International Journal of Environmental Science and Technology
https://doi.org/10.1007/s13762-024-05935-8

ORIGINAL PAPER

Experimental study on macro spray, combustion and emission characteristics of biodiesel and diethyl carbonate blends

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Received: 23 May 2023 / Revised: 20 July 2024 / Accepted: 6 August 2024

Abstract
Biodiesel has a higher viscosity and poorer atomization effect compared to traditional diesel fuel, resulting in lower combustion efficiency. This study investigated the effects of exhaust gas recirculation and a diethyl carbonate additive on the combustion and emission of biodiesel blends. Biodiesel was blended with DEC (90% volume biodiesel + 10% volume DEC: B90DEC10). Experiments were conducted at various brake mean effective pressures (BMEP: 0.2, 0.6, 1.0, 1.6, and 2.0 MPa) and EGR rates (0, 5, and 15%) on a direct injection and high pressure common rail diesel engine that met Euro VI emission standards. The results revealed that the viscosity and atomization effects of B90DEC10 were improved, and the premixed combustion period was longer and the heat release rate (HRR) was higher compared to traditional diesel. The maximum HRR of B90DEC10 was 20% higher than that of diesel at 0.6 MPa. Compared to diesel, the particulate matter emissions of B90DEC10 were reduced by 92%, and total hydrocarbons and carbon monoxide emissions were significantly reduced. However, there was an increase of 8–20% in nitrogen oxide emissions. At an EGR rate of 15%, the NO_x emissions of B90DEC10 were reduced by 45%, while having minimal impact on PM emissions and brake thermal efficiency. Therefore, it was concluded that B90DEC10 is a potential alternative to diesel, and the combustion performance and emissions control were optimized when using a higher EGR rate (15%).

Keywords Biodiesel · Diethyl carbonate · Spray · Combustion · Emission

Introduction

Biodiesel is a form of renewable energy, and its raw materials are mainly vegetable oils, animal fats, and certain fatty algae. Although there are numerous biodiesel production processes, transesterification is an easy, cheap, and widely used method to produce biodiesel. These attributes make it conducive to the wide application of biodiesel (Bhan et al. 2022). The use of biodiesel can reduce pollution emissions and relieve the pressure on fossil fuel supplies. Previous studies have confirmed that biodiesel has an R–(C=O)O–R' group, which can restrain the formation of soot precursors (Zhu et al. 2016; Chandrasekar et al. 2022). The main components of biodiesel are unsaturated fatty acids, which increase the cetane number, viscosity, and density compared to diesel. In the spray characteristics, compared to diesel, biodiesel has a longer spray tip penetration and higher injection velocity due to its higher density. Shao et al. (Shao et al. 2022) found that increasing the injection velocity of fuel can achieve a lower turbulent viscosity ratio and higher vortices, which can improve the combustion performance, achieve higher combustion efficiency, and reduce nitrogen oxide (NO_x) emissions. Due to its higher viscosity, biodiesel has a weaker atomization effect after injection, resulting in inadequate mixing of biodiesel and air and a reduction in combustion efficiency (Sharma et al. 2022; Xie et al. 2015).

Adding oxygen-containing compounds to biodiesel can reduce its viscosity. The most commonly used oxygenated compounds are alcohols, esters, and ethers. However, methanol and ethanol are short-chain alcohols and cannot be mixed

Editorial responsibility: S. Mirkia.

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Published online: 19 August 2024

directly with biodiesel. Chaitanya and Mohanty (2022) found that adding 30% pentanol to waste plastic oil (WPO) resulted in a 6.6% increase in brake thermal efficiency (BTE), a decrease of 0.06 kg/kWh in brake-specific fuel consumption (BSFC), a reduction of 127 ppm in NO_x emissions, a 7 ppm increase in total hydrocarbon (THC) emissions, and a 0.03% increase in carbon monoxide (CO) emissions compared to unmodified WPO. Although long-chain alcohols are soluble in biodiesel, their calorific value and oxygen content decrease, and their viscosity and surface tension increase when they introduced as additives(Kuszewski 2019; Karabektas and Yilancilar 2022). Ethers are also a common oxygen additive. Although the cetane value of ethers are large, ethers are volatile, explosive, and toxic, which results in safety risks during their transportation and use. Raju et al. (2020) found that adding 12% diethyl ether to mango seed methyl ester (MSME20) resulted in a significant improvement in brake thermal efficiency (BTE). Additionally, there was a notable decrease in CO, THC, NO_x, and particulate matter emissions, with reductions of 10.68, 33.33, 10.33, and 27.72%, respectively at maximum load. Carbonate esters have a high oxygen content and cetane number, as well as low viscosity, and excellent miscibility with biodiesel, making them a highly promising oxygen-containing additive. Aguado-Deblas et al. (2020) analyzed the combustion and emission of DEC mixed with vegetable oil. The study found that a combination of DEC and vegetable oil could completely replace diesel. Although the heating value and output power were lower than those of diesel, the particulate matter (PM) was only 5–8% of diesel, producing almost zero emissions. Hellier et al. (2013) combined respectively a mixture of dibutyl carbonate (DnBC), diethyl carbonate (DEC), and dimethyl carbonate (DMC) with *n*-malethane at a ratio of 30%, and found that these blends reduced soot and CO emissions when compared to diesel. Bridjesh and Kannaiyan Geetha (2020) mixed DEC with WPO-diesel blends and found that the additives not only optimized the combustion process but also reduced particulate matter emissions. Additionally, Wu et al. (2024) found that lower CO and NO_x emissions could be obtained by mixing diesel with waste tire pyrolysis oil at volume ratios of 85 and 15%. It was also found that supplying hydrogen through the intake manifold during combustion could increase the calorific value of the mixture, further improving torque and power, and reducing fuel consumption, therefore providing a promising alternative to diesel.

Previous studies have shown that exhaust gas recirculation (EGR), as an important means of engine combustion process control and NO_x emissions reduction, can substantially reduce combustion temperature and inhibit the formation of NO_x emissions (Zhang and Balasubramanian 2015; Zhou et al. 2020). However, as the EGR rate increases, particularly when it surpasses 40%, there is a significant increase in the emissions of CO, THC, and soot mentioned previously, and the optimal EGR rate is generally considered to be between 5 and 15% (Zhang et al. 2021a; Huang et al. 2022). When the EGR rate is less than 5%, there is less impact on the engine performance (Rami Reddy et al. 2021; Sun et al. 2020). Agnese et al. (Magno et al. 2016) found that a slight increase in EGR could reduce NO_x, while simultaneously not increasing PM emissions from biodiesel.

Diethyl carbonate is an oxygenated additive that is non-toxic, has excellent miscibility with biodiesel, and has a high cetane number. In the present study, we examined the spray characteristics of biodiesel, diesel and biodiesel blends mixed with DEC at a 10% volume ratio. Additionally, we investigated the combustion and emission characteristics on a direct injection engine using various EGR rates. The results of this study hold significant value for the engineering applications of biodiesel as a potential alternative to diesel fuel, as well as for DEC as a suitable oxygen-containing additive.

Materials and methods

Test fuels

The fuels used in the experiment were diesel (D100), biodiesel (B100), and biodiesel-DEC blends. Biodiesel was obtained from soybean oil through a transesterification process. The biodiesel-DEC blends had a volume fraction ratio of 90% biodiesel blended with 10% DEC (B90DEC10).

The thermophysical properties of fuels are displayed in Table 1.

Experimental setup

The engine used in this study was a 4-cylinder, high pressure common rail and direct injection (CRDi) diesel that met the Euro VI emission standards. It had a displacement of 2.0 L. The

Table 1 The thermophysical properties of the test fuels (25 ℃)

Fuel	Properties			
	D100	DEC	B100	B90DEC10
Density (g/cm^3)	0.83	0.938	0.848	0.857
Viscosity (mPa.s)	3.11	0.78	5.90	4.62
Surface tension (mN/m)	28.3	25.6	30.5	29.4
Fire point (°C)	71	46	83	72
Heating value (MJ/kg)	42.5	21.1	37.5	35.9
Cetane value	55	58	52	52.6
Latent heat of vaporization (kJ/kg)	260	360	300	304
Oxygen content (%)	0	40.7	10	13
Flash point (°C)	55	25	175	58

