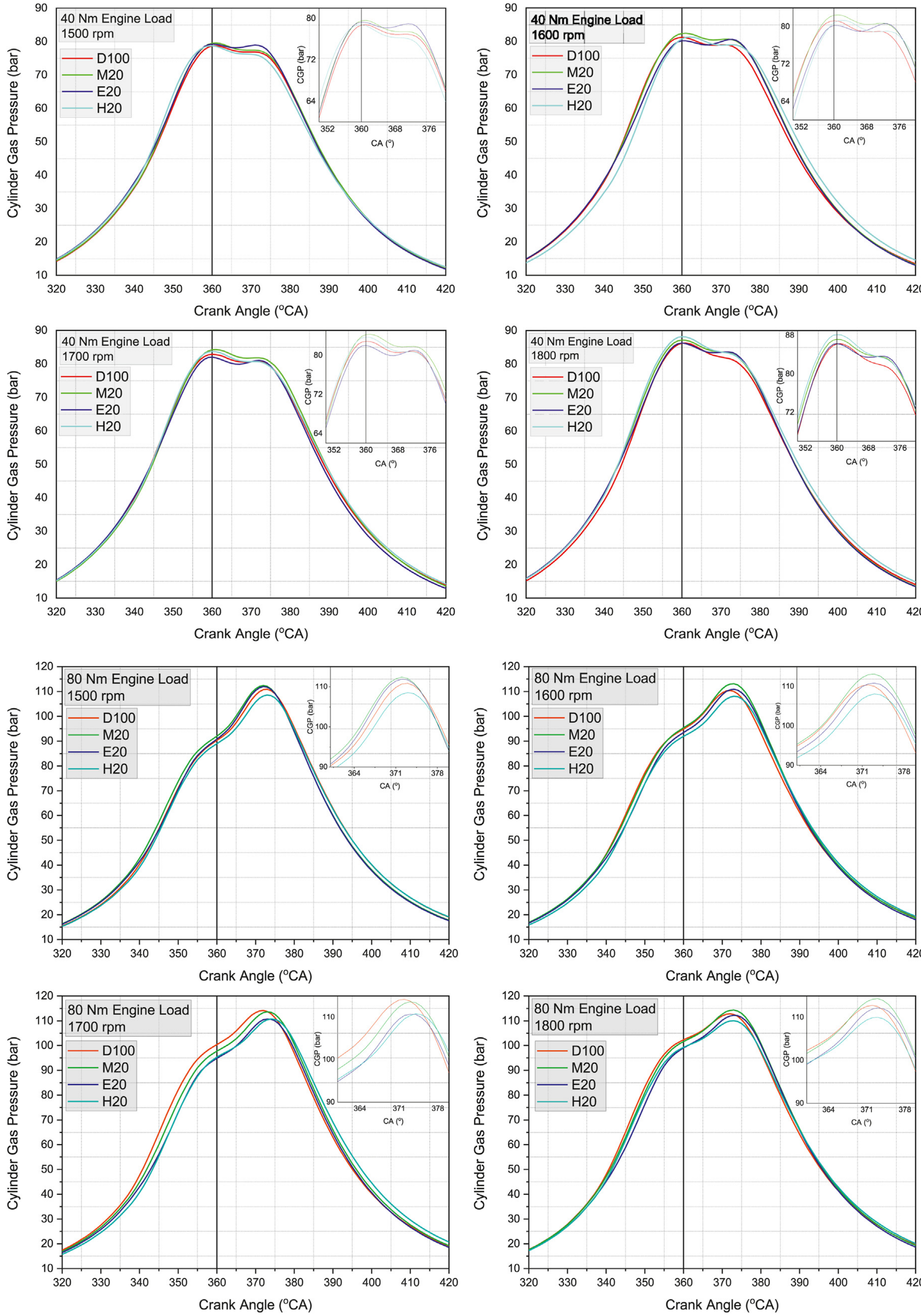

Fig. 6. Changes in cylinder gas pressure at different engine loads and speeds.

to the low viscosity of alcohol fuels, which improves atomization and the oxygen molecules they contain make combustion better.

3.5. Pressure rise rate

Pressure rise rate is used to express the pressure exerted on the piston at each crankshaft angle. The variation of the maximum pressure rise rate in dP/dθ should not exceed 10 bar/^{o}CA, otherwise knocking will occur and this may damage the engine [34]. In Fig. 7, the effect of the mixed fuels used on the pressure increase rate at different engine speeds and engine loads is given. For all fuel types, in the experiments carried out at 80 Nm engine load, it was observed that the pressure rise rates were higher than the pressure rise rates obtained from the experiments carried out at 40 Nm engine load. In the tests performed at 40 Nm engine load, it was calculated that the maximum pressure rise rates realized as a result of the use of H20 blend fuel were higher than the pressure increase rates released with use of D100 fuel. At 40 Nm engine load, 1500 rpm, 1600 rpm, 1700 rpm and 1800 rpm engine speeds, the pressure rise rates as a result of the use of E20 fuel were found to be lower and later than the pressure rise rates obtained by using other fuel types.

The highest maximum pressure rise rate in all test conditions was obtained at 346 ^{o}CA as 3.8 bar/^{o} in the tests performed at 1800 rpm engine speed and 80 Nm engine load in the use of D100 fuel. For all the fuel types, a tendency to increase in pressure rise rates was determined due to the increase in engine speed and engine load. In addition, as a result of increasing the engine load from 40 Nm to 80 Nm, the maximum pressure rise rate points were achieved earlier. The highest maximum pressure rise rate for all fuel types and test conditions occurred around the top dead centre (TDC). The fuel injection process of the test engine is divided into two as pilot and main fuel injection. Injected pilot fuel before TDC starts to burn around TDC. Pressure rise rates increase as combustion occurs in the decreasing cylinder volume towards the end of the compression period. Since the main fuel is burned in a controlled manner during the expansion time, the second peak becomes low. As the amount of fuel gets higher increase of the engine load, the pressure rise rates increase.

