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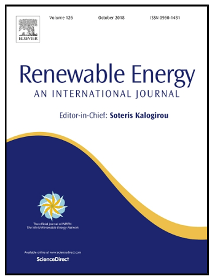

Effects of injection strategies on combustion and emission characteristics of a common-rail diesel engine fueled with isopropanol-butanol-ethanol and diesel blends

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Effects of injection strategies on combustion and emission characteristics of a common-rail diesel engine fueled with isopropanol-butanol-ethanol and diesel blends

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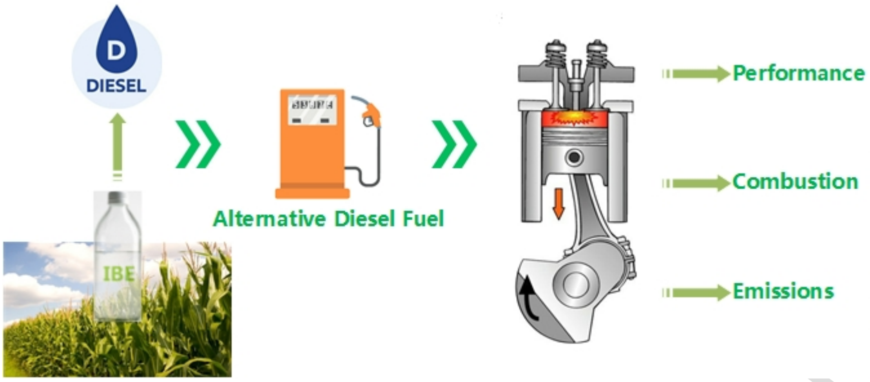
Abstract: This study is aimed to investigate the performance, combustion, and emissions of a common-rail diesel engine fueled with IBE and diesel blends. Two blends of IBE and diesel fuel, denoted as IBE15 (15% IBE and 85% diesel in volume) and IBE30 (30% IBE and 70% diesel in volume), were tested under different injection strategies. The experimental results show that compared with single injection, the in-cylinder pressure and heat release rate curves (*HRR*) for all the tested fuels under double injection cases are less severe. That is to say, a pilot injection can reduce knocking combustion and ringing intensity when blending a high ratio of IBE into diesel. Furthermore, double injection is helpful in improving both engine performance and economy for all the tested fuels, especially for IBE30. For almost all the tested conditions, both IBE15 and IBE30 present a potential to reduce soot emissions but increase NOx emissions. A pilot injection is favorable to reduce NOx emissions but causes the soot emissions to increase. Results also show that the flame lift-off length of IBE30 is much longer than pure diesel. This feature may result in better air-fuel mixing, which then contributes to reduce soot emissions.

Keywords: isopropanol-butanol-ethanol (IBE), IBE/diesel blends, injection strategy, combustion, emissions

Graphical abstract:



1. Introduction

Alternative fuels, such as ethanol, dimethyl ether, natural gas, bio-diesel, and butanol, are a good choice for conventional internal combustion engines to reduce the transportation fuel usage and to meet the stringent exhaust emission regulations [1-3]. Among those fuels, butanol has been considered as a promising replacement for diesel because of its good miscibility, high energy density and cetane number, low auto-ignition temperature, and evaporation pressure [4,5]. It has already been proven that after blending butanol into diesel, soot emissions of diesel engines could be drastically reduced [6,7]. Furthermore, high-ratio butanol and diesel blends coupled with EGR is capable of simultaneously reducing soot and NOx emissions [8]. In general, butanol is fermented from lignocellulosic (e.g., wheat straw, bagasse, switchgrass, corn straw, and barley straw, etc.) and non-cellulosic (e.g., sugarcane, glucose, sago, and corn, etc.) feedstock using *Clostridium beijerinckii* or *Clostridium acetobutylicum* [9-11]. Additionally, the lignocellulosic materials have been considered as the more abundant and cheaper feedstock for the fermentation process of butanol [12]. The intermediate fermentation product of butanol is acetone, n-butanol, and ethanol mixture (ABE) at a volume ratio of roughly 3:6:1. This ABE mixture undergoes a difficult separation and purification process to obtain pure butanol. Although butanol shows a great potential in diesel engines, the high separation cost and low production efficiency make it currently less competitive in comparison with diesel, gasoline, and ethanol.

In recent years, numerous researchers begun to directly use ABE as an alternative fuel in engines to utilize the numerous advantages of butanol and to avoid the aforementioned separation cost during butanol fermentation. Li et al. [13] investigated the water-containing ABE and gasoline blends in a spark ignition engine. They found that the addition of water-containing ABE in gasoline could improve the engine performance and reduce the CO and HC emissions. Nithyanandan et al. [14] compared the performance and emissions of different ABE fuels in a spark ignition engine. They reported that high-acetone ABE was more suited for use as an alternative fuel than low-acetone ABE when considering the thermal efficiency. Lee et al. [15] experimentally studied the combustion characteristics of ABE and diesel blends in a diesel engine. The results have shown that the indicated thermal efficiency increased with even a small ratio of ABE blended into diesel. Lin et al. [16] also investigated the combustion process of ABE in a diesel engine. They pointed out that the combustion process of ABE-containing fuels was premixed-dominant combustion. Additionally, there were some studies focused on the spray combustion characteristics and soot emission of ABE and diesel blends. Wu et al. [17,18] conducted various experiments in a constant volume chamber to investigate the spray combustion characteristics of the ABE blended with diesel. They found that ABE blended into diesel with a 50% ratio by volume could maintain diesel combustion characteristics but result in a lower natural flame luminosity and shorter combustion duration. Wu et al. [19] also conducted some experiments to study the soot formation of ABE and diesel blends. Their results showed that the soot cloud region and intensity reduced when ABE blended into diesel. Zhou et al. [20] also arranged various experiments to study the spray and soot emission characteristics of ABE and diesel blends in a constant volume chamber. They pointed that soot emissions of ABE and diesel blends could be further reduced by adopting low-temperature combustion. In order to study puffing and micro-explosions of ABE and diesel blends, Ma ex al. [21, 22] conducted various experiments on the droplet evaporation through the droplet suspension technique. They reported that strong puffing was observed during the ABE–diesel blends evaporation process at 823 K but currently no micro-explosion phenomenon was captured in their test conditions.