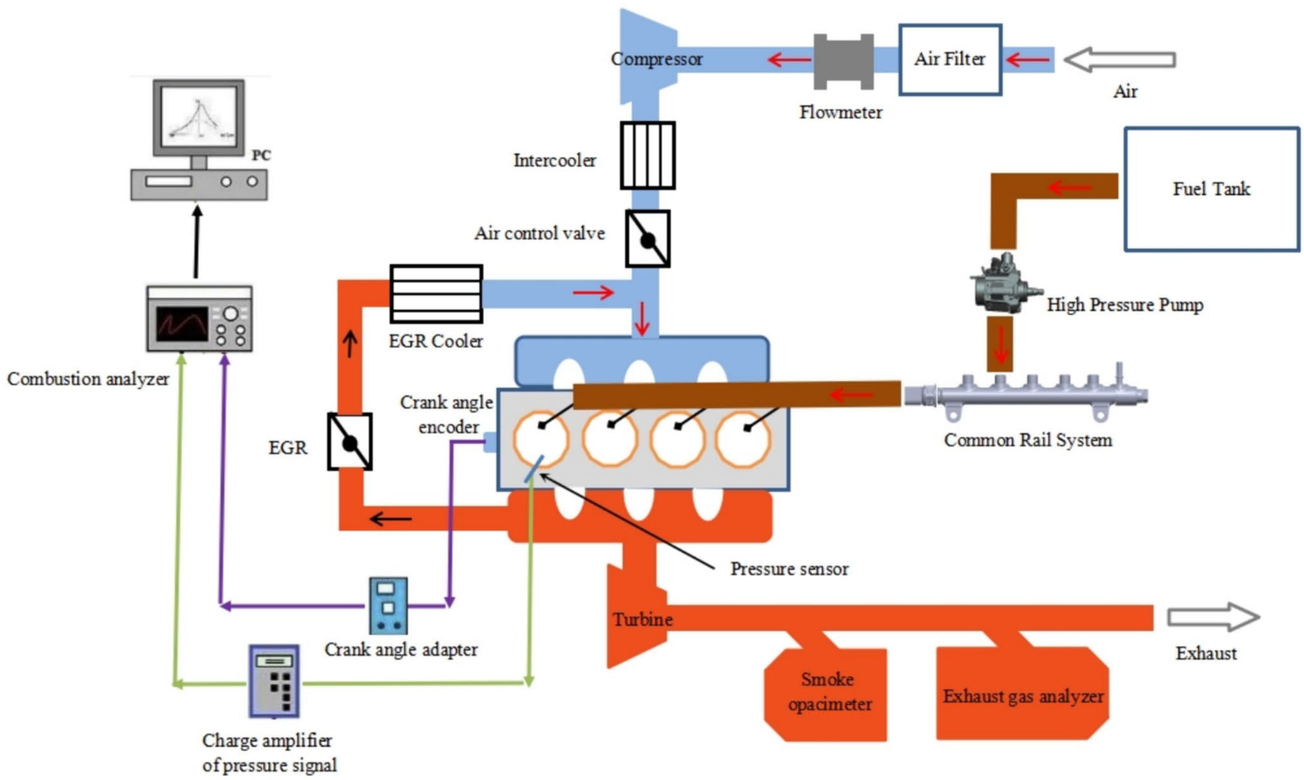

Fig. 1 Experimental schematic diagram

Fig. 2 Actual experimental setup diagram

engine had a rated output power of 95 kW and rated speed of 3200 r min^{-1}. The intake was turbocharged and intercooled, and the engine exhaust was circulated into the cylinder through the EGR valve and cooler. Table 2 displays the fundamental parameters of the diesel engine. In Fig. 1, a schematic diagram of the experimental setup is depicted. Additionally, Fig. 2 illustrates a diagram of the actual experimental setup.

During the test, a Schneider water-cooled electric dynamometer was utilized to measure the output torque and speed of the diesel engine, and an AVL puma-open engine measurement and control system was used to regulate the testing procedure. A fuel consumption analyzer (AVL 7351 CME) was used to monitor the fuel consumption during the test. A cylinder pressure sensor (KISTLER 6056A) was installed in the cylinder near the center of the combustion chamber to monitor the combustion pressure in the cylinder. The cylinder pressure sensor (KISTLER 6056A) was used to collect the cylinder pressure charge signal. These signal was firstly amplified by the charge amplifier in the combustion analyzer and then passed through the data acquisition card, data interface module and data conversion module to generate the cylinder pressure data. A Horiba MEXA-7100D tail gas measurement system was used to measure CO, THC, NO_x, and other conventional gas emissions in tail gas emissions, and an opaque opacimeter was used to analyze PM in the tail gas emissions. The cylinder pressure was determined by an AVL combustion analyzer at an interval of 1°CA. The DEWESoft analysis software of the combustion analyzer could calculate the instantaneous heat release rate and CA50 according to the collected cylinder pressure data. For all equipment, the specific models and their accuracy are shown in Table 3.

Table 2 The fundamental parameters of the test diesel engine

Air intake type	Turbocharger with WGT
Injection type	CRDi
Cooling type	Water
Number of cylinders	4
Stroke	4
Displacement (L)	2.0
Bore × stroke (mm × mm)	86 × 86
Compression	16.5:1
Rated power (kW)	95
Rated speed (r.min^{-1})	3200
Max. torque (Nm)/speed (r min^{-1})	320/(1400 – 2400)
Injector hole diameter (mm)	0.15

Table 3 Type, production, and accuracy of measured parameters

Parameter	Type	Manufacturer	Accuracy
Electric dynamometer	129,106	Schneider	± 0.1 Nm
Fuel consumption instrument	7351 CME	AVL	± 0.01 kg/h
Combustion analyzer	Indicom V2.5	AVL	/
Opacimeter	483	AVL	± 0.001 mg/m^3
Emission Test Systems	MEXA-7100D	Horiba	± 1 ppm
Angle encoder	TRD-GK3600-RZ	Koyo	500 P/R
Crankshaft Position Sensor	0281002315	Bosch	± 0.1°CA
Exhaust gas temperature sensor	DARTS200	Delphi	± 1 K
Cylinder pressure transducer	6056A	KISTLER	± 0.1 MPa

Table 4 Test conditions of the engine

Speed (r min^{-1})	1600
BMEP (MPa)	0.2/0.6/1.0/1.6/2.0
Injection pressure (MPa)	120
EGR (%)	0/5/15
Main injection timing (°CA BTDC)	4
Pilot injection timing (°CA BTDC)	17

Experimental method

The engine's test conditions are displayed in Table 4. During the test, the engine speed was set at 1600 r/min, and the engine loads were varied at 10, 30, 50, 80, and 100%. The converted to the brake mean effective pressures (BMEPs) for each load were recorded as 0.2, 0.6, 1.0, 1.6, and 2.0 MPa. The test was conducted without making any modifications to the diesel engine's structure. The first test was conducted using diesel (D100) to determine the reference value. Before each fuel was replaced, we emptied the fuel remaining in the fuel tank and fuel consumption instrument, and cleaned the fuel pipe by running stably for 0.5 h under idling conditions.

After passing through the EGR valve and cooler, the combustion exhaust entered the intake manifold and mixed with fresh air before entering the cylinder. The amount of exhaust gas could be regulated by adjusting the EGR valve opening as required. The EGR rates were determined by calculating the mass of recirculated exhaust gases entering the cylinder (m_{EGR}) in relation to the total intake mass entering the cylinder (m_{intake}):

$$EGR = \frac{m_{EGR}}{m_{\text{int}\,ake}} \times 100\% \tag{1}$$

The instantaneous heat release rate (HRR) was calculated by integrating the combustion pressure and volume in the cylinder with the crank angle. The formula for this calculation was as follows:

$$\frac{dQ}{d\theta} = \frac{\gamma - 1}{\gamma} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \tag{2}$$

where θ is the crank angle; V and P represent the instantaneous working volume and in-cylinder combustion pressure, respectively; γ is a constant that represents the adiabatic index of mixture gases (fixed at 1.35).

Results and discussion

Macro spray characteristics

In order to compare and analyze the breakup and atomization of the fuels, their spray characteristics were tested in