3.6. Cumulative heat release (CHR)

Cumulative heat release, which is an important indicator of combustion efficiency, is an important parameter for performance, combustion in the cylinder and exhaust emissions in general [52]. The total heat release occurring in the cylinder between the beginning and the end of combustion is defined as the cumulative heat release in this study. The cumulative heat release was calculated by adding up the amount of the heat released in each crank angle (CA) during a cycle. The calculated total heat amount gives important information about the effective combustion time. In Fig. 8, the variation of the cumulative heat release as a function of the crank angle is given as a result of the use of fuel mixtures at different engine speeds and engine loads. In addition, the experiments carried out at 40 Nm engine load and 1500 rpm, the highest maximum cumulative heat release was obtained as 414.2 J by using M20 fuel, while D100, H20 and E20 (411.6 J, 399.5 J and 391.2, respectively) fuels. For D100 fuel, an increase in cumulative heat release of more than 15% was observed as a result of the speed increase at 40 Nm. Therefore, the tests performed at 1600 rpm for both engine loads, it was determined that the cumulative heat release values obtained by using H20 and M20 fuels were higher than the cumulative heat release obtained by using D100 fuel. Among all experimental conditions and fuel types, the highest maximum cumulative heat release was obtained as 893.8 J at 410 ^{o}CA at 80 Nm engine load and 1700 rpm engine speed in the use of H20 fuel. Moreover, the addition of n-heptane fuel to fossil-based diesel fuel, the viscosity, cetane number and flash points of the fuel changed, and these changes were found to be compatible with the combustion characteristics of the H20 fuel. For all the fuel types, a significant increase in cumulative heat ratio values was observed as a result of doubling the engine load. Compared to D100 fuel, it was seen in the experiments performed at 80 Nm engine load that the cumulative heat rate values reached the peak earlier with the use of fuel mixtures. It is thought that the reason for the increase in CHR for all the fuel types due to the increase in engine load is that more fuel is sent into the cylinder at high engine loads.

3.7. Heat release rate (HRR)

In Fig. 9, the effect of the mixed fuels used on the cumulative heat rate increase rate at different engine speeds and engine loads is given. Considering all test conditions and fuel types, the highest heat release rate was measured as 44.3 J/^{o}CA (at 374 ^{o}CA) at 80 Nm engine load and 1700 rpm engine speed with the use of H20 fuel. In the tests carried out at 80 Nm engine load, an increase in the rate of heat release rate was obtained with the use of mixed fuels compared to D100 fuel for 1600 rpm, 1700 rpm and 1800 rpm engine speeds. At the same engine load test conditions, it was observed that the changes in engine speed did not have a significant effect on the heat release rate measured as a result of the use of E20 fuel. In addition, doubling the engine load, a significant increase in heat release rate was obtained for all fuels. In the tests carried out at 40 Nm engine load, the lowest heat release rate was determined at 1500 rpm for all fuel types. At 40 Nm engine load, the highest maximum heat release rate for E20 and H20 fuels were observed at 1600 rpm (28 J/^{o}CA and 28.7 J/^{o}CA, respectively). Compared to D100 fuel, it was determined that the maximum heat release rate was higher at 40 Nm engine load and at all engine speeds as a result of using M20 fuel.

3.8. CO_2 emission

CO_2 emission is known as the natural emission resulting from the combustion of hydrocarbon-based fuel. Although CO_2 emissions are not considered as polluting gas, they are among the most important factors causing the greenhouse effect [49]. Fig. 10 shows the effect of fuel mixtures on CO_2 emissions at 40 Nm and 80 Nm at different engine speeds. Compared to the tests performed at 80 Nm engine load, it was observed that the CO_2 emission values were higher at the same fuel type and at the same engine speed in the tests performed at 40 Nm engine load. In the experiments carried out at 80 Nm engine load, it was observed that the CO_2 emissions are lower as a result of the use of mixed fuels compared to the D100 fuel. In the experiments performed at 80 Nm engine load and 1800 rpm, the highest CO_2 emissions were obtained by using D100 as 740 g/kWh, at the same test conditions use of H20, E20 and M20 fuels (740 g/kWh, 711.5 g/kWh, 699.7 g/kWh and 677.6 g/kWh, respectively) caused lower CO_2 emission. While the lowest CO_2 emission at 40 Nm engine load was obtained as 770 g/kWh at 1600 rpm in the use of E20 fuel, the lowest CO_2 emission was determined in the use of M20 fuel as 677.6 g/kWh at 1800 rpm at 80 Nm engine load. In the experiments carried out at 1500 rpm and 80 Nm engine speed, it was observed that the CO_2 emission values

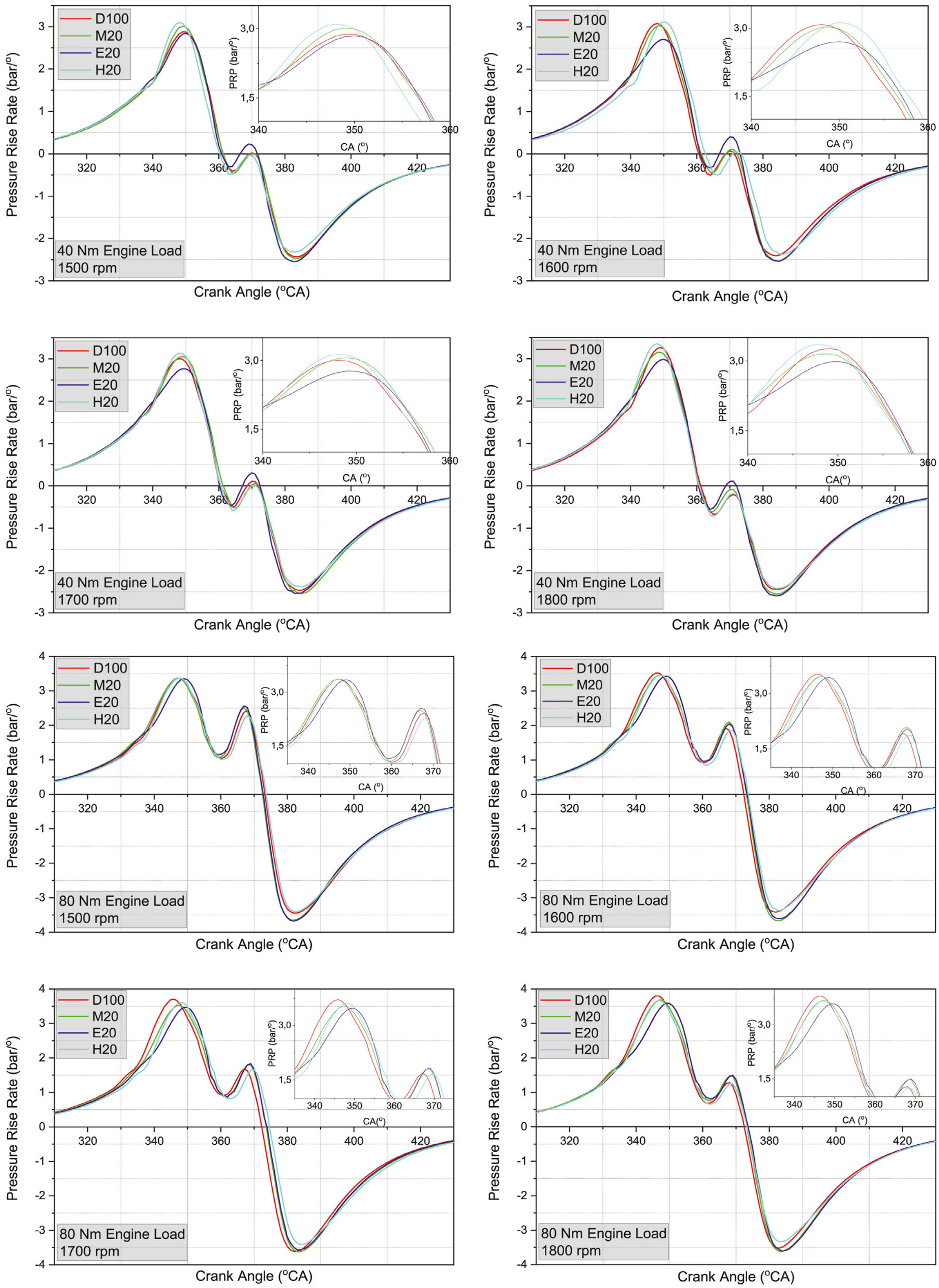

Fig. 7. Changes in pressure rise rate at different engine loads and speeds.