Even though ABE has numerous advantages when utilized in internal combustion (IC) engines, the poor properties of acetone in ABE restrict its further development. First, the acetone in ABE is corrosive to rubber engine parts [23-24]. Furthermore, the acetone in ABE has a very low flash boiling point, which makes ABE hard to transport and store in traditional implements [25]. In this respect, various metabolic engineering strategies have been employed to convert acetone into isopropanol, which seems to have more favorable physicochemical properties in comparison to acetone including higher viscosity, higher energy density, and higher flash point. Some gene-edited bacterial strains such as *Clostridium acetobutylicum ATCC824* [26], *Clostridium acetobutylicum DSM792-ADH* [27] and *Clostridium acetobutylicum BKM19* [28], that capable of substantially converting acetone into isopropanol were developed. The new biofuel consisting of isopropanol, butanol and ethanol (IBE) has already been studied by some researchers in internal combustion engines. Li et al. [29,30] experimentally studied the combustion and emissions of IBE and gasoline blends in a spark ignition engine. They indicated that the addition of IBE into gasoline could improve the engine thermal efficiency and reduce pollutant emissions. Lee et al. [31] has investigated the combustion and emission characteristics of the IBE in a diesel engine and found that with the addition of IBE lead to a reduction in soot emission. It also has been proved by Li et al. [32] that NOx and soot emissions can be reduced simultaneously by using high-ratio IBE under EGR conditions. Unfortunately, the previously studies only blended IBE in diesel with a relatively low ratio and tested the IBE in some conventional engine operation conditions. In this respect, it is necessary to further study the combustion and emission characteristics of the diesel engine fueled with IBE and diesel blends.

In this study, the authors conducted a detailed experimental study on a single-cylinder and common-rail diesel engine fueled with IBE and diesel blends to evaluate the potential of IBE as an alternative fuel. Two blends of butanol and diesel fuel, denoted as IBE15 (15% IBE and 85% diesel in volume) and IBE30 (30% IBE and 70% diesel in volume), were tested. Moreover, experiments are performed to investigate how the injection strategy influences the performance, combustion, and emissions of the diesel engine fueled with different fuels. The IBE mixture used in this study was mixed by analytical surrogate fuels with volumetric ratios of 3:6:1 to simulate the fuel mixtures from the fermentation process of butanol.

2. Test apparatus and procedure

2.1. Experimental setup

The experiments were conducted on a direct injection, single cylinder and common-rail diesel engine (AVL 5402) manufactured by the AVL company in Austria. The specifications of the test engine are presented in Table 1. An eddy current dynamometer (GE TCL-15, 4-35-1700) with a rated power up to 26.1 kW was connected to the test engine. This dynamometer has a three-phase asynchronous motor, which can both absorb and supply torque. The engine torque and speed were controlled by the Dyne system controller (DYN-LOC IV).

Table 1. The specifications of the diesel engine.

Items	Specifications
Model	AVL 5402
Type	1-cylinder, direct injection
Fuel injection system	BOSCH CP3 common rail
Bore (mm)× stroke (mm)	85 × 90
Displacement (L)	0.51
Compression ratio	17.5:1
Number of valves	4
Rated power (kw @ rpm)	6 @ 4200
Number of injection holes	5
Diameter of injection holes (mm)	0.18

Fig. 1 shows the schematic of the experimental apparatus used in this study. The fuel injection system of the test engine was an advanced common-rail system, which is capable of four injections per cycle and a rail pressure up to 100 MPa. This common-rail system is coupled with an open-loop prototype ETAS engine control unit designed by AVL. The injection parameters such as injection quantity, injection pressure, and injection timing could be controlled using the INCA software. Two fuel pumps (a high-pressure pump and a low-pressure pump) were used in the fuel supply system to provide a stable supply of fuel from fuel tank to the common-rail.

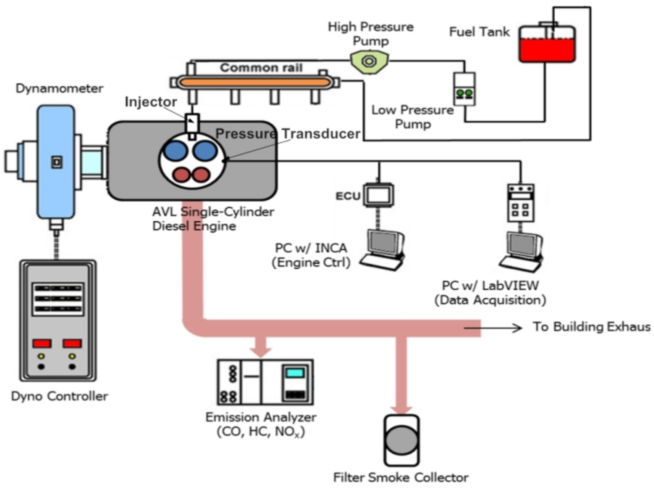


Fig.1. Schematic of experimental apparatus.

Air-fuel ratios and NOx emissions were measured using a Horiba MEXA-270 analyzer. HC and CO emissions were measured using a Horiba MEXA-554JU analyzer. Soot emissions were obtained using a standard filter paper method, which will be described at the later section. The in-cylinder pressure signal was measured with a piezoelectric pressure transducer (6152B, Kistler) coupled with a charge amplifier (3057-AO1, AVL). Moreover, a crank angle encoder (BEI XH25D) with a resolution of 0.1° crank angle (CA) was mounted at the end of the crankshaft to record the crank angle position. In each test condition, the in-cylinder pressure data was collected over 150 continuous cycles with a crank angle (CA) sampling interval of 0.25 °CA by a LabVIEW acquisition system, which consists of a NI SCXI-1000 board, a NI PCI-MIO-16E4 board and a self-made LabVIEW code. The in-cylinder pressure data were averaged to eliminate the effect of cycle-to-cycle variations. The various combustion parameters such as heat release rate (*HRR*) and combustion phasing were calculated from the averaged pressure trace. The measuring accuracy of each instrument is given in Table 2.

Table 2. Accuracy of measured parameters.

Equipment	Measured parameters	Range	Accuracy

Eddy current dynamometer	Speed	1-5000 rpm	0.2%
	Torque	0-300 N m	0.5%
Horiba MEXA-554JU analyzer	CO emission	0-10 vol.%	0.06%
	HC emission	0-10000 ppm	1 ppm
Horiba MEXA-270 analyzer	NOx emission	0-3000 ppm	3%

2.2. Experimental procedure

Two blends of butanol and diesel fuel, denoted as IBE15 and IBE30, were tested in this study. Detailed experiments were carried out at different injection strategies at the engine load of 0.54 MPa (brake mean effective pressure, BMEP) and an engine speed of 1500 r/min. To investigate the effects of the single injection, the engine was operated only with a main injection with its timing varied from 6 °CA to 18 °CA BTDC in steps of 3 °CA. To investigate the effects of double injection, the engine was operated with the main injection coupled with a pilot injection. In this stage, the pilot injection timing (PIT) was swept from 25 °CA to 45 °CA BTDC in steps of 5 °CA, but the main injection timing (MIT) was fixed at 9 °CA BTDC. The detailed test conditions are presented in Table 3. At the beginning of the test, the engine was warmed-up until the coolant and lubricating oil temperature reached about 85℃. Then the engine was adjusted to the test conditions using the INCA software. After the engine had reached steady-state, the in-cylinder pressure and exhaust emissions were recorded. Each test condition was repeated more than three times to reduce the experimental uncertainties and to ensure that the results were repeatable.

