Enabling Dual Fuel Sequential Combustion using Port Fuel Injection of High Reactivity Fuel Combined with Direct Injection of Low Reactivity Fuels

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Abstract: This paper presents a preliminary experimental study on the combustion and emission characteristics of Dual Fuel Sequential Combustion (DFSC) mode, in which port fuel injection of n-heptane combined with in-cylinder, directly injected ethanol, n-butanol and n-amyl alcohol are used in a single-cylinder engine at fixed directly injection timing. The results show that the heat release can be divided mainly into three stages: low temperature reaction, high temperature reaction of n-heptane and the directly injected fuel combustion stage. The amount of port injected n-heptane plays a key role in the maximum in-cylinder pressure (P_{max}), maximum in-cylinder mass averaged temperature (T_{max}) and the maximum pressure rise rate. For the high overall lower heating values (LHVs) per-cycle, the CO emissions decrease with the increase of the premixed ratio. By contrast, the CO emissions increase with the premixed ratio when the overall LHVs per-cycle are kept at medium and low levels. The NO_x and soot emissions are all kept at low levels for the experimental conditions. In particular, the higher latent heat, lower cetane values and the shorter carbon chains associated with ethanol lead to lower NO_x and soot emissions than those of n-butanol and n-amyl alcohol. When directly injection of n-butanol and at low loads, with optimized premixed ratio, the indicated thermal efficiency can be higher than 46% meanwhile maintaining low emissions.

Key words: DFSC, n-heptane, ethanol, n-butanol, n-amyl alcohol, combustion and emissions

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1. Introduction

To address global energy problems and increasingly stringent emission regulations, biomass-based alternative fuels and new combustion modes are being widely developed [1, 2]. As alternative fuels, biomass-based fuels, which are commonly known as biofuels, possess many advantages over fossil fuels [3]. First, biofuels are easily available from common biomass sources. Second, the application of biofuels enables carbon circulation between the air and fuel, meaning problems such as greenhouse gas emissions and energy shortages can be solved together. Third, most biofuels, such as biodiesel and ethanol, have suitable physicochemical properties that permit effective combustion in internal combustion engines with or without minor modifications.

Ethanol is the only large-scale renewable fuel used currently [4, 5, 6, 7]. Because the physical and chemical properties of ethanol are similar to that of gasoline, ethanol is added to gasoline in the United States, Canada, China and many other countries. A large number of studies have focused on the application of gasoline mixed with ethanol in a spark ignition (SI) engine [8, 9, 10]. Meanwhile, to realize the high efficiency of ethanol, and to find a solution to the problem of high NO_x and soot emissions from traditional diesel engines, many scholars have studied using ethanol in a diesel engine [11, 12, 13]. He et al. [14] studied the mixing proportion of ethanol in diesel fuel blends and noted that as the proportion of ethanol mixed with diesel increases, the density, cetane number, kinematic viscosity, higher heating value and aromatic fractions of the blends decrease. Ignition improvement is needed to enhance the cetane number. Lu et al. [15] studied the combustion of diesel/ethanol blends and found that the combustion efficiency can be improved. For higher ethanol blending ratios, NO_x and soot amounts decrease, whereas CO emissions increase. Li et al. studied the combustion and emissions characteristics of diesel/ethanol blends with 5%, 10%, 15%, and 20% ethanol/diesel ratio blends [16]. They found that the brake specific fuel consumption and brake thermal efficiency increased with the ethanol content for the overall operating conditions.

Diesel blended with ethanol can reduce soot emissions [17, 18], improve combustion efficiency [19, 20] and

under certain conditions, can also decrease NO_x emissions [21, 22]. However, this decline has not achieved a breakthrough, and NO_x emissions are still at high levels [23, 24]. In addition, because ethanol and diesel oil is difficult to stabilize, additives or solvents are required to make diesel/ethanol blends fuel stable miscible [25]. However, diesel/ethanol blends have not yet gone into commercial applications or large scale production.