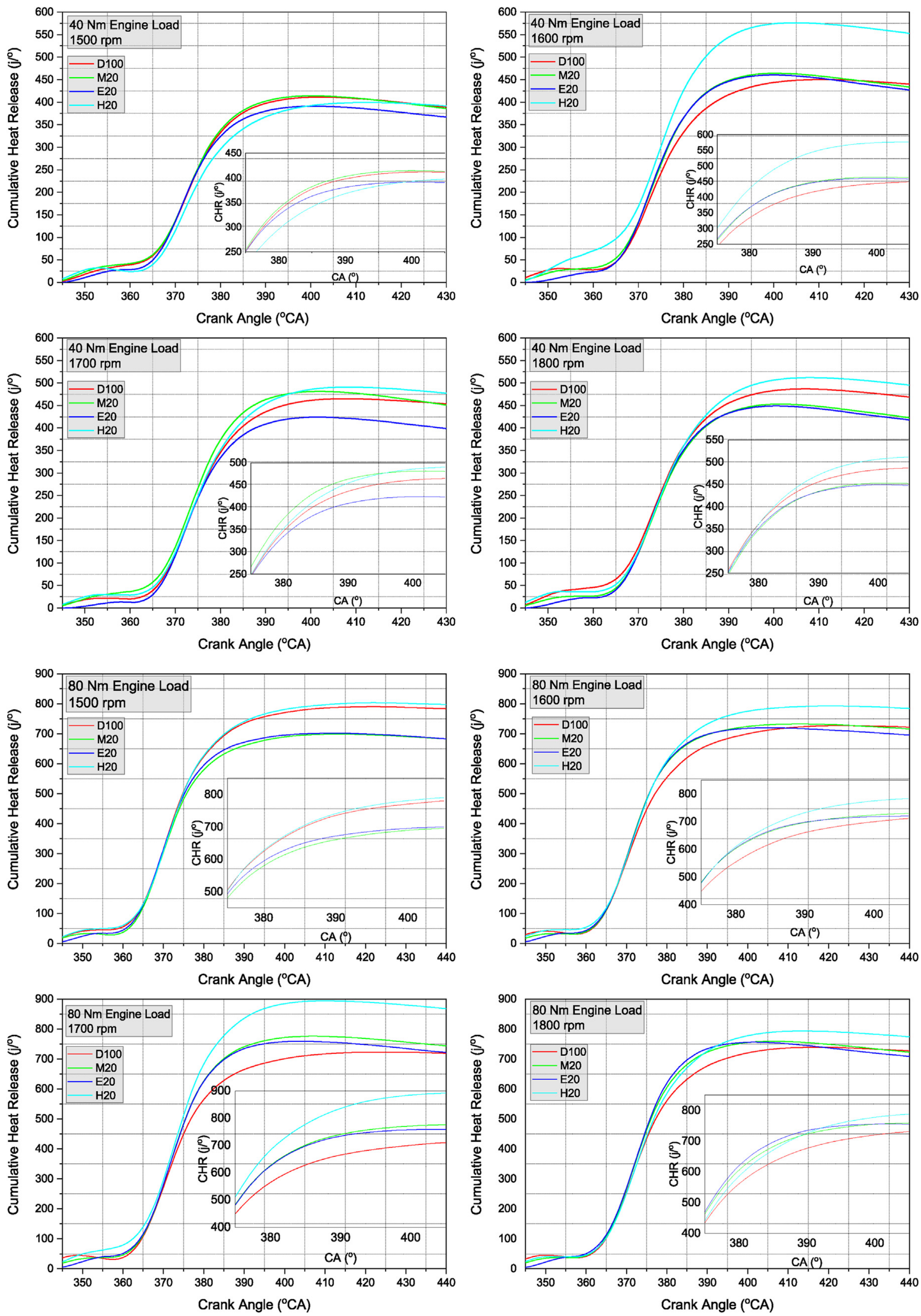

Fig. 8. Changes in cumulative heat release at different engine loads and speeds.