Table 3. Test conditions.

Items	MIT Cases	PIT Cases

Fuels	Diesel, IBE15, IBE30	
Engine speed (rpm)	1500	
Engine load (MPa)	0.54	
Pilot injection timing (°CA BTDC)	None	25,30,40,45
Main injection timing (°CA BTDC)	6,9,12,15,18	9
Injection pressure (MPa)	60	

2.3. Test fuels

Commercial diesel with a sulfur content lower than 50 ppm was selected as baseline fuel in this study. The IBE mixture was prepared with analytical grade surrogate fuel, namely isopropanol (99.5%), n-butanol (99.5%) and ethanol (99.8%). A temperature-controlled magnetic stirrer was used to mix the IBE mixture to a volumetric ratio of 3:6:1 (A:B:E). This ratio was to simulate the composition of the fuel mixtures from the fermentation process of bio-butanol. The detailed physicochemical properties of the test fuels are listed in Table 4. It should be noted that the cetane number of the individual fuels in Table 4 was taken from reference [33]. The properties of the IBE mixture were calculated according to simple mixing rules in [34]. Afterwards, the IBE mixture were mixed with diesel to prepare the fuel blends. The fuel blends used in this study were denoted as IBE15 and IBE30, which means the volumetric ratios of the IBE mixture in the fuel blends were 15% and 30%, respectively.

Table 4. Fuel properties.

Parameters	Individual fuels				Fuel blends
	Diesel	ethanol	Isopropanol	n-Butanol	IBE
Chemical formula	C_{10}-C_{22}	C_2H_5OH	C_3H_7OH	C_4H_9OH	-

Cetane number	52.65	8	12	15.92	13.952
Octane number	-	100	112	87	95.8
Oxygen content (wt.%)	-	34.8	26.6	21.6	24.4
Density (kg/m^3)	820-860	795	786	813	803.1
Lower heating value (MJ/kg)	42.7	26.8	30.4	33.1	31.7
Boiling temperature (℃)	282-338	78	84	118	-
Latent heat (kJ/kg)	260	904	758	582	667
Stoichiometric AFR	14.3	9.0	10.4	11.2	10.7
Auto-ignition temperature	250	420	399	343	-

2.4. Data processing

2.4.1. Brake Specific Fuel Consumption

Since IBE and diesel have different lower heating values, the equivalent heat brake specific fuel consumption (BSFC) was used in this study to indicate the thermal efficiency. The equivalent BSFC could be calculated as follows.

$$\mathrm{BSFC} = B_e \times \frac{H_D \times \rho_D \times V_D + H_{IBE} \times \rho_{IBE} \times V_{IBE}}{H_D \times \left(\rho_D \times V_D + \rho_{IBE} \times V_{IBE} \right)} \tag{1}$$

where, BSFC represents the equivalent BSFC of each fuel, namely pure diesel, IBE15 or IBE30. B_e represents the actual BSFC of each fuel. V_D(V_{IBE}) represents the volumetric fraction of diesel (or IBE). ρ_D(ρ_{IBE}) represents the density of diesel (or IBE). H_D(H_{IBE}) represents the lower heating value of diesel (or IBE). It should be noted that the lower heating value of IBE used in this study was calculated according to the simple mixing rules described in reference [34].

2.4.2. Soot Emissions

Soot emissions were measured using a standard filter paper method. Raw exhaust gases were drawn through a 7/8” round filter paper using a vacuum pump. Rectangular strips of filter paper supplied by Grainger Industrial Supply (#6T167) were cut into discs and placed in a filter holder taken from a Bacharach True-Spot smoke meter. Condensed water and oil were removed by installing a line filter after the vacuum pump. The filter paper blackening (PB) was measured using a digital scanner after the samples were collected. The PB value could be considered as the filter smoke number (FSN) as described in [14,15], given by

$$\mathrm{PB} = \left(100 - \frac{R_P}{R_f} \times 100\%\right) / 10 \qquad (2)$$

where R_p and R_f are the reflectometer value of sample and unblackened paper, respectively.

2.4.3. Combustion Phasing

The ignition delay is defined as the interval of crank angle between the start of injection and the start of combustion (SOC). The combustion duration is defined as the interval of crank angle between the start of combustion and the end of combustion (EOC) [35]. In this study, the start of injection was recorded from the INCA software. The SOC and EOC were calculated according to the first law of thermodynamics. Firstly, the net heat release rate was calculated as follows:

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} V \frac{dV}{d\theta} + \frac{1}{\gamma - 1} P \frac{dV}{d\theta} \qquad (3)$$

where γ is the ratio of specific heats, P and V are the in-cylinder pressure and volume at a crank angle, respectively. Then the cumulative heat release from the air-fuel mixture can be obtained by integrating Eq. (3). Subsequently, the crank angle where 10% and 90% of the total heat released can be found to define SOC and EOC, respectively [36]. The combustion center, which defined as the crank angle where 50% of the total heat released, can be found in the same way.

3. Results and discussion

3.1. Effects of single injection

3.1.1. Combustion characteristics

Fig. 2 shows the in-cylinder pressure and *HRR* for different fuels at various MITs. The results show that the MIT has remarkable influence on the combustion process of the tested fuels. When the MIT is advanced, the peak in-cylinder pressure and *HRR* increase for all the test fuels significantly. The main reason is the temperature at the crank angle where the fuel injected (T_{inj}) [37]. When the MIT is gradually advanced, T_{inj} decreases. Thus, the fuel injected to the cylinder undergoes a long ignition delay, which can potentially increase the amount of homogeneous air-fuel mixture formed during the ignition delay and therefore enhance the premixed combustion. For all the tested fuels, the combustion processes advanced as the MIT is advanced. However, as seen from an equal MIT condition, the combustion processes of IBE15 and IBE30 are delayed in comparison to pure diesel, especially in the case of IBE30. This is mainly because the combustion processes of IBE15 and IBE30 are also greatly affected by the fuel properties of IBE, including the cetane number and latent heat. When IBE and diesel blends are injected into the cylinder, they need to absorb heat to vaporize and hence lower the in-cylinder temperature at the compression stroke. On the other hand, the two fuel blends are hard to ignite at low temperatures due to their poor ignitability and low activation energy. All those factors contribute to prolonging the ignition delay and retarding the combustion process. It also can be seen that almost in all test cases, the peak values of the in-cylinder pressure and *HRR* for the two fuel blends are much higher than that of pure diesel because of the enhanced premixed combustion when IBE is blended in diesel. Nevertheless, at the MIT of 6 °CA BTDC, the pressure curve for the IBE30 exhibits a lowest peak in comparison to pure diesel and IBE15 due to the over retarded combustion process.