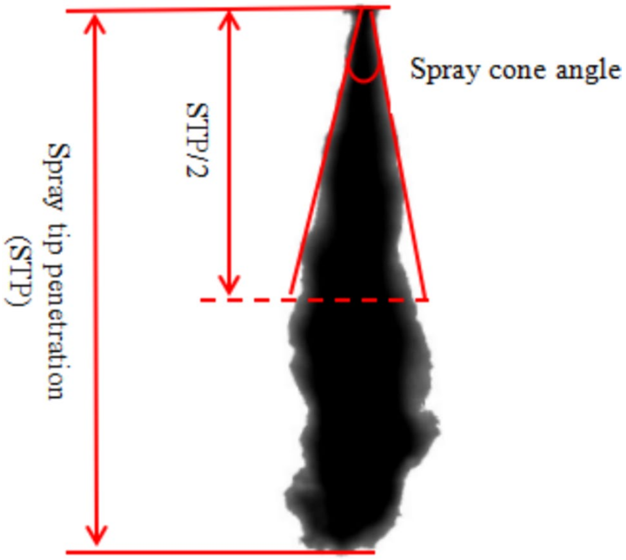

Fig. 3 Definition of the spray characteristic parameters

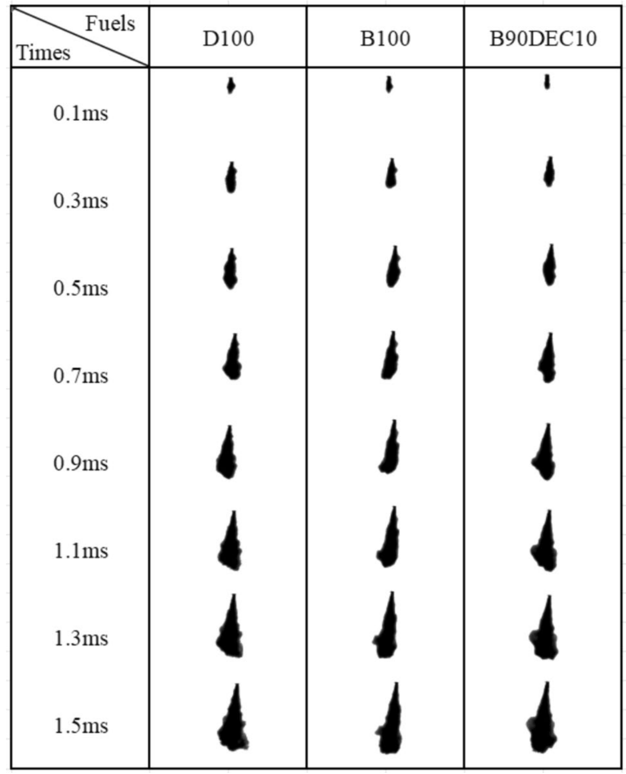

Fig. 4 Spray morphology of the test fuels

a constant volume device using the Schlieren method (Li et al. 2018). The fuel injection system was a high-pressure common rail system. The injection pressure for the test was 120 MPa, the ambient pressure was 2 MPa, and the ambient temperature was room temperature. Figure 3 shows the definition of the macro spray characteristics parameters. The spray tip penetration was the vertical distance from the bottom of the spray to the nozzle (Zhao et al. 2023). A triangle was formed between the bottom of the nozzle and the spray width at half of the spray tip penetration, the two sides of the triangle were tangent to the outer edge of the spray. The angle formed by the two sides was referred to as the spray cone angle, and it was determined using a trigonometric function.

Figure 4 displays the spray morphology resulting from the fuel spray. The spray morphology of diesel was a symmetrical triangle. The spray morphology of biodiesel was elongated, which was directly related to the high density. This was because a high density can increase the initial kinetic energy after injection and increase the spray tip penetration (Fu et al. 2019). When biodiesel was mixed with DEC in a 10% volume ratio, the tail morphology of B30DEC10 showed more significant perturbation and diffusion than the unmodified biodiesel. This was because DEC reduces the viscosity of blended fuel and intensifies the fragmentation of blended fuel droplets under the effect of air entrainment.

Figure 5 displays the findings of quantitative analysis on the spray characteristics of various fuels. It is evident that biodiesel exhibits a greater spray tip penetration. Specifically, the spray tip penetration of biodiesel was 3 mm greater than that of diesel at 1.4 ms. This was because the densities of biodiesel were greater than that of diesel, resulting in an increased the momentum of the fuel. The spray tip penetration of B90DEC10 was slightly lower than that of biodiesel, but it was still greater than that of diesel. This is because although B90DEC10 has a highest density, its surface tension and viscosity are lower than those of biodiesel, which promotes B90DEC10 to be broken up more easily and the momentum leaving the nozzle is decreased (Zhang et al. 2021b).

Biodiesel exhibited the smallest spray cone angle, while B90DEC10 had a larger spray cone angle. This was due to the fact that when DEC was mixed with biodiesel, the surface tension and viscosity of the blended fuel were decreased, leading to improved droplet breaking ability and atomization quality. The atomization of biodiesel was poor, resulting in a lower projection area compared to diesel. Due to its low viscosity and high density, B90DEC10 exhibited increased spray tip penetration and spray cone angle compared to biodiesel. Additionally, the spray projection area was 2% larger than that of biodiesel. When analyzing Schlieren images, the threshold of the gray value is defined as 30, and the image region with a gray value < 30 is regarded as the liquid core area (Yi et al. 2018). The percentage of the liquid core area to the spray area was defined as the spray concentration (Zhao et al. 2023). In Fig. 5, Biodiesel had the lowest spray concentration, which indicated that the diffusion

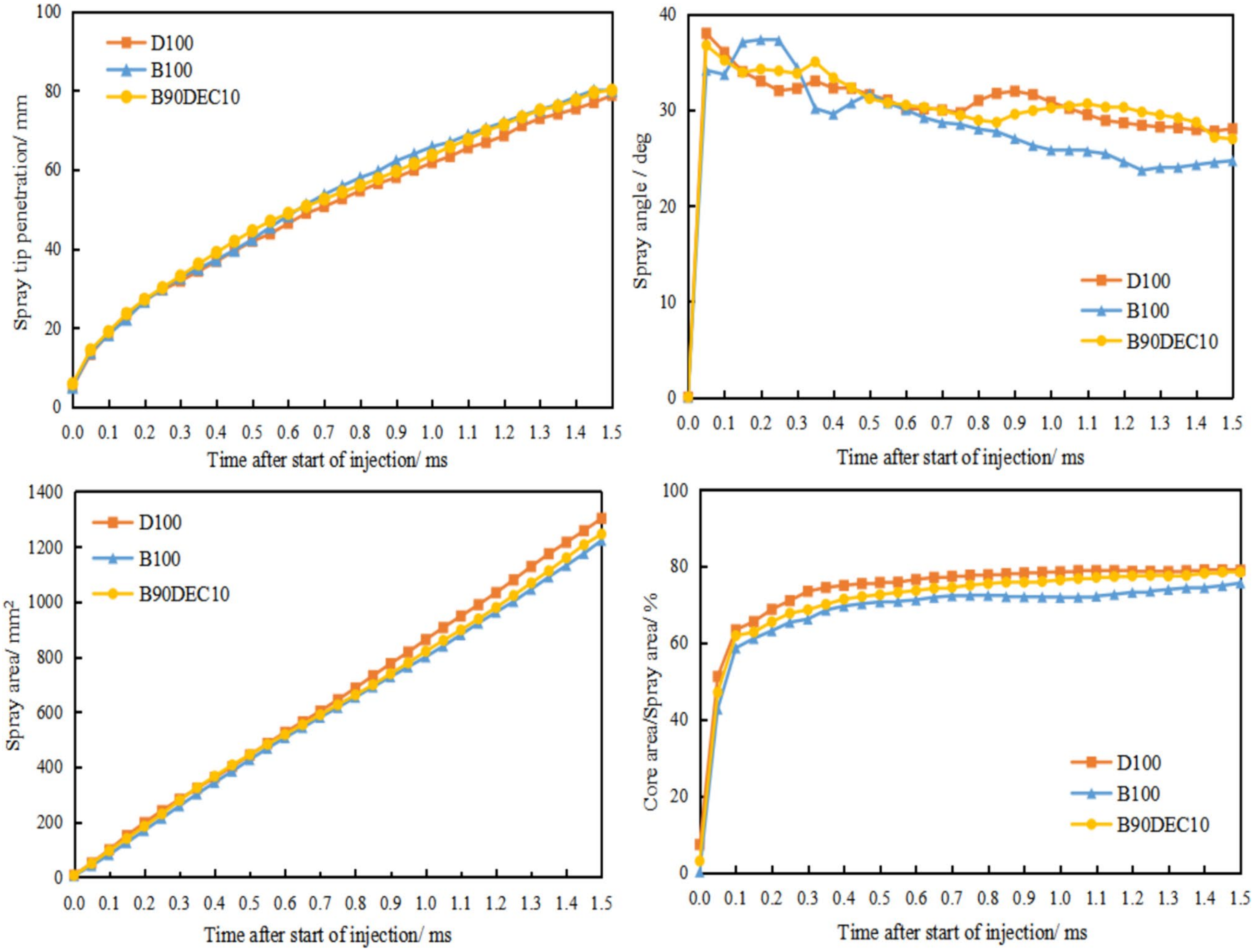

Fig. 5 Quantitative analysis on the macroscopic spray characteristics of various fuels

ability of biodiesel after injection was weaker than for the other fuels. However, the average spray concentration of B90DEC10 was 4.38% higher than that of biodiesel. This indicated that adding a 10% volume ratio of DEC to biodiesel could enhance atomization quality and improve combustion efficiency.