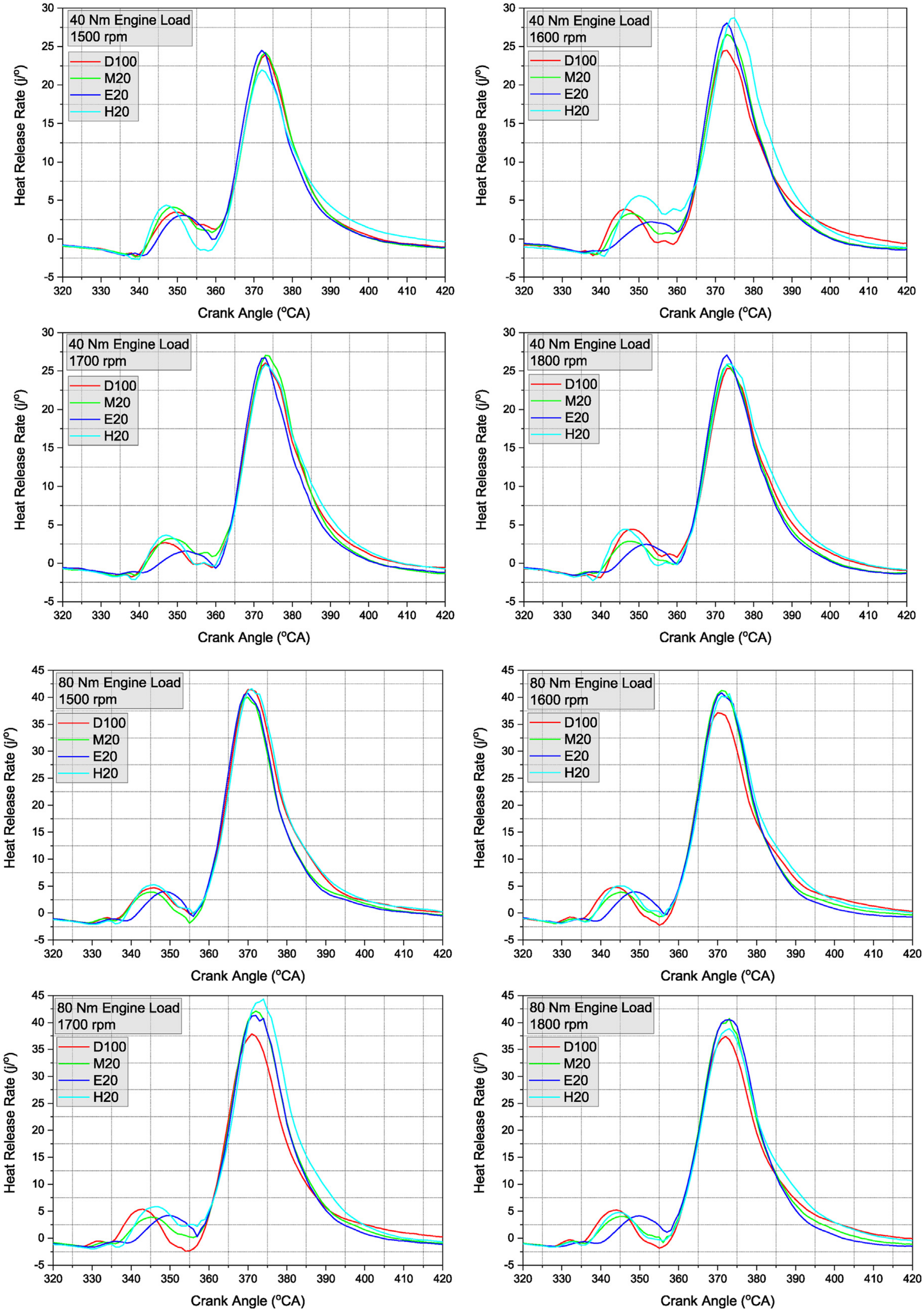

Fig. 9. Changes in heat release rate at different engine loads and speeds.

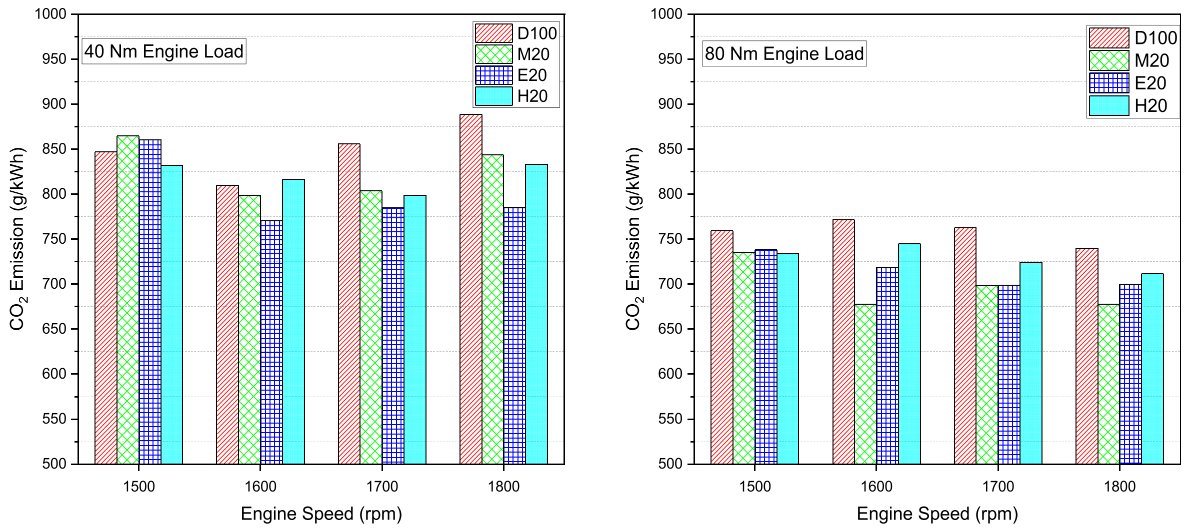

Fig. 10. Changes in CO_2 emission at different engine loads and speeds.

released because of the use of mixed fuels were closer to each other and lower than the CO_2 emissions released with use of D100 fuel. The reason for the higher CO_2 emissions than M20 and E20 fuels as a result of the use of D100 and H20 fuels can be explained by the high C/H ratio of D100 and H20 fuels. As the engine load increases, the amount of the fuel sent to the cylinder increases. In this case, while there was an increase in CO_2 emissions, the increase in engine power at the same time led to a decrease in specific CO_2 emissions.

3.9. HC emission

Hydrocarbon emissions generally refer to the fuel thrown out as a result of incomplete combustion in the cylinder and at low temperatures. While HC emissions increase due to the low temperature in the cylinder, NO emissions decrease at low temperatures in the cylinder due to the inability of the fuel to oxidize well. Towards the end of the combustion phase, the temperature near the cylinder walls is low mostly due to the heat loss [34,53]. The effect of fuel mixtures on HC emissions at 40 Nm and 80 Nm at different engine speeds is given in Fig. 11. In the experiments, it was observed that the test results at 80 Nm engine load were more stable than the test results at 40 Nm engine load. In all test conditions, it was determined that the HC emissions from the use of D100 fuel were lower than the HC emissions from the use of mixed fuels. In all the test conditions, the maximum HC emission was obtained as approximately 0.09 g/kWh at 40 Nm engine load and 1500 rpm engine speed with the use of E20 fuel, while the minimum HC emission was detected as 0.033 g/kWh with the use of D100 fuel at 80 Nm engine load and 1500 rpm engine speed. At engine load of 80 Nm, the highest HC emissions were seen at 1500 rpm, 1600 rpm and 1700 rpm engine speeds as a result of the use of H20 fuel (0.04 g/kWh, 0.045 g/kWh and 0.044 g/kWh, respectively). In the tests performed at 40 Nm engine load and 1600 rpm engine speed, the highest HC emissions were obtained as 0.071 g/kWh as a result of the use of M20 and E20 fuels, while the HC emission were 0.066 g/kWh with the use of H20 fuel and was found as 0.048 g/kWh with the use of D100 fuel. The highest HC