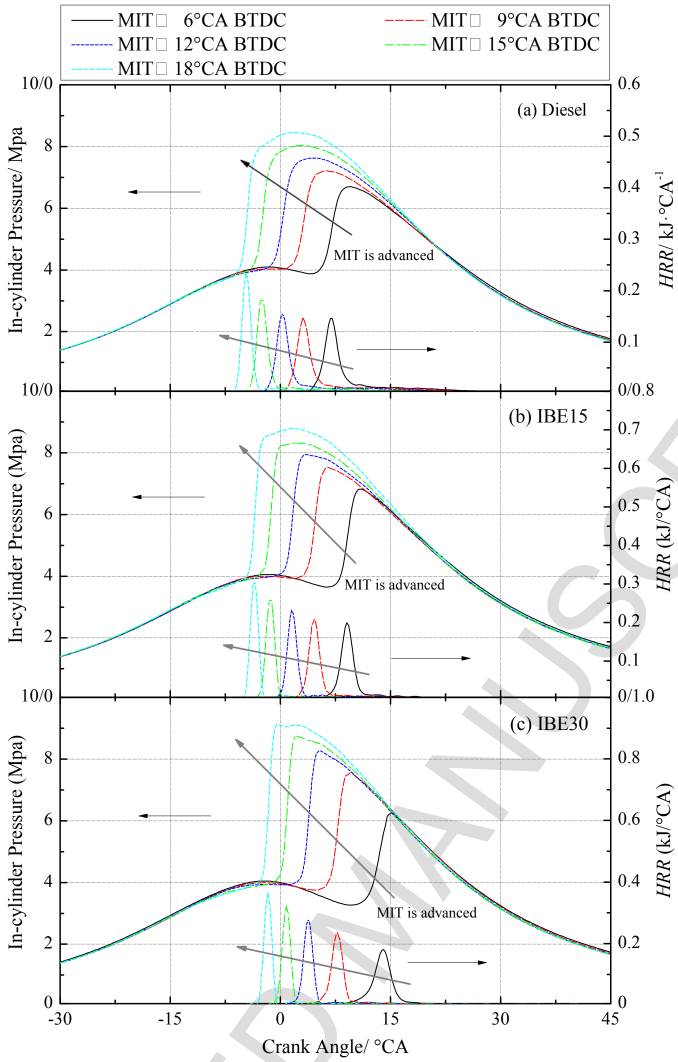


Fig. 2. In-cylinder pressure and *HRR* for different fuels with various MITs.

Fig. 3 displays the changes of the ignition delay and combustion duration of different fuels with various MITs. It can be seen that for all test fuels, the ignition delay increases as the MIT is advanced. Additionally, the blend of IBE in diesel has further prolonged the ignition delay for the same reason discussed in the above section. For the combustion duration, it noticeably shortened when the MIT is advanced. This is mainly because at an early MIT, an abundant amount of homogeneous air-fuel mixture is formed during the ignition delay and afterwards burns at a rapid speed. In most test conditions, the combustion duration for the two fuel blends is shorter than that of pure diesel. One reason is that the OH radicals produced by IBE improves the combustion speed. Another reason is that the long ignition delay from the IBE-diesel blend enhances the premixed combustion. However, at 6 °CA BTDC, IBE30 has the longest combustion duration in comparison to diesel and IBE15. This is because at 6 °CA BTDC, the combustion phasing of IBE30 has been overly retarded, which noticeably slows down the combustion speed of air-fuel mixture and causes the combustion of IBE30 to deteriorate.

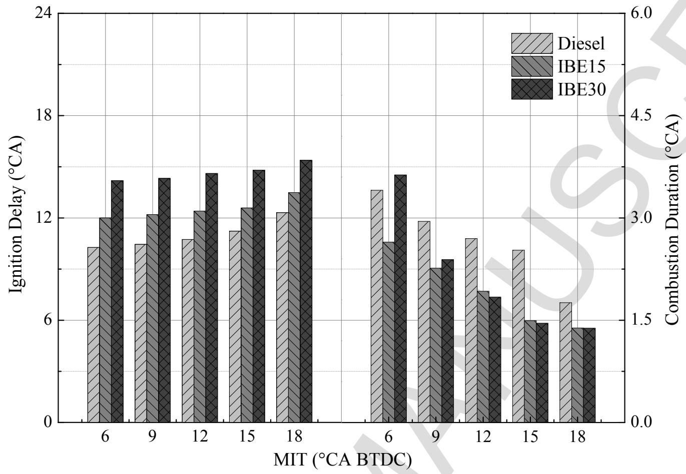


Fig. 3. Ignition delay and combustion duration for different fuels with various MITs.

3.1.2. Engine performance

The effects of MIT on the BTE and BSFC of different fuels have been illustrated in Fig. 4. The BTE indicates how well an engine can convert the chemical energy in a fuel into mechanical energy. It can be seen that for all the tested fuels, BTE first increases and then decreases when the MIT is advanced. At the beginning, the combustion process is gradually swept to TDC when the MIT is advanced, which causes the BTE to increase. Then when the MIT is further advanced, BTE reduces because the combustion process is swept to the compression stroke and separated from TDC. Compared with pure diesel and IBE30, the BTE of IBE15 is always higher due to the reduced combustion duration and increased extent of constant volume combustion. Although the premixed combustion of IBE30 is stronger than that of pure diesel and IBE15, the BTE of IBE30 is the lowest almost in all the test conditions. At small MIT, the low BTE of IBE30 is caused by the retarded combustion process, but at large MITs, it is caused by the increased negative work. It also can be seen that BSFC shows an opposite trend with BTE. When the MIT is advanced, it first decreases and then increases for all the tested fuels. Almost in all the test conditions, the BSFC for IBE15 is lower, but for IBE30 it is higher in comparison to pure diesel. First, the premixed combustion is enhanced when IBE is blended in diesel and thus contributes to a reduction in BSFC. Second, more fuel is needed for maintaining the engine runs at the same load since the energy density is reduced when a large amount of fuel IBE is blended in diesel. In the competition of all these mechanisms, the previous mechanism plays the dominate role for IBE15 and the last mechanism plays the dominate role for IBE30. The optimal engine performance and economy for the pure diesel and IBE15 is found at 12 °CA BTDC but for IBE30 is found at 9 °CA BTDC. The main reason is that IBE30 has the strongest premixed combustion among the tested fuels, thereby it can weaken the negative effect of retarded MIT and combustion phasing. This result means that IBE30 is more tolerant to retard MIT than pure diesel and IBE15.