In recent years, butanol fermented from sugar, starch, or lignin converted from crops has drawn significant attention [26, 27]. Szwaja et al [28] studied the combustion characteristics of n-butanol with an SI engine and found that the optimal ignition timing and antiknock properties of butanol are similar to gasoline with an octane number of approximately 87. Gu et al. [29] researched the effects of EGR on the emissions from n-butanol/gasoline blends in a gasoline engine. Studies show that when using an n-butanol/gasoline mixture, the HC, CO and NO_x emissions can be reduced. EGR helps to reduce NO_x emissions when using n-butanol/gasoline blends. Rakopoulos et al. [30] studied the combustion properties of n-butanol/diesel blends in a diesel engine. Their study found that mixing butanol into the blend can effectively reduce soot emissions and slightly decrease NO_x emissions. Yao et al. [31] studied the effect of n-butanol as a diesel additive on engine performance and emissions for a heavy duty diesel engine. They found that while the NO_x emissions are roughly the same, n-butanol can effectively improve the soot and CO emissions. Doğan et al. [32] studied the effects of n-butanol/diesel blends on engine performance and emissions and found that as the volume percentage of butanol increases, the soot, CO and NO_x emissions decrease. However, the HC emissions increase. Rakopoulos et al. [33] studied the effects of n-butanol/diesel blends on engine performance and emissions in a high-speed diesel engine. Their results showed that as the volume percentage of n-butanol increased, the soot and CO emissions decreased gradually and that NO_x emissions decreased slightly. However, HC emissions increased with the volume percentage of butanol.

Besides the exploration of new clean renewable alternative fuels and to optimize combustion efficiency and emissions limits for diesel traditional compression ignition engines, different modes of combustion have also been

researched. In recent years, many scholars have performed studies on the homogeneous charge compression ignition (HCCI) combustion mode. These studies have produced a series of achievements, but for the existing technical conditions and control system, the full load range and variable operating conditions for HCCI combustion are still very difficult to control [34, 35]. To achieve high efficiency and low emissions for different operating conditions, reactivity controlled compression ignition (RCCI) combustion mode is proposed [36, 37]. The internal combustion is well controlled by high EGR ratios and two stage fuel injection strategies [38, 39]. Recently, Lu et al proposed a dual fuel sequential compression combustion (DFSC) mode [40, 41]. DFSC introduces a well-mixed, lean fuel/air mixture into the cylinder by injecting a high cetane number fuel at the intake port, followed by the direct injection of a high-octane number fuel near top dead center (TDC).

In the DFSC mode, the port injected fuel, which has a high cetane value, burns first and combusts in the HCCI low temperature combustion mode. Meanwhile, the combustion of high cetane value fuels provides a suitable temperature and releases a large number of active radicals, which is beneficial to the combustion of directly injected fuels. In addition, the lower cetane value of direct injected fuel results in a delayed burn, which is beneficial to the mixing of the directly injected fuel. Furthermore, a direct injection fuel gasification process may lead to temperature drops in the cylinder, which effectively inhibits the generation of NO_x.

Based on these reasons and considering the alcohol fuel properties, which generally include a high latent heat, high fuel oxygen content, and a low cetane value, an experimental study was conducted on the combustion and emissions characteristics of DFSC in a single-cylinder engine applying port injection of n-heptane combined with in-cylinder direct injection of ethanol, n-butanol and n-amyl alcohol, respectively. The combustion characteristics and emission properties are analyzed in detail.

2 Experimental setup

2.1 Engine and instrumentation

The experiments were performed on a modified single-cylinder, direct-injection, 4-stroke, water-cooled, naturally aspirated diesel engine, as shown in Fig. 1. The main engine specifications are listed in Table 1. A port fuel injection system was employed to supply n-heptane into the intake manifold to form the homogeneous charge. The injector for this system was mounted approximately 0.45 m upstream from the intake valve, with an injection pressure of 5 bar and port fuel injection timing of 340°CA BTDC. A direct injector (DI) with a cone angle of 154° was used to inject different alcohols directly into the cylinder. The DI timing can be held constant by maintaining the fuel supply advance angle at 25°CA BTDC.