Combustion characteristics

Figure 6 illustrates the comparison of in-cylinder combustion pressure and heat release rate (HRR) at various loads. Due to the pilot injection, a small wave peak occurred before peak of the instantaneous heat release rate. The start of combustion (SOC) was latest for B90DEC10, while diesel was the earliest. This was because B90DEC10 had the lowest cetane number and highest latent heat of vaporization (Liang et al. 2021; Qi et al. 2021). The maximum HRR of B90DEC10 was 20% higher than that of diesel at 0.6 MPa. It is due to that the air/fuel ratio in the cylinder was relatively smaller at low loads, and B90DEC10 can provide more oxygen atoms for combustion, which was beneficial for promoting the combustion (Natesan et al. Mar 2019). Under medium to high loads, the intake air volume in the cylinder increased, so the maximum HRR of B90DEC10 was similar to that of diesel after 1.0 MPa. Biodiesel and B90DEC10 had the highest fuel injection quantity and also had inherent oxygen rich properties, leading to an increase in the premixed combustion period (Alruqi et al. 2023). B90DEC10 released more heat during premixed combustion due to its longer premixed combustion period. The peak in-cylinder combustion pressure is primarily determined by the rate of combustion during the premixed combustion period (Raju et al. 2020). The low viscosity of diesel engines improved the uniformity of the diesel and air mixture, resulting in a faster combustion rate and higher peak cylinder pressure during the initial stage of combustion. Moreover, diesel had the largest heating value, resulting in the largest peak in-cylinder combustion pressure. Although the peak in-cylinder

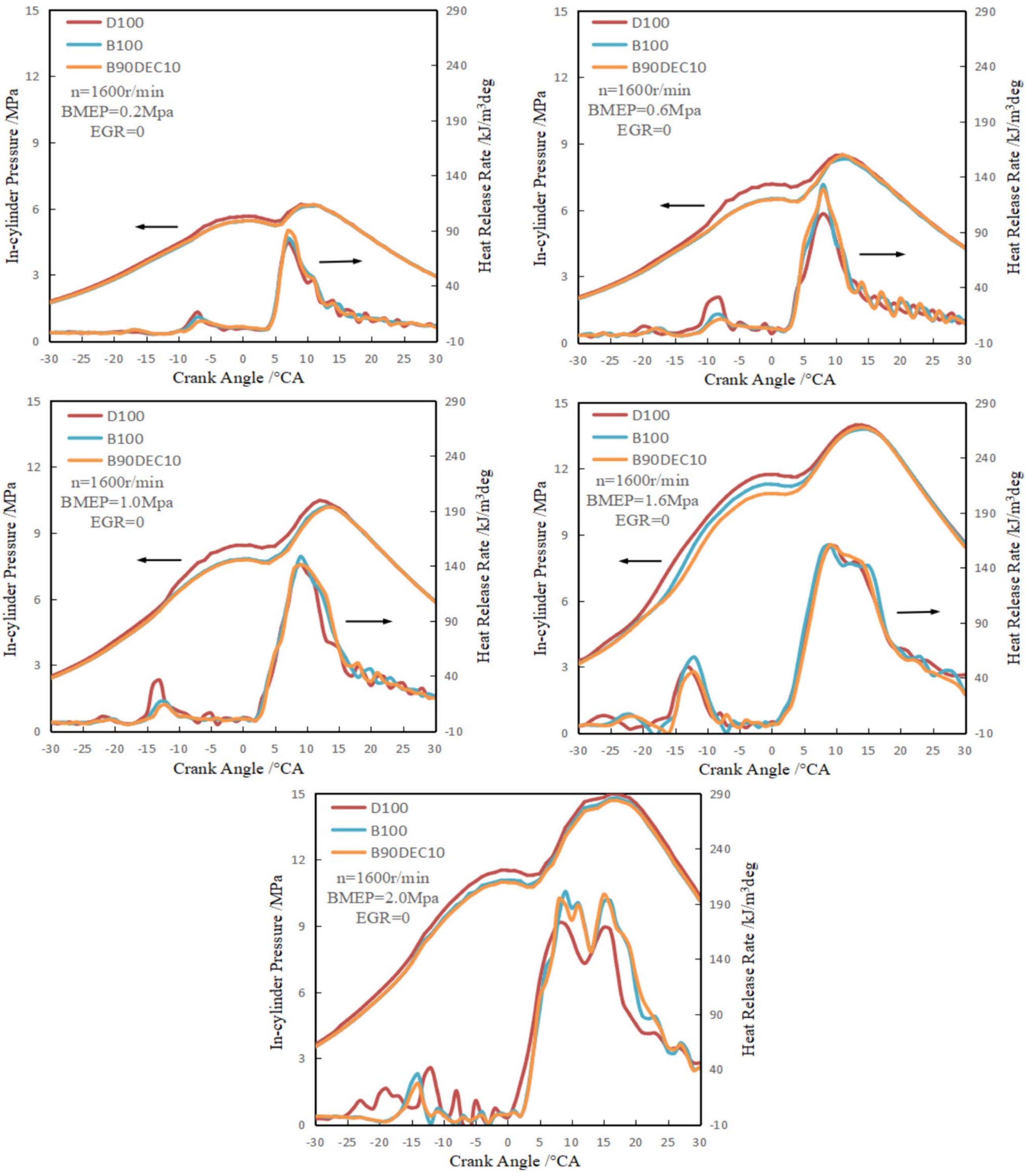

Fig. 6 Comparison of in-cylinder combustion pressure and the HRR at various loads

combustion pressure of B90DEC10 was 0.2–3% lower than that of diesel, it was 0.3–2% higher than that of biodiesel. Obviously, the lower viscosity of B90DEC10 promoted the atomization of fuel and improved the combustion efficiency.

Figure 7 illustrates the comparison of in-cylinder combustion pressure and heat release rate at various EGR rates. At an EGR rate of 5%, there were no significant decrease in the maximum heat release rate (HRR) and the peak in-cylinder combustion pressure for both biodiesel and

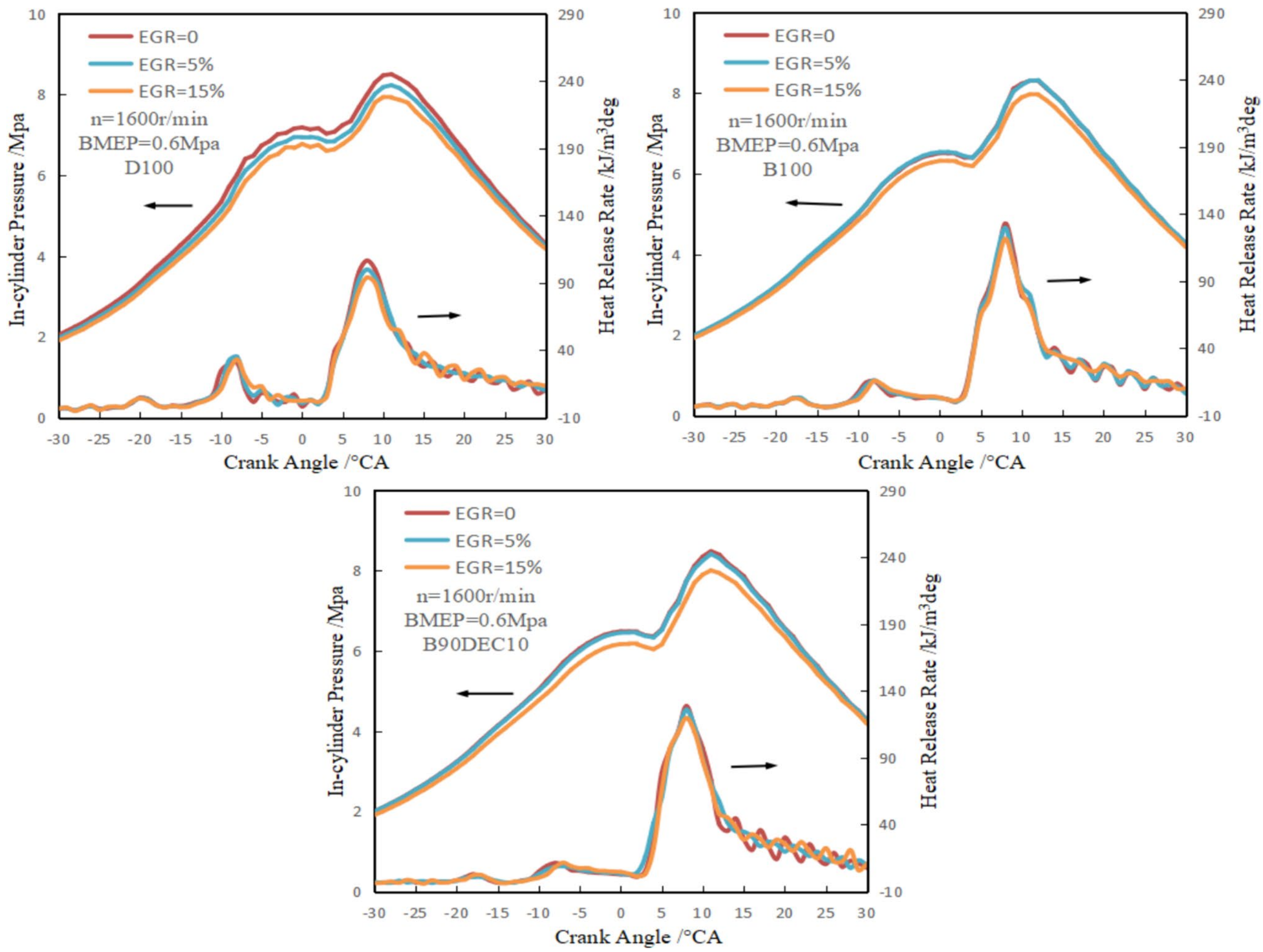

Fig. 7 Comparison of in-cylinder combustion pressure and the HRR at various EGR rates