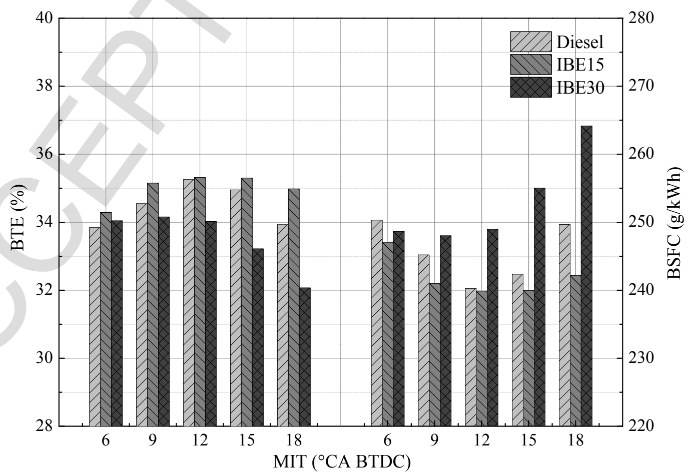


Fig. 4. BTE and BSFC for different fuels with various MITs.

3.1.3. Emission characteristics

The variation of the CO and HC emissions with different MITs for the different fuels are given in Fig. 5. It can be clearly seen that for all of the tested fuels, the CO and HC emissions reduce when the MIT is advanced from 6 to 15 °CA BTDC. This is mainly because the advanced MIT results in longer ignition delay and higher combustion efficiency, and hence reduce the fuel consumption and incomplete combustion. However, when the MIT is further advanced, the CO and HC emissions for all of the tested fuels increase. The main reason is that as the MIT is further advanced, the whole combustion process is swept towards the compression stroke, which causes the negative work to increase significantly. In this stage, more fuel is needed to maintain the engine runs at an identical load and then the air-fuel mixture undergoes a rich combustion. Also, it can be seen that almost in all of the test conditions, the CO and HC emissions of the IBE15 and IBE30 are higher than that of diesel emissions. This can be explained by the following reasons. First, the poor ignitability and high cooling effect of IBE causes the in-complete combustion to increase. Second, some unburned IBE-air mixtures are exhausted into emissions directly during the scavenging process. Third, the quenching effect at the chamber crevice and cylinder wall, which may cause HC emissions to increase, is strengthened when the IBE mixture is blended into diesel. At the MIT of 18, the CO and HC emissions of IBE15 and IBE30 are lower than that of pure diesel. This is mainly because the oxygen content and OH radicals produced by IBE suppresses the increase of CO and HC emissions at the over advanced MIT.

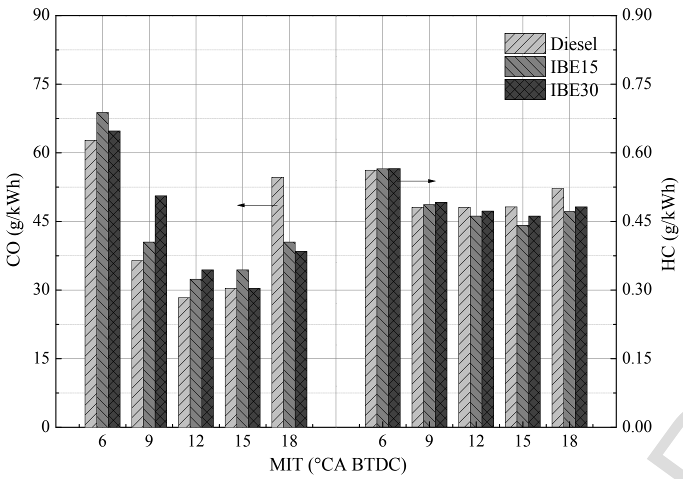


Fig. 5. CO and HC emissions for different fuels with various MITs.

The changes of NOx and soot emissions along with MITs for different fuels are presented in Fig. 6. It is observed that NOx emissions of all the tested fuels decrease notably as the MIT is advanced due to the enhanced premixed combustion. Furthermore, NOx emissions for the two fuel blends are higher than that of pure diesel since the premixed combustion is further enhanced and oxygen content is increased when IBE is blended in diesel. For the soot emissions, it first decreases as the MIT is advanced from 6 to 15 °CA BTDC but then increases as the MIT is advanced to 18°CA BTDC. This is because at the MIT of 18°CA BTDC, the increased negative work and deteriorated combustion efficiency makes the fuel consumption increase and the air-fuel mixture to become rich. In addition, it is observed that the soot emissions of IBE15 and IBE30 are always lower than that of pure diesel. The detailed mechanisms are shown as follows. First, the smoke precursors can be effectively oxidized by the OH radicals produced by IBE. Second, the alcohol fuel has a higher chance to prevent the formation of polycyclic aromatic hydrocarbons, which is the main ingredient of soot precursors [38-39]. Finally, the premixed combustion is enhanced and the diffusion combustion is weakened when the IBE is blended into diesel, and thus may have the potential to further reduce soot emissions.

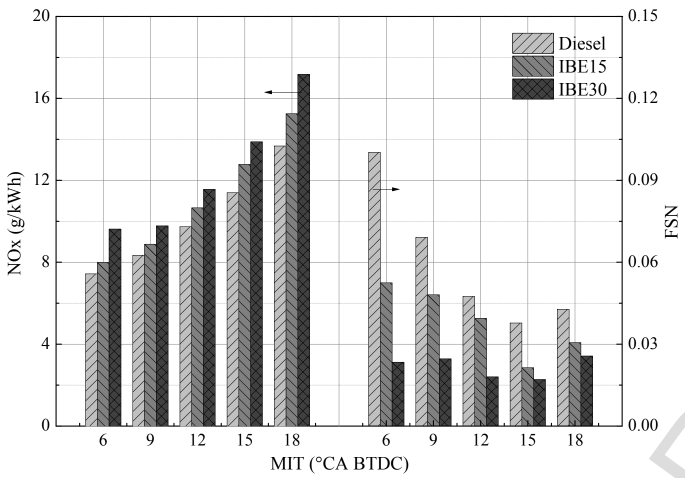


Fig. 6. NOx and soot emissions for different fuels with various MITs.

3.2. Effects of double injection

3.2.1 Combustion characteristics

Fig. 7 shows the in-cylinder pressure and *HRR* for different fuels at various PITs. The results show that with a pilot injection, the pressure and *HRR* curve for all the tested fuels become less severe and their peak values are noticeably lower in comparison to the single-injection conditions. This is because the homogeneous air-fuel mixture and activated energy prepared by the pilot injected fuels shorten the ignition delay and weaken the premixed combustion [40]. The gentle pressure and *HRR* curve lead to the reduction in engine noise and ringing intensity. In most test conditions, the peak in-cylinder pressure and *HRR* noticeably increase as the PIT is advanced. The main reason is that more homogeneous air-fuel mixture formed during the ignition delay at large PIT, which afterwards leads to a strong premixed combustion. The *HRR* curve for diesel and IBE15 presents a double or triple peak profile at a small PIT while a single peak profile at large PIT. However, for IBE30, its *HRR* curve always presents a single peak profile due to prolonged ignition delay and enhanced premixed combustion. For diesel and IBE30, low temperature combustion can be seen from the *HRR* curve at the PIT of 25 °CA BTDC.