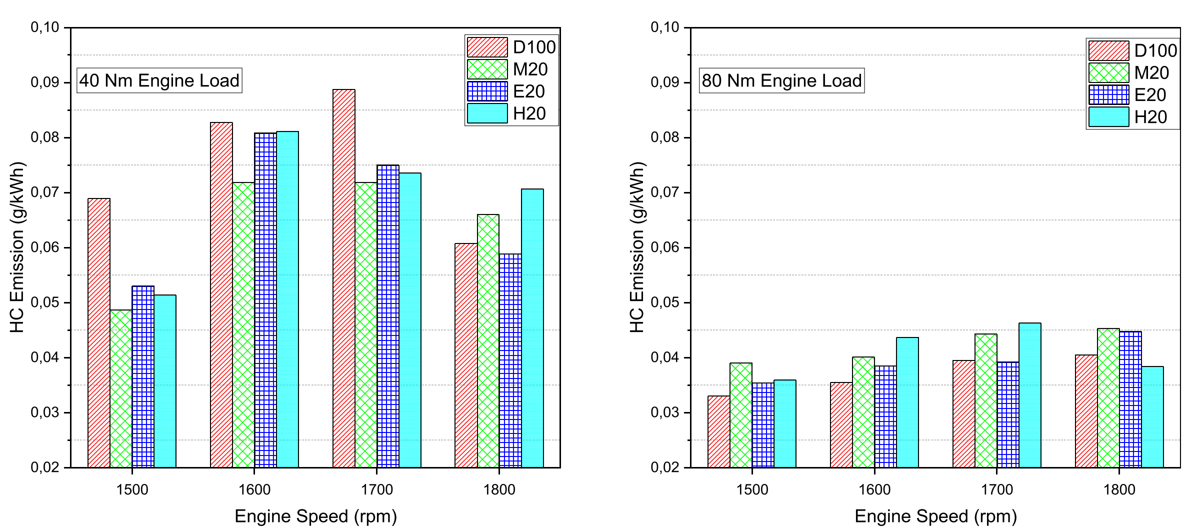

Fig. 11. Changes in HC emission at different engine loads and speeds.

emission at 1800 rpm engine speed at 40 Nm engine load was determined as 0.08 g/kWh in the use of M20 fuel. In the tests performed at 40 Nm engine load, HC emissions are thought to be lower because of better combustion in the cylinder as a result of using mixed fuels compared to D100 fuel. Compared to the experiments carried out at 40 Nm engine load, lower HC emissions were obtained for all the fuel types in the experiments carried out at 80 Nm engine load. This can be explained by the decrease in the amount of unburned fuel, since the combustion in the cylinder is more stable and better in the tests carried out at 80 Nm engine load.

3.10. O_2 emission

The effect of fuel mixtures on O_2 emissions at 40 Nm and 80 Nm at different engine speeds is given in Fig. 12. Compared to the experiments performed at 40 Nm engine load, a decrease in O_2 emissions was determined as a result of the tests performed at 80 Nm engine load for all the fuel types at the same engine speed. In the tests performed at 80 Nm engine load, an increase in O_2 emissions was observed since the increase in speed from 1500 rpm to 1800 rpm for all the fuel types. In the tests carried out at 80 Nm engine load, the highest O_2 emission at all the engine speeds was detected in the use of E20 fuel. In the experiments carried out at 40 Nm engine load, the lowest O_2 emissions were seen as 1717.5 g/kWh, 1747.4 g/kWh, 1778.4 g/kWh and 1810 g/kWh in the use of H20 fuel at all the engine speeds (1500 rpm, 1600 rpm, 1700 rpm and 1800 rpm, respectively). In the use of M20 and E20 fuels, a decrease in O_2 emissions was observed as a result of an increase in the engine speed from 1500 rpm to 1600 rpm, while a significant increase in O_2 emissions was observed due to the increased engine speed from 1600 rpm to 1700 rpm and 1800 rpm. In the experiments carried out at 80 Nm engine load and 1800 rpm engine speed, it was observed that the emissions from the use of M20 and H20 fuels were lower when compared to the emissions from the use of D100 fuel, while the emissions from the use of E20 fuel were higher. When the fuels were compared in all test conditions, the maximum O_2 emission was obtained as approximately 2202 g/kWh with M20 fuel at 40 Nm engine load and 1800 rpm engine speed. In general, an increase in O_2 emissions has been observed with the use of alcohol-diesel fuel mixtures, which can be explained by the oxygen contained in alcohol fuels. In addition, it is thought that the lower O_2 emission values of H20 fuel compared to D100 fuel, are result of the amount of air required for the combustion of n-heptane fuel with air, and higher than the amount of air required for the combustion of diesel fuel, hence less O_2 comes out of the combustion result.

3.11. NO emission

The formation of NO emission generally depends on the combustion temperature in the cylinder to around 1800 K and the oxygen concentration in the cylinder [34]. The effect of fuel mixtures on NO emissions at 40 Nm and 80 Nm at different engine speeds are given in Fig. 13. The lowest NO emission was obtained in the use of H20 fuel in all the experimental conditions. At 80 Nm engine load, a slight decrease in NO emission was detected as a result of the increase in the engine speed with the use of H20 fuel. In the tests carried out at 40 Nm engine load, a decrease in NO emission was observed as a result of increasing the engine speed from 1500 rpm to 1600 rpm for all fuels. In all test conditions, maximum NO emission was observed as 5.8 g/kWh with using E20 and M20 fuels at 80 Nm engine load and 1500 rpm engine speed, while minimum NO emission was determined as 2.7 g/kWh at 40 Nm engine load and 1700 rpm. Compared to D100 fuel at 40 Nm engine load, NO emission was observed to be higher at 1500 rpm and 1600 rpm engine speeds with using M20 and E20 blended fuels. The highest NO emission in the use of D100 fuel was measured as 5.2 g/kWh at 1600 rpm engine speed and 80 Nm engine load. As a result of the use of H20 fuel, NO emission was determined as approximately 4 g/kWh at 80 Nm at 1800 rpm engine speed, while NO emission was obtained as 3 g/kWh with a decrease of approximately 25% at 40 Nm. It was determined that NO emissions are higher as a result of increasing the engine load from 40 Nm to 80 Nm at the same engine speed and for the same fuel type. For all the fuels, it is thought that the increase in the in-cylinder combustion temperature is the reason for the significant increase in NO emissions, which are the result of the increase in engine load. In addition, since they improve combustion due to their oxygen content, the use of M20 and E20 fuels resulted in an increase in NO emissions in general when compared to D100 fuel, while a decrease in NO emissions was observed with use of H20 fuel. The decrease in NO emissions can be explained by the fact that the combustion in the cylinder occurs at a lower temperature, due to the deterioration of the atomization of the fuel in the cylinder because of the low viscosity of the H20 fuel.