B90DEC10. However, the maximum HRR and the peak in-cylinder combustion pressure of diesel fuel decreased by 3 and 6%, respectively. This was due to the different thermophysical properties of the fuels. Firstly, the oxygen content of biodiesel and B90DEC10 could compensate for the insufficient concentration of oxygen in the combustion chamber when using 5% EGR rate. In addition, their low cetane number and poor flammability also prolonged the ignition delay, resulting in a more uniform mixture of fuel/air and more conducive to promoting combustion. However, diesel does not contain oxygen, and EGR has a more obvious inhibitory effect on diesel combustion, which reduced the premixed combustion period and the HRR. When the EGR rate reached 15%, the peak in-cylinder combustion pressure of biodiesel and B90DEC10 decreased by 4 and 6%, respectively; and the maximum HRR decreased by 9 and 7%, respectively (compared with EGR0). In contrast, the maximum HRR and the peak in-cylinder combustion pressure of diesel decreased by 14 and 7% respectively, which was a higher reduction rate than that of B90DEC10. This was because the amount of exhaust gas entering into the combustion chamber increased as the EGR rate increased, which significantly hindered combustion. However, oxygen-containing fuels can alleviate the hindrance of the increasing EGR rate.

Figure 8 shows the variation of the CA50 with various EGR rates and loads. The CA50 was defined as the time at which 50% of the total heat release during combustion occurred. This metric is commonly used to analyze the center of gravity of the combustion process. Due to its high viscosity and low evaporability, biodiesel had a slower combustion rate, resulting in a delayed CA50. The CA50 of B90DEC10 was 0.3–0.7°CA earlier than that of biodiesel. This was due to the fact that B90DEC10 possesses the qualities of lower viscosity and higher oxygen content, which are advantageous in promoting combustion and accelerating the combustion rate. As a result, it leads to an earlier timing of CA50. As the EGR rate increased, the combustion rate decreased and the CA50 was delayed. For instance, at an EGR rate of 5%, the CA50 of B90DEC10 was delayed

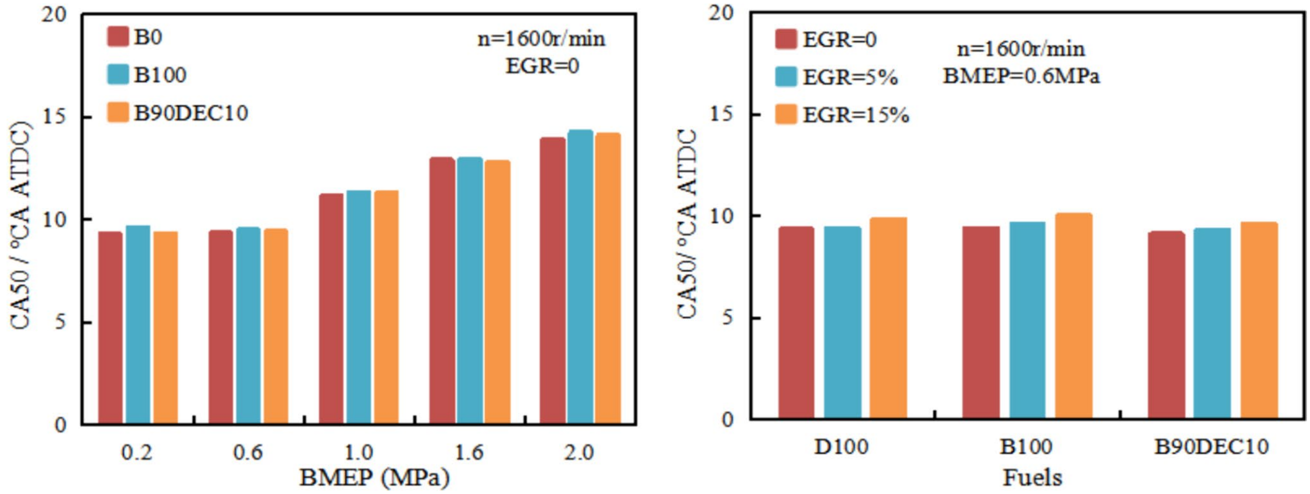

Fig. 8 Variation of the CA50 with various loads and EGR rates

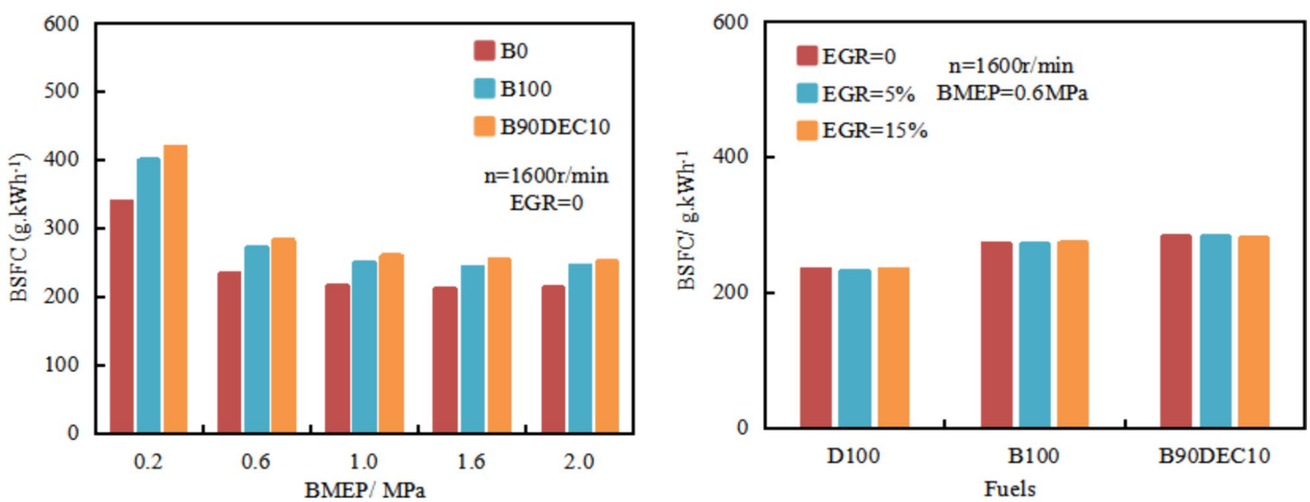

Fig. 9 Comparison of BSFC at various loads and EGR rates

by 0.2°CA; However, at an EGR rate of 15%, the CA50 of B90DEC10 was delayed by 0.5 CA.

Figure 9 displays a comparison of brake-specific fuel consumption (BSFC) at various EGR rates and loads. The BSFC of B90DEC10 and biodiesel were 15 and 20% higher than that of diesel, respectively. This was because diesel had the highest heating value, followed by B100, and DEC had the lowest heating value. As the EGR rate increased, the BSFC of fuels increased slightly. Figure 10 displays a comparison of BTE at various load and EGR rates. Diesel had the highest BTE, and the BTE of B90DEC10 was 1.1–2.8% higher than that of biodiesel. Compared to biodiesel, B90DEC10 has a lower viscosity and improves the atomization effect. This difference promotes the full mixing of B90DEC10 with air, resulting in improved combustion. At the same time, the addition of DEC to biodiesel increased the oxygen content of mixture and promoted full combustion. Moreover, the CA50 of B90DEC10 was advanced and became closer to the top dead center, making expansion work more effective.

The exhaust gas temperature before the turbine (EGT) is crucial indicator for analyzing the combustion performance of a diesel engine. According to Fig. 11, the exhaust gas temperature before the turbine follows the order of diesel, biodiesel and B90DEC10. This was because diesel had the highest heating value and a good uniformity in its mixing with air, resulting in sufficient combustion. In contrast, biodiesel had a high viscosity and poor atomization effect, resulting in insufficient combustion. Due to B90DEC10 having the lowest heating value, its exhaust gas temperature before the turbine was also the lowest.