Nevertheless, the same phenomenon cannot be found at the *HRR* curve of IBE30 since its poor ignitability. When the PIT is larger than 25 °CA BTDC, the low temperature combustion disappears for all the tested fuels since the pilot fuel is hard to auto-ignite at an even lower temperature condition.

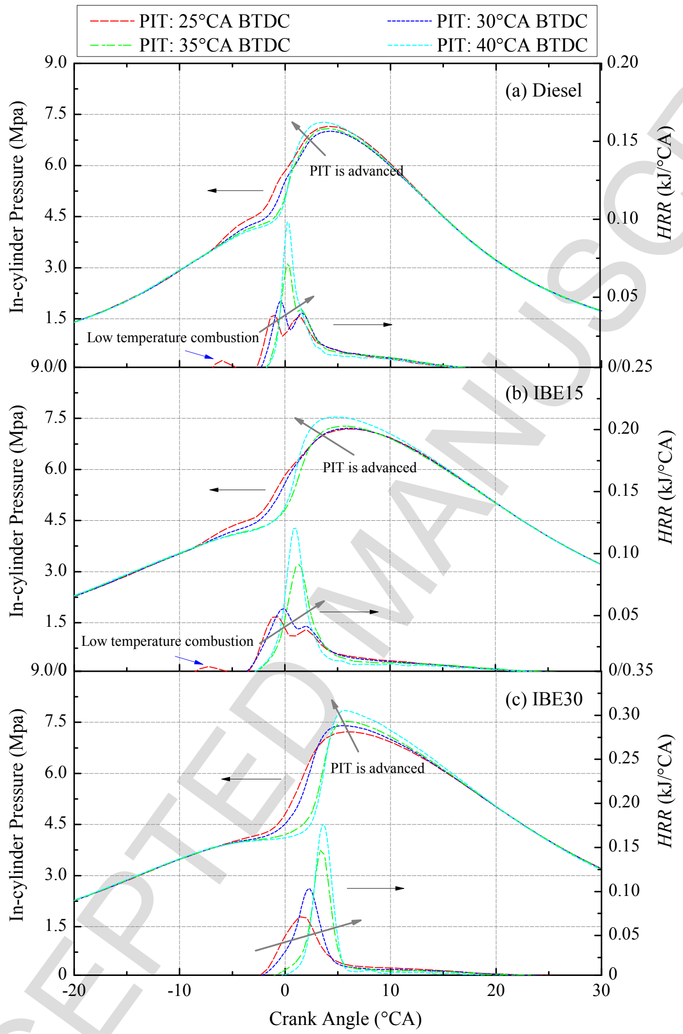


Fig. 7. In-cylinder pressure and *HRR* for different fuels with various PITs.

Fig. 8 displays the changes of the ignition delay and combustion duration of different fuels with various PITs. It can be observed that, with a pilot injection, the ignition delay is shortened and the combustion duration is prolonged for all the tested fuels. At the PIT of 25 °CA BTDC, the combustion duration of the diesel and IBE15 are almost three times longer than that of single injection due to diffusion combustion. When the PIT is advanced, the ignition delay for all the tested fuels is prolonged. This is attributed to the reduced temperature at the point where the pilot fuel is injected. This prolonged ignition delay results in a stronger premixed combustion and a shorter combustion duration as shown in Fig. 8. Compared with pure diesel, the two fuel blends always show a longer ignition delay but shorter combustion duration in the tested conditions because the fuel properties of IBE discussed before.

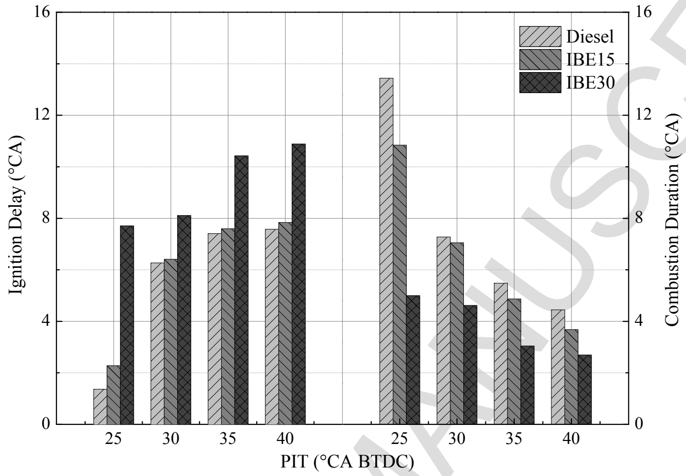


Fig. 8. Ignition delay and combustion duration for different fuels with various PITs.

3.2.2 Engine performance

The effects of PIT on the BTE and BSFC of different fuels are indicated in Fig. 9. The results show that for almost at all the tested conditions the engine performance and fuel economy for the tested fuels improved with a pilot injection because the pilot helped the main spray develop and combust, unlike the results of single injection. IBE30 exhibits a better engine performance and fuel economy than diesel at most tested conditions when the pilot injection is adopted. This is mainly because the strong premixed combustion of IBE30 is further enhanced and promoted by the pilot fuel. The results also show that when the PIT is advanced, both the engine performance and fuel economy first deteriorate and then improve. This is because the negative effect of the retarded combustion process is strengthened when the PIT is gradually swept from 25 to 35 °CA BTDC, but then weakened by the significantly enhanced premixed combustion at the PIT of 40 °CA BTDC.

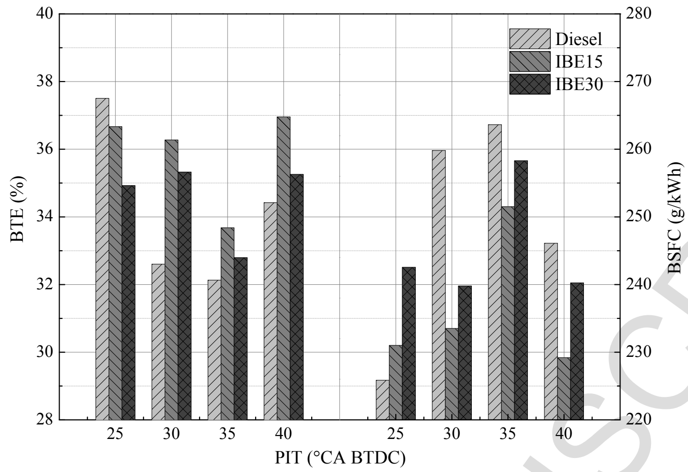


Fig. 9. BTE and BSFC for different fuels with various PITs.