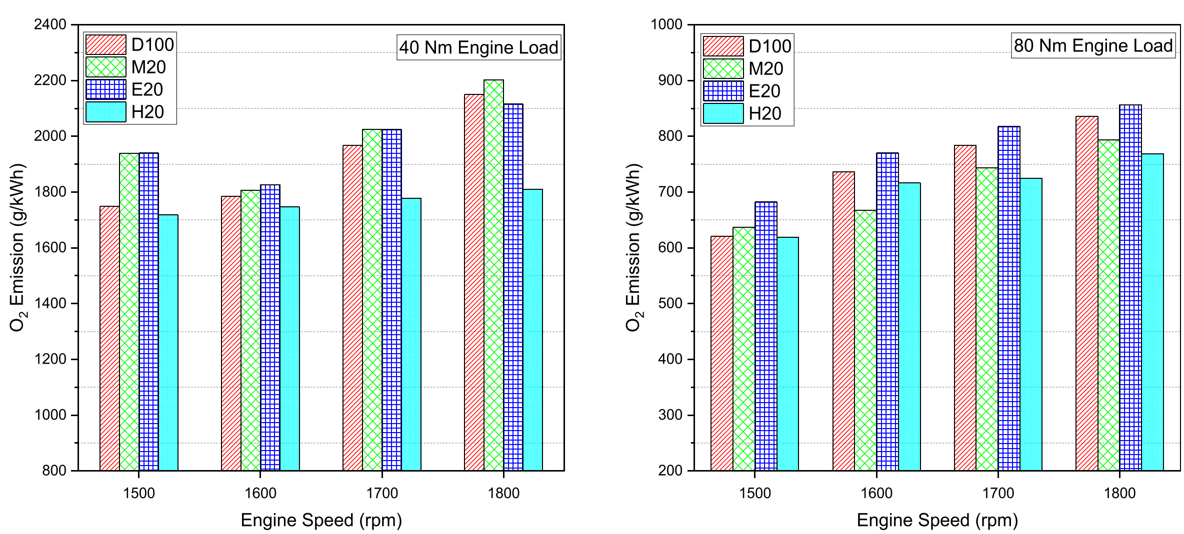

Fig. 12. Changes in O_2 emission at different engine loads and speeds.

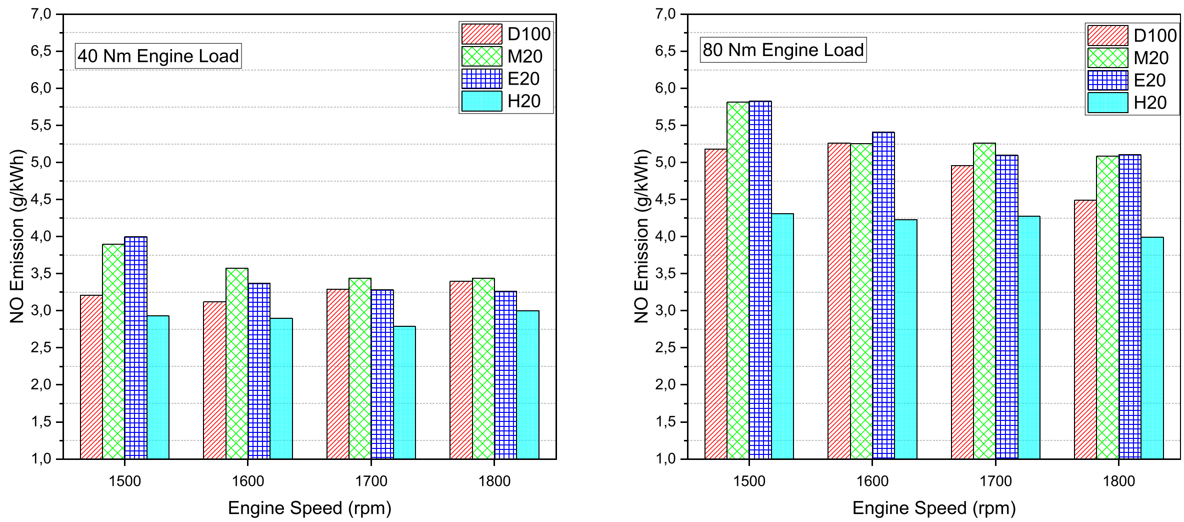

Fig. 13. Changes in NO emission at different engine loads and speeds.

4. Conclusion

In this study, in a common rail diesel engine use of H20, M20 and E20 fuels prepared by mixing n-heptane, methanol and ethanol fuels added to fossil-based diesel fuel at a rate of 20% by volume, at different engine loads (40 Nm and 80 Nm) and at different engine speeds (1500 rpm, 1600 rpm, 1700 rpm and 1800 rpm). It is aimed to contribute to the literature on the usability of alternative fuels by examining the engine performance, combustion and exhaust emission characteristics with the experiments carried out by comparing the data obtained as a result of the use of D100 fuel, which is the reference fuel.

- When mixing fossil-based diesel fuel with methanol and ethanol at a rate of 20% by volume, it is recommended to mix the blends continuously with a mixer in the fuel tank. On the other hand, it was observed that there was no visible phase separation as a result of mixing n-heptane with fossil-based diesel fuel.
- Compared to D100 fuel, the ignition delay increased up to about 30% with M20 and E20 fuels, and decreased by more than 10% with H20 fuel. The increase in the engine speed and engine load generally resulted in longer ignition delays for all the fuel types. It was determined that the combustion durations of M20, E20 fuels are reduced by more than 20°CA.
- It was seen that the use of M20 fuel mixture significantly improves BTE values. As a result of the use of H20 fuel and D100 fuel, it was observed that the BTE values were close to each other. Although it has the lowest lower heating value, up to 30% improvement in BSFC values were detected with the use of M20 fuel in some test conditions.
- With the increase in engine load and engine speed, the maximum cylinder gas pressure values were improved for all fuel types. The highest maximum cylinder gas pressure value was determined as 114.3 bar at 1800 rpm and 80 Nm engine load in the use of M20 fuel. Although it was seen that the pressure rise rates are generally close to each other, it was noted that the lowest pressure rise rates at 40 Nm and 80 Nm engine loads are realized as a result of the use of E20 fuel.
- For all the fuel types, more than 65% increases in cumulative heat release rates were determined due to the increase in the amount of the fuel entering the cylinder with the increase in engine load. In order to make comparisons between fuels at the same engine load and engine speed, a regular trend was not observed and at some points, higher cumulative heat release rates with fuel mixtures were found, while at some points higher cumulative heat release rates were obtained with D100 fuel.
- More than 10% reduction was found in CO_2 emissions with increased engine load for all the fuel types. In general, the highest CO_2 emissions were obtained as a result of the use of D100 fuel, which has a high C/H ratio, while the maximum CO_2 emission was found to be approximately 888 g/kWh when using D100 fuel at 40 Nm engine load and 1800 rpm. It was observed that HC emissions decreased significantly with the increase of the engine load. In the tests performed at 40 Nm engine load, more than 20% reduction was achieved with the use of blended fuels compared to D100 fuel.
- Due to the oxygen content, an increase in O_2 emissions was detected in the use of E20 and M20 fuels, compared to D100 and H20 fuels. The maximum O_2 emission was observed to be more than 2200 g/kWh as a result of the use of M20 fuel, which is the fuel with the highest oxygen content.
- With the use of M20 and E20 fuels, NO emissions were generally higher than D100 and H20 fuels, and the highest NO emissions were measured as 5.82 g/kWh with E20 fuel and 5.81 g/kWh with M20 fuel at 80 Nm engine load, respectively.

Credit author statement

Mustafa Vargün: Conceptualization, Methodology, Visualization, Investigation, Writing – original draft preparation. Ilker Turgut Yılmaz: Conceptualization, Methodology, Writing- Reviewing and Editing, Cenk Sayın: Writing- Reviewing and Editing, Supervision.