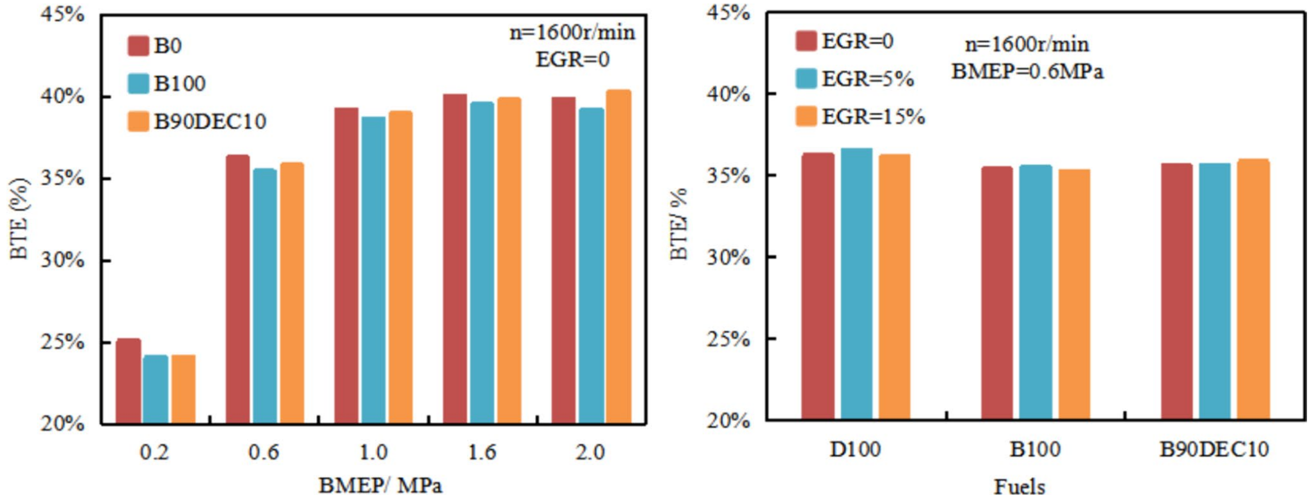

Fig. 10 Comparison of BTE at various loads and EGR rates

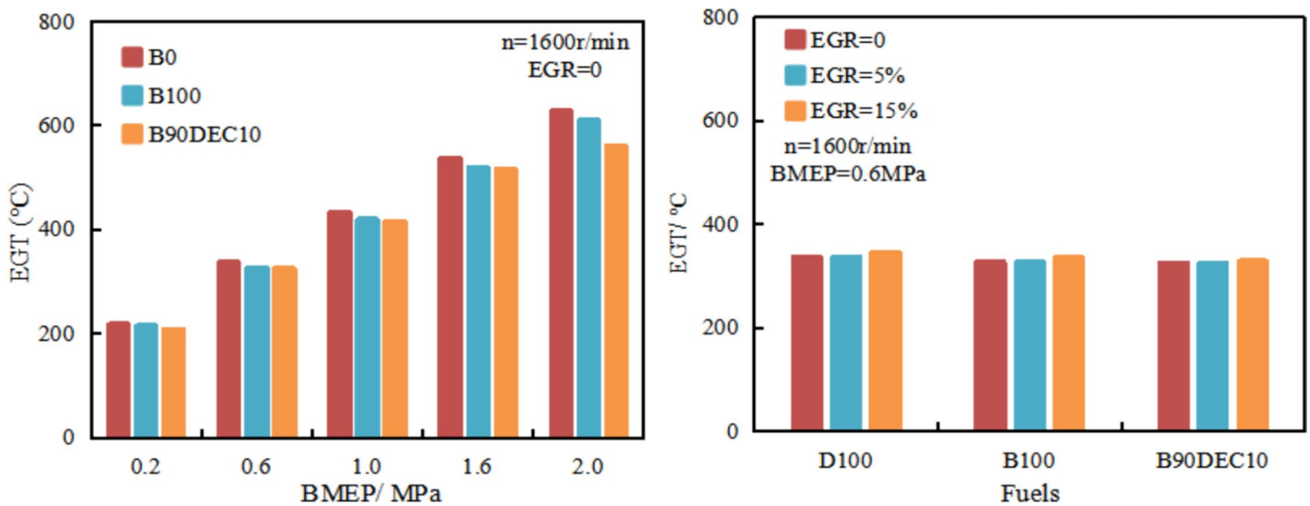

Fig. 11 variation of EGT before the turbine with various loads and EGR rates

Table 5 Regulatory limits of Euro VI for heavy-duty diesel engine

WHSC	PM	NO_x	THC	CO
Unit	g/kWh	g/kWh	g/kWh	g/kWh
Regulatory limits	0.01	0.4	0.13	1.5

Emission characteristics

The PM, NO_x, THC, and CO emissions were measured in real-time during the combustion process. The emission variation patterns of diesel, biodiesel, and biodiesel/DEC were studied. Table 5 shows the regulatory limits of Euro VI for heavy-duty diesel engine according to the World Harmonized Stationary Cycle (WHSC). In this paper, the measured emission results are compared with the regulatory limits of Euro VI.

Figure 12 displays a comparison of PM emissions at various loads and EGR rates. Biodiesel and B90DEC10 had lower PM emissions than diesel. B90DEC10 had the lowest PM emissions, with emissions being 92% lower compared to diesel. This was because both DEC and biodiesel were oxygenated fuels, which promoted the oxidation of PM. The mixture also contained low levels of aromatic hydrocarbons, which were beneficial to reduce the formation of PM precursors (Serhan et al. 2018). Compared to diesel, the PM emissions of biodiesel were approximately 80% lower at 30% load, and B90DEC10 generally had the lowest PM emissions. When the EGR rate reached 15%, there was a notable increase in PM emissions from both diesel and biodiesel.

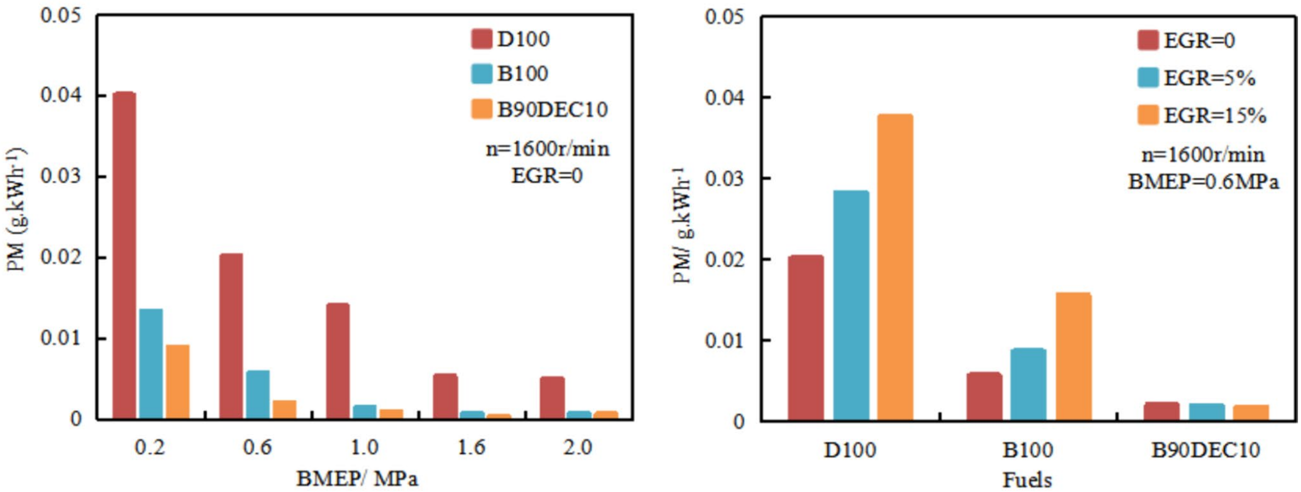

Fig. 12 Comparison of the PM emissions at various loads and EGR rates

However, there was no notable increase in PM emissions from B90DEC10, as its oxygen content had reached 13%. Therefore, when B90DEC10 was used instead of diesel, PM emissions were far below the regulatory limit of the Euro VI standard.