3.2.3. Emission characteristics

The variation of the CO and HC emissions with different PITs for the different fuels are given in Fig. 10. The results show that when the PIT is advanced, the CO and HC emissions for the double injection are first lowered and then increased in comparison to the single injection conditions. In general, the evaporation or combustion of the pilot fuel can promote the combustion of the main injection fuel and thus cause the CO and HC emissions to reduce. Nevertheless, when the PIT is over advanced, the pilot fuels are hard to auto-ignite and undergo a poor evaporation process, which may have a negative effect on the main injection. In addition, the over advanced PIT postpones the combustion phasing significantly, thus more unburned fuels are trapped in crevices and deposits escape to exhaust emissions directly. It also shows that a small PIT is helpful for IBE15 and IBE30 to reduce CO and HC emissions. Compared with pure diesel, CO and HC emissions for the two fuel blends are lower at small PIT. However, when the PIT is advanced, the CO and HC emissions for IBE15 and IBE30 increase rapidly. Eventually, the CO and HC emissions for IBE15 and IBE30 are higher than that of pure diesel.

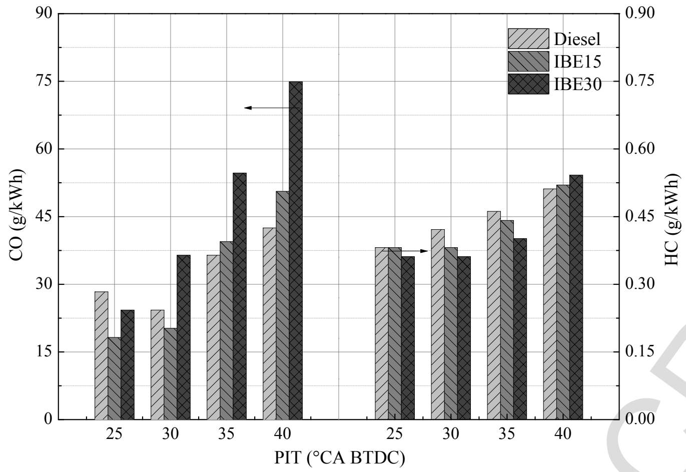


Fig. 10. CO and HC emissions for different fuels with various PITs.

The changes of NOx and soot emissions along with PITs for different fuels are presented in Fig. 11. Compared to the previous case with no pilot injection, it can be clearly seen that the addition of a pilot injection can reduce NOx emissions by almost a quarter at the same power output. This is mainly because the pilot injection helps to shorten the ignition delay, and then contributes to the improvement of the diffusion combustion and weakens the premixed combustion. However, the soot emissions for the double injection increase in comparison with the single injection condition. This is also due to the improved diffusion combustion. Additionally, it can be found that when PIT is advanced, NOx emissions for all the tested fuels slightly increase since the premixed combustion is enhanced by the prolongation of ignition delay and the enhancement of premixed combustion. On the contrary, soot emissions decrease for all the tested fuels as the PIT is advanced due to the same reason. Compared with pure diesel, both IBE15 and IBE30 also present a promising potential to reduce soot emissions under the conditions of double injection.

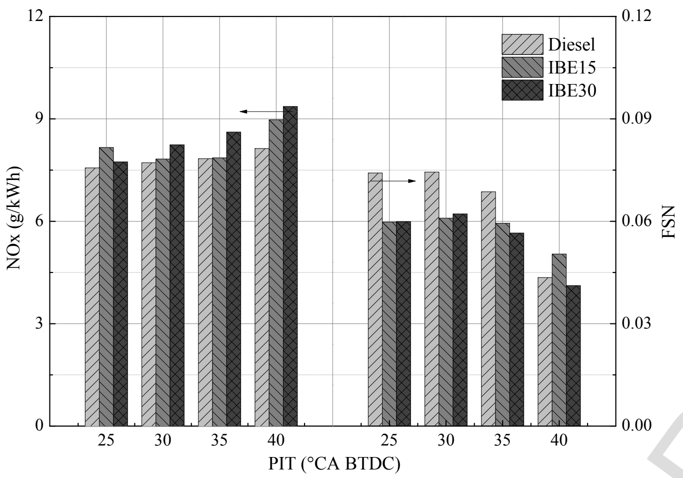


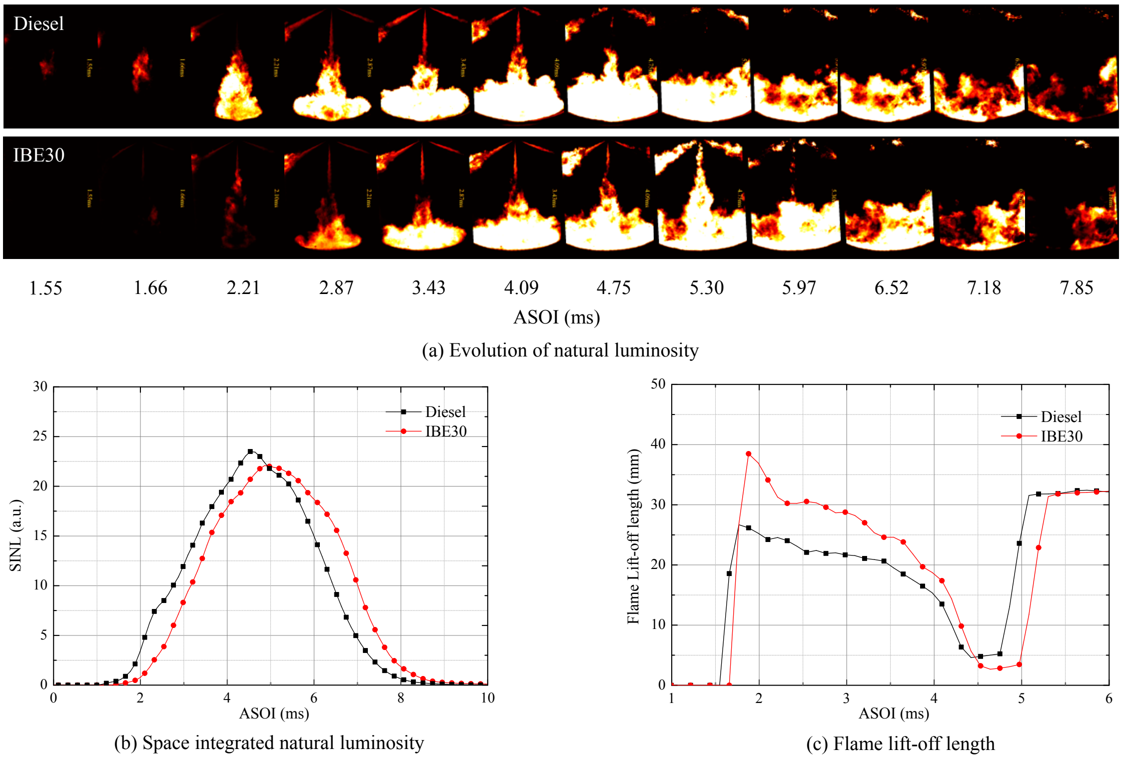
Fig. 11. NOx and soot emissions for different fuels with various PITs.