Declaration of competing interest

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

Data availability

The authors are unable or have chosen not to specify which data has been used.

References

[1] Sarıdemir S, Etem Gürel A, Ağbulut Ü, Bakan F. Investigating the role of fuel injection pressure change on performance characteristics of a DI-CI engine fuelled with methyl ester. Fuel 2020;271:117634. https://doi.org/10.1016/J.FUEL.2020.117634.
[2] Gupta A, Verma JP. Sustainable bio-ethanol production from agro-residues: a review. Renew Sustain Energy Rev 2015;41:550–67. https://doi.org/10.1016/J.RSER.2014.08.032.
[3] Tunç N, Karagöz M, Çiftçi B, Deniz E. Environmental pollution cost analysis of a diesel engine fueled with biogas-diesel-tire pyrolytic oil blends. Eng. Sci. Technol. Int. J. 2021;24:631–6. https://doi.org/10.1016/J.JESTCH.2020.10.008.
[4] Arias S, Molina F, Palacio R, López D, Agudelo JR. Assessment of carbonyl and PAH emissions in an automotive diesel engine fueled with butanol and renewable diesel fuel blends. Fuel 2022;316:123290. https://doi.org/10.1016/J.FUEL.2022.123290.
[5] Chia SR, Nomanbhay S, Ong MY, Shamsuddin AH bin, Chew KW, Show PL. Renewable diesel as fossil fuel substitution in Malaysia: a review. Fuel 2022;314:123137. https://doi.org/10.1016/J.FUEL.2022.123137.
[6] Wang X, Wang Y, Bai Y, Wang P, Zhao Y. An overview of physical and chemical features of diesel exhaust particles. J Energy Inst 2019;92:1864–88. https://doi.org/10.1016/j.joei.2018.11.006.
[7] Ghadikolaei MA, Wong PK, Cheung CS, Ning Z, Yung KF, Zhao J, et al. Impact of lower and higher alcohols on the physicochemical properties of particulate matter from diesel engines: a review. Renew Sustain Energy Rev 2021;143. https://doi.org/10.1016/j.rser.2021.110970.
[8] Hazar H, Gul H. Modeling analysis of chrome carbide (Cr3C2) coating on parts of combustion chamber of a SI engine. Energy 2016;115:76–87. https://doi.org/10.1016/j.energy.2016.08.083.
[9] Hazar H, Telceken T, Sevinc H. An experimental study on emission of a diesel engine fuelled with SME (safflower methyl ester) and diesel fuel. Energy 2022;241. https://doi.org/10.1016/j.energy.2021.122915.
[10] Yusuf AA, Inambao FL. Progress in alcohol-gasoline blends and their effects on the performance and emissions in SI engines under different operating conditions. Int J Ambient Energy 2021;42:465–81. https://doi.org/10.1080/01430750.2018.1531261.
[11] Pan W, Yao C, Han G, Wei H, Wang Q. The impact of intake air temperature on performance and exhaust emissions of a diesel methanol dual fuel engine. Fuel 2015;162:101–10. https://doi.org/10.1016/j.fuel.2015.08.073.
[12] Balat M, Balat H, Öz C. Progress in bioethanol processing. Prog Energy Combust Sci 2008;34:551–73. https://doi.org/10.1016/j.pecs.2007.11.001.
[13] Çelebi Y, Aydın H. An overview on the light alcohol fuels in diesel engines. Fuel 2019;236:890–911. https://doi.org/10.1016/j.fuel.2018.08.138.
[14] Mohankumar S, Senthilkumar P. Particulate matter formation and its control methodologies for diesel engine: a comprehensive review. Renew Sustain Energy Rev 2017;80:1227–38. https://doi.org/10.1016/j.rser.2017.05.133.
[15] Sarathy SM, Oßwald P, Hansen N, Kohse-Höinghaus K. Alcohol combustion chemistry. Prog Energy Combust Sci 2014;44:40–102. https://doi.org/10.1016/j.pecs.2014.04.003.
[16] Yusri IM, Mamat R, Najafi G, Razman A, Awad OI, Azmi WH, et al. Alcohol based automotive fuels from first four alcohol family in compression and spark ignition engine: a review on engine performance and exhaust emissions. Renew Sustain Energy Rev 2017;77:169–81. https://doi.org/10.1016/j.rser.2017.03.080.
[17] Jin C, Pang X, Zhang X, Wu S, Ma M, Xiang Y, et al. Effects of C3–C5 alcohols on solubility of alcohols/diesel blends. Fuel 2019;236:65–74. https://doi.org/10.1016/j.fuel.2018.08.129.
[18] Huang J, Wang Y, Li S, Roskilly AP, Yu H, Li H. Experimental investigation on the performance and emissions of a diesel engine fuelled with ethanol-diesel blends. Appl Therm Eng 2009;29:2484–90. https://doi.org/10.1016/j.applthermaleng.2008.12.016.
[19] Zhang ZH, Tsang KS, Cheung CS, Chan TL, Yao CD. Effect of fumigation methanol and ethanol on the gaseous and particulate emissions of a direct-injection diesel engine. Atmos Environ 2011;45:2001. https://doi.org/10.1016/j.atmosenv.2010.12.019. –8.
[20] Asfar KR, Hamed H. Combustion of fuel blends, vol. 39; 1998.
[21] Zhang Z, Li J, Tian J, Dong R, Zou Z, Gao S, et al. Performance, combustion and emission characteristics investigations on a diesel engine fueled with diesel/ethanol/n-butanol blends. Energy 2022;249. https://doi.org/10.1016/j.energy.2022.123733.
[22] Celik M, Solmaz H, Serdar Yucesu H, Yilmaz E. Examination of the effects of N-heptane addition to the canola methyl ester on engine performance and combustion characteristics. J Therm Sci Technol. 2016;36:9–16.
[23] Durgun O. İçten yanmalı motorlar temel ilkeler. TMMOB Gemi Mühendisleri Odası; 2018.
[24] Kolaitis DI, Founti MA. On the assumption of using «-heptane as a "surrogate fuel" for the description of the cool flame oxidation of diesel oil. Proc Combust Inst 2009;32 II:3197–205. https://doi.org/10.1016/j.proci.2008.06.073.
[25] Saravanan P, Kumar NM, Ettappan M, Dhanagopal R, Vishnupriyan J. Effect of exhaust gas re-circulation on performance, emission and combustion characteristics of ethanol-fueled diesel engine. Case Stud Therm Eng 2020;20:100643. https://doi.org/10.1016/J.CSITE.2020.100643.
[26] Emiroğlu AO, Şen M. Combustion, performance and emission characteristics of various alcohol blends in a single cylinder diesel engine. Fuel 2018;212:34–40. https://doi.org/10.1016/j.fuel.2017.10.016.