Figure 13 displays a comparison of NO_x emissions at various fuels and EGR rates. High temperature and oxygen-rich environment within the cylinder are the primary factors that contribute to the formation of NO_x (Chen et al. 2019). Due to its higher oxygen content, biodiesel emitted 1.1–1.2 times more NO_x emission than diesel (Ayhan et al. 2020). The NO_x emissions of B90DEC10 were 8–20% larger than those of diesel. When the BMEP was lower than 1.6 MPa, the NO_x emissions of B90DEC10 were comparable to those of biodiesel, with no significant difference observed. This was because DEC has a high latent heat of vaporization, and its evaporation needs to absorb large amounts of heat during combustion, which in turn lowers the temperature in combustion chamber. This decrease in temperature helps to suppress the formation of NO_x emissions. When the BMEP was 2.0 MPa, there was a noticeable rise in NO_x emissions from B90DEC10. This is due to that higher combustion temperatures were more likely to promote the generation of NO_x emissions from oxygenated fuels under high load. We are aware that that NO_x emissions can be effectively reduced through EGR. For instance, When the EGR rate was increased to 5%, there was a reduction of 12–22% in NO_x emissions. When the EGR rate was further increased to 15%, there was a significant reduction of 45% in NO_x emissions. Additionally, when the EGR rate reached 15%, both B100

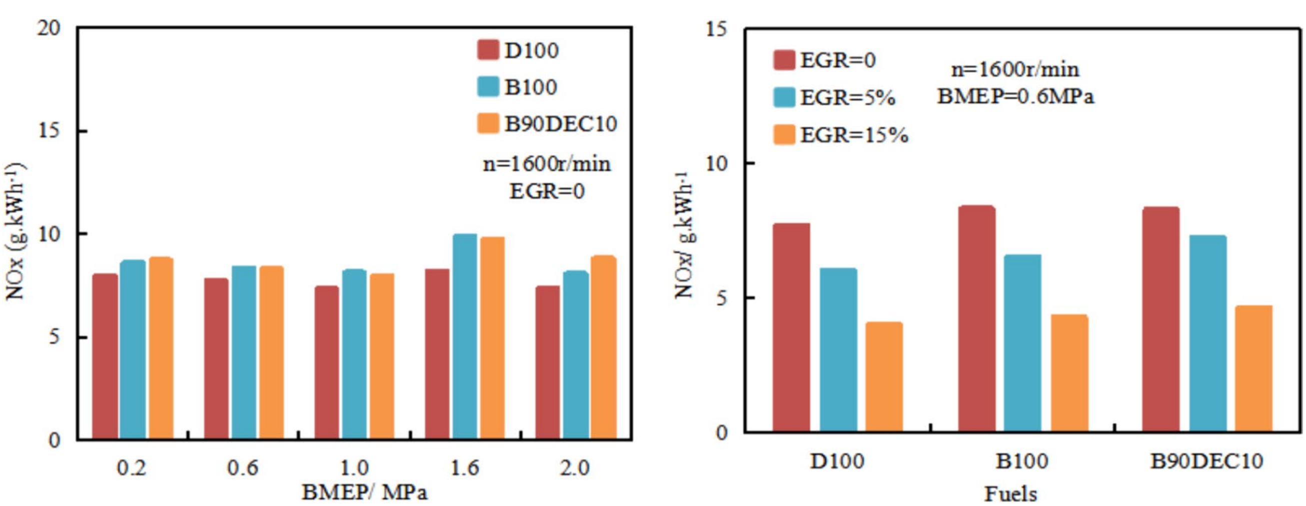

Fig. 13 Comparison of the NO_x emissions at various loads and EGR rates

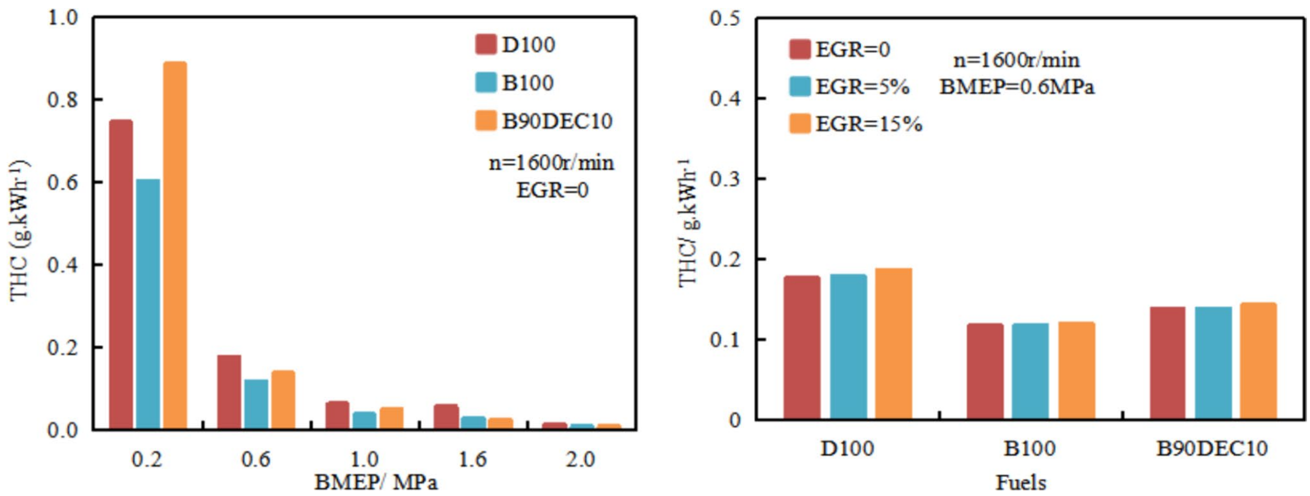

Fig. 14 Comparison of THC emissions at various loads and EGR rates

and B90DEC10 showed a noticeable increase in PM emissions, which breaked the trade-off balance between PM and NO_x emissions (Öztürk and Can 2022).

Figure 14 displays a comparison of THC emissions at various loads and EGR rates. Hydrocarbons are mainly produced in the wall quenching area, or in areas where the mixture is too lean or too rich (Venu et al. 2019). The mixture being too rich can lead to a high temperature in combustion chamber, resulting in an increase in THC emissions. On the other hand, if the mixture is too lean, it can lead to a low temperature in combustion chamber, resulting in higher THC emissions. The THC emissions were lowest for biodiesel because the higher oxygen content of biodiesel was conducive to their oxidation, and biodiesel contained almost no aromatic hydrocarbons. Additionally, the cracking of hydrocarbons was reduced in the biodiesel. Biodiesel had been found to produce THC emissions that were 20–30% lower than those of diesel at low loads. Additionally, the THC emissions of biodiesel fell below the regulatory limit. However, although the oxygen content was highest in B90DEC10, the THC emissions of B90DEC10 were larger than those of biodiesel at medium and low loads. Compared to biodiesel, the THC emissions increased by 20–45%. This was because the latent heat of vaporization of B90DEC10 was larger than for the other fuels, which significantly reduced the combustion temperature and led to an increase in the size of the quenching area in the cylinder (Babu and Anand 2017). Additionally, the density of B30DEC10 was larger, and the combustion flame may strike the wall. The wall surface can rapidly cool the combustion flame in a process called cold shock or quenching, which will generate active free radicals in the combustion flame slow or even halt the chemical reaction, resulting in increased THC emissions. However, the THC emissions of B90DEC10 were lower than those of biodiesel at high loads. This was due to the high oxygen content of B90DEC10 promoting the oxidation of THC. Additionally, the thermal conditions in the cylinder were effectively improved and the combustion temperature was higher with the increasing load, which was conducive to the oxidation of THC (Raju et al. 2020). When the EGR rates were below 15%, low EGR rates had little effect on THC emissions.

Figure 15 displays a comparison of CO emissions at various loads and EGR rates. The main reasons for the increase in CO emissions were the lower temperature or insufficient concentration of oxygen during the combustion reaction of the fuel (Esakki et al. 2022; Jayabal et al. 2020). The combustion temperature and air intake in the engine cylinder both increased as the load increased, which promoted the oxidation of CO and decreased CO emissions. Diesel had the lowest CO emissions at low loads (BMEP = 0.2 MPa), and biodiesel had 45% higher CO emissions than diesel. When DEC was added to biodiesel, the CO emissions were higher than those of biodiesel. This was because CO emissions were related to the oxygen concentration during the combustion reaction, and an excessive oxygen concentration led to an increase in CO emissions. After adding DEC to biodiesel, the oxygen content of the fuel increased to 13%, resulting in an excessive air/fuel ratio and increasing the CO emissions (Chaitanya and Mohanty 2022). The latent heat of vaporization is another important influencing factor. The high latent heat of evaporation of B90DEC10 means that it needs to absorb more heat compared to other fuels during the evaporation process. This results in a lower temperature of combustion in the cylinder, leading to an increase in CO emissions. When the BMEP was greater than 0.6 MPa, diesel produced the highest CO emissions. This was because the pressure and temperature in the combustion chamber