3.3. Visualization of combustion process

In order to further study the combustion process of the IBE and diesel blends, detailed experiments were conducted on a premixed combustion heated constant volume chamber. The detailed specifications of the chamber and the schematic of the chamber setup are given in [17-19]. The difference is that the injector used in current work was uninstalled from the AVL diesel engine. It had six holes and the orifice diameter measured by a microscopic was 0.15 mm. During the experiments, the injection duration and pressure were maintained for 3 ms and 60 MPa, respectively. Furthermore, the ambient density of the chamber was kept at 15 kg/m^3 and the ambient temperature at the time where the fuel injected was kept at 1000 K. Moreover, the experiments were conducted under a normal oxygen concentration, namely 21%, to simulate the engine combustion without EGR.

The combustion process of the pure diesel and IBE30 in the constant volume chamber are displayed in Fig. 12. Fig. 12(a) shows the natural flame evolution of pure diesel and IBE30. The original gray images have been converted to pseudo-color images in MATLAB for a better display effect. The colors in the images only represent an arbitrary unit scale consistent over the whole combustion process. The images presented in a single row are sequences from one individual injection. In each image, the center of the top edge is the injector tip while the other edge is the curved wall. It can be found that compared with pure diesel, the intensity of the flame luminosity of the fuel blends is much lower. Since the natural luminosity is mainly contributed by the soot incandescence, it is reasonable to say this result is a strong support of the FSN results recorded from diesel engine. The ignition delay or illumination delay can be identified as the interval between the injection timing and the start point of the flame luminosity. It can be seen that the IBE30 has a longer ignition delay over pure diesel. That is also exactly the same result as diesel engine.

Fig. 12 (b) gives the changes of the space integrated natural luminosity (SINL). The SINL was calculated by summing up the values of all pixel in the entire luminosity image and averaged over five different runs at the same experimental conditions. It is shown that the SINL of IBE30 is lower than that of pure diesel. This is mainly due to the lower soot emissions of IBE30. The flame lift-off length for both pure diesel and IBE30 are given in Fig. 12 (c). The flame lift-off length is defined as the interval between the injector tip and the point where significant flame luminosity is captured. The results shown that the flame lift-off length of IBE30 is longer than that of pure diesel because that the flame lift-off length is mainly controlled by the ignition delay, and a longer ignition delay usually results in a longer flame lift-off length. In general, the fuel undergoes more time to mixing at long flame lift-off length, hence the equivalence ratio at the flame zone can be effectively reduced. This can be used to explain the results why soot emissions of IBE30 are lower than that of pure diesel.



(a) Evolution of natural luminosity

(b) Space integrated natural luminosity

(c) Flame lift-off length

Fig. 12. Combustion process of pure diesel and IBE30 in a constant volume chamber.

4. Conclusions

A detailed experimental investigation was conducted on a single cylinder and common-rail diesel engine fueled with IBE and diesel blends to evaluate the effects of injection strategies on engine performance, combustion, and emissions. The conclusions are drawn as follows:

(1) In the single injection cases, the peak in-cylinder pressure and *HRR* increase significantly for all the tested fuels and their peak values are gradually advanced as the MIT is advanced. Compared with pure diesel, the peak in-cylinder pressure and *HRR* of IBE15 and IBE30 are higher and their peak values appear later. Taking in consideration of the engine performance and economy, the optimal MIT for the pure diesel and IBE15 is 12 °CA BTDC but for IBE30 is 9 °CA BTDC.

(2) Additionally, CO and HC emissions for all the tested fuels reduce when the MIT is advanced from 6 to 15 °CA BTDC, but then increase when the MIT is further advanced. At

almost all test conditions, both IBE15 and IBE30 can effectively reduce soot emissions, but contribute to an increase in NOx emissions. Further optimization for soot emissions can be achieved by advancing the MIT, but the NOx emissions may increase in this case.

(3) When the pilot injection is adopted, the pressure and *HRR* curve for all the tested fuels become less severe and their peak values are lower in comparison to the single-injection condition. The *HRR* curve for diesel and IBE15 presents a double or triple peak profile at a small PIT while a single peak profile at large PIT. However, for IBE30, its HRR curve always presents a single peak profile. Furthermore, a pilot injection is helpful to improve both engine performance and economy for all the tested fuels, especially IBE30.

(4) A small PIT is helpful to reduce CO and HC emissions but a large PIT causes the CO and HC emissions to increase. Additionally, it is found that CO and HC emissions for the two fuel blends are lower than that of pure diesel at a small PIT. However, when the PIT is advanced, the CO and HC emissions for IBE15 and IBE30 increase rapidly, and eventually higher than that of pure diesel. A pilot injection is favorable to reduce NOx emissions, but results in higher soot emissions. Furthermore, as the PIT is advanced, the NOx emissions are slightly elevated but the soot emissions decrease significantly.

(5) Compared with pure diesel, the intensity of flame luminosity of the IBE30 is lower and the illumination delay is shorter. Moreover, the flame lift-off length of IBE30 is much longer than pure diesel. The longer flame lift-off length means the fuel undergoes more time to mixing, which causes the equivalence ratio at the flame zone to reduce. This feature may contribute to lean homogenous combustion and less soot emissions.

(6) From the results presented in this study. The optimum condition is IBE15 with main

injection timing of 9 °CA BTDC and pilot injection of 30 °CA BTDC. At this condition, the engine has high BTE but low BSFC. Furthermore, both NOx and soot emissions are kept in a low level.

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Abbreviation

ABE	acetone-butanol-ethanol
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
BTDC	before top dead center
BTE	brake thermal efficiency
CA	crank angle
EOC	end of combustion
FSN	filter smoke number

HC	hydrocarbons
HRR	heat release rate
IBE	isopropanol-butanol-ethanol
MIT	main injection timing
NOx	nitrogen oxide
PB	paper blackening
PIT	pilot injection timing
SI	spark ignition
SINL	space integrated natural luminosity
SOC	start of combustion
TDC	top dead center
T_{inj}	temperature at the crank angle where the fuel injected

Highlights

Characters of IBE and diesel blends at different injection strategies were studied.

Injection strategy affects the combustion process of IBE and diesel blends greatly.

Maximum pressure and HRR are obviously reduced when adopting a pilot injection.

Both engine performance and economy are improved under double injection conditions.

Double injection is helpful to reduce NOx emissions while increase soot emissions.