[27] Sayin C, Canakci M. Effects of injection timing on the engine performance and exhaust emissions of a dual-fuel diesel engine. Energy Convers Manag 2009;50:203–13. https://doi.org/10.1016/j.enconman.2008.06.007.
[28] Chen H, Su X, He J, Xie B. Investigation on combustion and emission characteristics of a common rail diesel engine fueled with diesel/n-pentanol/methanol blends. Energy 2019;167:297–311. https://doi.org/10.1016/j.energy.2018.10.199.
[29] Han J, Wang S, Maria Vittori R, Somers LMT. Experimental study of the combustion and emission characteristics of oxygenated fuels on a heavy-duty diesel engine. Fuel 2020;268. https://doi.org/10.1016/j.fuel.2020.117219.
[30] Zhang C, Zhang C, Xue L, Li Y. Combustion characteristics and operation range of a RCCI combustion engine fueled with direct injection n-heptane and pipe injection n-butanol. Energy 2017;125:439–48. https://doi.org/10.1016/j.energy.2017.02.148.
[31] Guo X, Deng B, Huang H, Lei H, Tong C, Huang R, et al. The feasibility study of single-hydrocarbylbenzene/n-heptane model diesel fuels for diesel characterization. Fuel 2021;288. https://doi.org/10.1016/j.fuel.2020.119841.
[32] Wang H, Deneys Reitz R, Yao M, Yang B, Jiao Q, Qiu L. Development of an n-heptane-n-butanol-PAH mechanism and its application for combustion and soot prediction. Combust Flame 2013;160:504–19. https://doi.org/10.1016/j.combustflame.2012.11.017.
[33] Li R, Liu Z, Han Y, Tan M, Xu Y, Tian J, et al. Experimental study of the combustion and emission characteristics of ethanol, diesel-gasoline, n-heptane-iso-octane, n-heptane-ethanol and decane-ethanol in a constant volume vessel. Fuel 2018;232:233–50. https://doi.org/10.1016/j.fuel.2018.05.148.
[34] Calam A, Aydoğan B, Halis S. The comparison of combustion, engine performance and emission characteristics of ethanol, methanol, fusel oil, butanol, isopropanol and naphtha with n-heptane blends on HCCI engine. Fuel 2020;266. https://doi.org/10.1016/j.fuel.2020.117071.
[35] Sabnis P, Aggarwal SK. A numerical study of NOx and soot emissions in methane/n-heptane triple flames. Renew Energy 2018;126:844–54. https://doi.org/10.1016/j.renene.2018.04.007.
[36] Han J, Bao H, Somers LMT. Experimental investigation of reactivity controlled compression ignition with n-butanol/n-heptane in a heavy-duty diesel engine. Appl Energy 2021;282. https://doi.org/10.1016/j.apenergy.2020.116164.
[37] IEA. Transport biofuels. Paris: IEA; 2021. Https://WwwIeaOrg/Reports/Transport-Biofuels.
[38] Abu-Qudais M, Haddad O, Qudaisat M. The effect of alcohol fumigation on diesel engine performance and emissions. 2000.
[39] Liu Y, Jiao W, Qi G. Preparation and properties of methanol-diesel oil emulsified fuel under high-gravity environment. Renew Energy 2011;36:1463–8. https://doi.org/10.1016/j.renene.2010.11.007.
[40] Pradelle F, Braga SL, de Aguiar Martins ARF, Turkovics F, Pradelle RNC. Stabilization of diesel–biodiesel–ethanol (DBE) blends: formulation of an additive from renewable sources. J Braz Soc Mech Sci Eng 2017;39:3277–93. https://doi.org/10.1007/s40430-017-0862-1.
[41] Jin C, Geng Z, Liu X, Ampah JD, Ji J, Wang G, et al. Effects of water content on the solubility between Isopropanol-Butanol-Ethanol (IBE) and diesel fuel under various ambient temperatures. Fuel 2021;286. https://doi.org/10.1016/j.fuel.2020.119492.
[42] Vargün M, Özsezen AN. Investigation of effect of dual fuel injection stages on combustion parameters in a diesel engine using ethanol-butan-2-ol-fossil diesel blends. Arabian J Sci Eng 2022. https://doi.org/10.1007/s13369-021-06441-6.
[43] Yilmaz N. Comparative analysis of biodiesel-ethanol-diesel and biodiesel-methanol-diesel blends in a diesel engine. Energy 2012;40:210–3. https://doi.org/10.1016/j.energy.2012.01.079.
[44] Jin C, Yao M, Liu H, Lee CFF, Ji J. Progress in the production and application of n-butanol as a biofuel. Renew Sustain Energy Rev 2011;15:4080–106. https://doi.org/10.1016/j.rser.2011.06.001.
[45] Heywood J. Internal combustion engine fundamentals. New York: McGraw-Hill; 1998.
[46] Pilusa TJ, Mollagee MM, Muzenda E. Reduction of vehicle exhaust emissions from diesel engines using the whale concept filter. Aerosol Air Qual Res 2012;12:994–1006. https://doi.org/10.4209/aaqr.2012.04.0100.
[47] Canakci M. Combustion characteristics of a turbocharged DI compression ignition engine fueled with petroleum diesel fuels and biodiesel. Bioresour Technol 2007;98:1167–75. https://doi.org/10.1016/j.biortech.2006.05.024.
[48] Sayin C, Ozsezen AN, Canakci M. The influence of operating parameters on the performance and emissions of a DI diesel engine using methanol-blended-diesel fuel. Fuel 2010;89:1407–14. https://doi.org/10.1016/j.fuel.2009.10.035.
[49] Pulkrabek WW. Engineering fundamentals of the internal combustion engine. J Eng Gas Turbines Power 2004;126:198. https://doi.org/10.1115/1.1669459. second ed.
[50] Ren Y, Huang Z, Miao H, Di Y, Jiang D, Zeng K, et al. Combustion and emissions

of a DI diesel engine fuelled with diesel-oxygenate blends. Fuel 2008;87: 2691−7. https://doi.org/10.1016/j.fuel.2008.02.017.
[51] Ning L, Duan Q, Chen Z, Kou H, Liu B, Yang B, et al. A comparative study on the combustion and emissions of a non-road common rail diesel engine fueled with primary alcohol fuels (methanol, ethanol, and n-butanol)/diesel dual fuel. Fuel 2020;266. https://doi.org/10.1016/j.fuel.2020.117034.
[52] Ozsezen AN, Canakcı M. Performance and combustion in a direct injection diesel engine fuelled with waste Palm and canola oil methyl esters, vol. 24; 2009.
[53] Shanmugam R, Murugesan P, Gedisa &, Guye G, Duraisamy B. ENVIRONMENTAL AND ENERGY MANAGEMENT Effect of additives on the stability of ethanol-diesel blends for IC engine application n.d. https://doi.org/10.1007/s11356-020-10934-6/Published.