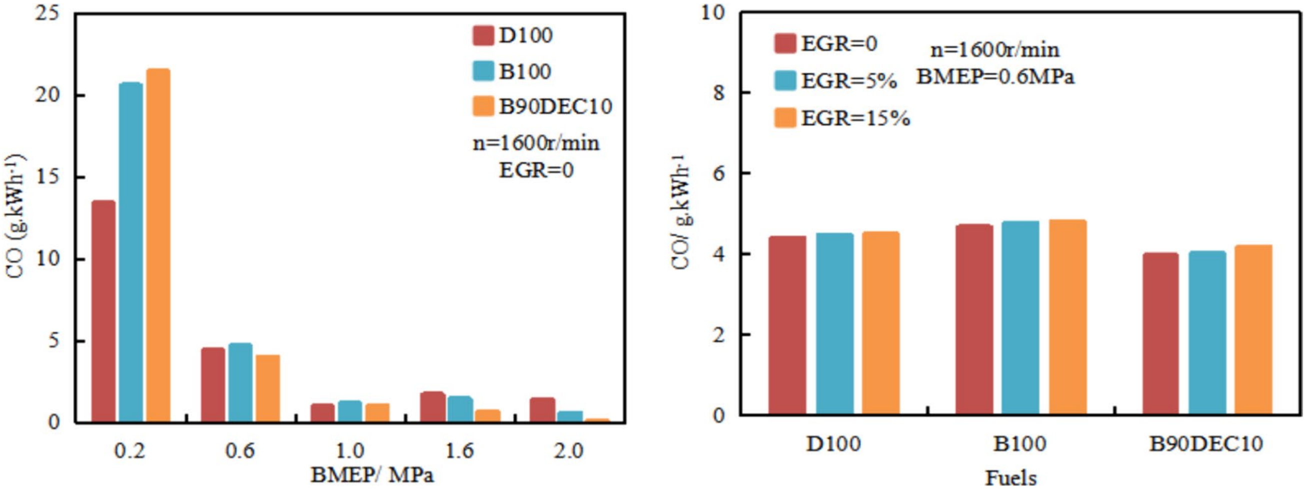

Fig. 15 Comparison of CO emissions at various loads and EGR rates

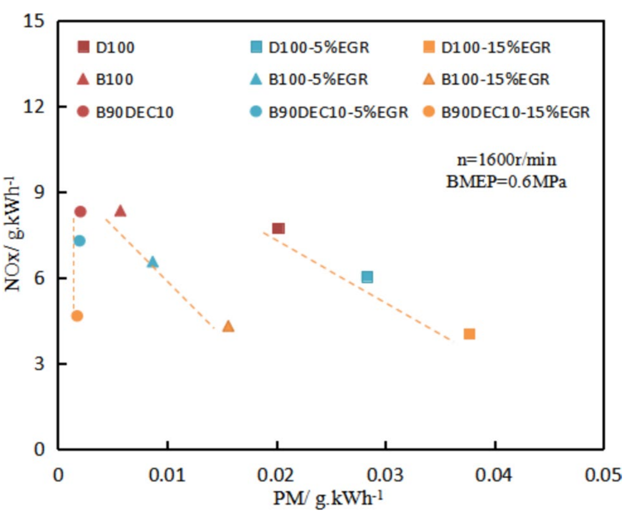

Fig. 16 Trade-off relationship between NO_x and PM emissions

were higher, and the oxygen concentration in the combustion chamber was the important factor affecting CO emissions. The CO emissions increased with the increase of EGR rate. When the EGR rate was 15%, CO emissions increased by 12% on average at low loads. This was because EGR reduced the temperature and oxygen concentration in combustion chamber and inhibited the oxidation of CO. The effects of EGR on NO_x emission reductions were more effective and significant than the increase in THC and CO emissions.

Figure 16 illustrates the trade-off relationship between NO_x and PM emissions. The trade-off curve applies to conventional diesel engines. Particulate matter is easily oxidized in the oxygen-rich and high temperature environment of the cylinder, but these are also favorable conditions for the generation of NO_x (Wu et al. 2022). Generally, the concentration of oxygen and the temperature in combustion chamber decreases with the increase of EGR rate. This effectively inhibits the generation of NO_x emissions, but it also leads to an increase in PM emissions. In this study, the use of B90DEC10 could disrupt this PM-NO_x trade-off. As the EGR rate increased, there was a little increased in PM emissions. Exhaust gas recirculation had a significant impact on reducing NO_x emissions, and NO_x emissions decrease linearly with the increase of EGR rate. According to the Zelovich mechanism, the main chemical reaction for generating NO_x is: $O_2 + N \rightarrow NO + O$, $O + N_2 \rightarrow NO + N$, while the chemical reaction at high temperatures is: $N_2 + O_2 \rightarrow 2NO$, $NO + O_2 \rightarrow NO_2$. The rate of these chemical reactions increases exponentially with an increase in temperature. Exhaust gas recirculation introduced CO_2 and other inert gases into the cylinder, diluting the local oxygen concentration in the mixture. At the same time, the specific heat capacity of CO_2 and other inert gases was larger, which reduced the combustion temperature and inhibited the generation of NO_x emissions. When the EGR rate was increased to 15%, the PM emissions of diesel increased by 100%, whereas the PM emissions of B90DEC10 remained almost unchanged. For B90DEC10, the EGR rate of 15% achieved a better PM-NO_x trade-off.

Figure 17 illustrates the relationship between BTE and CO emissions at various EGR rates. While increasing the EGR rate could effectively reduce NO_x emissions, it also decreased the concentration of oxygen in the combustion chamber. This could limit combustion efficiency, resulting in reduced thermal efficiency and increased CO emissions. Therefore, the relationship between thermal efficiency and CO emissions could be used to determine an appropriate EGR rate (Natesan et al. Mar 2019). Diesel and biodiesel exhibited the highest thermal efficiency at an EGR rate of 5%, with only a minimal increase in CO emissions. Therefore, an EGR rate of 5% was more appropriate. If the

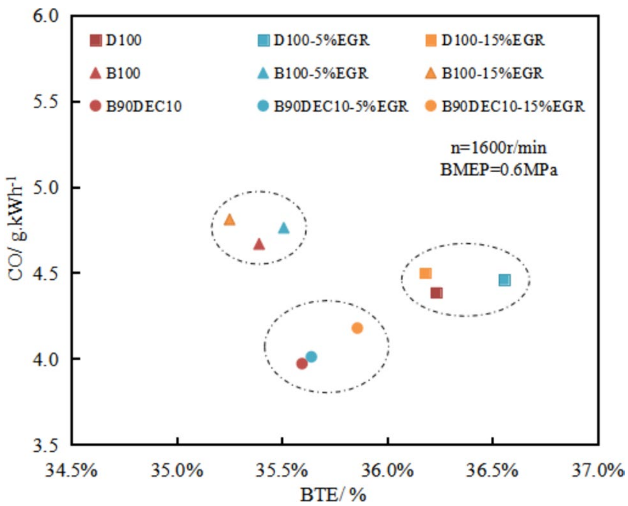

Fig. 17 The relationship between BTE and CO emissions at various EGR rates

EGR rate reaches 15%, B90DEC10 has the higher thermal efficiency and the lower CO emission compared to diesel and biodiesel. Because the EGR rate of 15% was conducive for reducing NO_x emissions, the most appropriate EGR rate for use with B90DEC10 was considered to be 15%.

Conclusion

The spray, combustion and emission characteristics of diesel, biodiesel and biodiesel/DEC mixtures were studied and compared in an unmodified CRDi diesel engine that met European VI emission standards. The following conclusions could be obtained through the above research.

The viscosity of B90DEC10 was reduced, and the atomization effect was improved compared to the other fuels. The maximum HRR of B90DEC10 was 20% greater than that of diesel at effective pressures below 0.6 MPa. The duration of the premixed combustion period was longer and the heat release rate (HRR) during this period was higher compared to diesel at effective pressures above 1.0 MPa. The BSFC of B90DEC10 was 2–5% greater than that of biodiesel. However, the BTE of B90DEC10 was 1.1–2.8% greater than that of biodiesel. If the EGR rate reached 15%, the peak in-cylinder combustion pressure and the maximum HRR decreased by 5 and 6%, respectively, and BTE was slightly reduced. The 15% EGR rate had minimal effect on engine thermal efficiency when using B90DEC10.

The oxygen content of fuel had a crucial role on the particulate matter emission. B90DEC10 produced the lowest PM emissions, with a 92% reduction compared to diesel, and disrupted PM-NO_x emissions trade-off relationship. The NO_x emissions of B90DEC10 were 8–20% larger than those of diesel. Exhaust gas recirculation significantly reduced NO_x emissions. If the EGR rate reached 15%, NO_x emissions of B90DED10 decreased by 45%. Interestingly, there was no significant increase in PM emissions observed. Under low load, the B90DEC10 had higher THC and CO emissions compared to diesel. Under medium and high loads, the pressure and temperature in the combustion chamber were enhanced, and the oxygen content of B90DEC10 was beneficial for the oxidation of THC and CO emissions, resulting in the lowest THC and CO emissions among the test fuels. B90DEC10 is therefore a potential alternative to diesel, with the best performance and emissions obtained when using a higher EGR rate (15% EGR).

Acknowledgements This work is supported by the National Key R&D Program of China (Grant No. 2021YFF0601004).

Author contributions LZ: Conceptualization, Methodology, Validation, Formal analysis, Investigation, Resources, Writing original draft, Writing review & editing, XZ: Investigation, Resources, Formal analysis, Writing original draft, RZ: Resources, Formal analysis, Writing original draft, JT: Investigation, Resources, Formal analysis, DQ: Resources, Supervision, Project administration, QL: Writing review & editing, Funding acquisition.

Data Availability All data, models, or code that support the findings of this study are available from the corresponding author upon reasonable request.

Declaration

Conflict of interest The authors have no financial or proprietary interests in any material discussed in this article.

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