The in-cylinder gas pressure was measured using a pressure transducer (Kistler mode 6125B). The charge output from this transducer was converted to an amplified voltage using an amplifier (Kistler mode 5015A) and recorded at a resolution of 0.5°CA. The exhaust gas components of the CO, UHC (unburned hydrocarbon), and NO_x emissions were measured by a gas analyzer (AVL Digas 4000). The smoke opacity was measured by a smoke meter (AVL Dismoke 4000). The measured parameters and their accuracy are summarized in Table 2. For all data presented, 0°CA is defined as TDC (top dead center) for the compression stroke. In consideration of the accuracy of g/kw-hr is highly relied on the accurate measurement of inlet air mass. For single cylinder engine, uneven intake flow leads to difficult accurate measurement of inlet air mass. In this work, ppm is used to discuss the gas emissions.

2.2 Test fuels

To compare the ignition and combustion characteristics of the DFSC fueled with n-heptane/alcohols fuels, high purity n-heptane, ethanol, n-butanol and n-amyl alcohol are used. The properties of these fuels are listed in Table 3.

2.3 Experimental procedure

Before the tests start, the engine is warmed up until the lubrication oil and coolant fluid temperatures are 85°C.

The engine speed is set to 1500 rpm. The test cylinder is fired in the HCCI mode with port injected n-heptane to heat the combustion chamber and exhaust. Then, the direct injector is used to inject high octane fuel (in this study, the high octane fuels are ethanol, n-butanol and n-amyl alcohol). Measurements are only taken after the CO₂ content in the exhaust flow has stabilized. In this experiment, the intake pressure is maintained at 1 atm. The intake temperature is controlled to be approximately 25°C. The in-cylinder gas pressure was averaged over 100 consecutive cycles for each operating point.

3 Definition of the combustion parameters

To investigate the combustion and emissions characteristics of DFSC fueled with n-heptane/alcohols fuels, several basic combustion parameters are defined. The ignition timing of the DFSC is defined as the crank angle position at which 10% of the fuel is burned, which is also referred to as CA₁₀. CA₅₀, which is another key parameter used to assess the combustion phasing, is defined as the crank angle position at which 50% of the fuel is burned. The combustion burn duration is defined as the crank angle interval between 10% and 90% of the mass fraction burned and overall lower heating values (LHVs) per-cycle, which are the sum of the energy of the premixed and directly injected fuels. In this paper, the premixed ratio (RP) is defined as the ratio of the energy of the premixed fuel Q_{PI} to the total energy Q . The premixed ratio can be calculated using the following formula:

$$RP = \frac{Q_{PI}}{Q} = \frac{m_{PI} \times H_{u,PI}}{m_{PI} \times H_{u,PI} + m_{DI} \times H_{u,DI}},$$

where m_{PI} and m_{DI} represent the mass consumption rates of the premixed and directly injected fuels, respectively. The variables $H_{u,PI}$ and $H_{u,DI}$ are the lower heating values of the premixed and directly injected fuels, respectively.

4. Experimental results and discussion

4.1 Effects of premixed ratio on DFSC fueled with n-heptane/alcohols

In DFSC mode, the in-cylinder combustible mixture is organized by port injection of high cetane number fuel combined with direct injection of high octane number fuel. For this condition, the premixed ratio directly determines the components and concentration stratification of the in-cylinder mixture, which has a significant impact on the combustion and emissions during the DFSC mode. Therefore, in this part, six premixed ratios (0.38, 0.46, 0.52, 0.63, 0.71 and 0.82) are selected and the overall LHVs per-cycle is maintained at 1.01 kJ/cycle. The effects of the premixed ratio on the combustion and emission properties of n-heptane/alcohols fuels in DFSC mode were investigated.

Figure 2 shows the effects of the premixed ratio on the in-cylinder gas pressure and heat release rate curves for the DFSC mode using n-heptane/ethanol and n-heptane/n-butanol as fuels, respectively. From Fig. 2, it can be seen that under the DFSC combustion mode, the heat release rate curve exhibits a three-stage combustion progress, which includes low and high temperature reactions of n-heptane and the combustion of the directly injected alcohols. As the premixed ratio increases, the heat release rates for the low and high temperature reactions increase, while the peak value of the heat release rate for the directly injected fuel decreases.

From a further comparison of the heat release rate curves, it can be found that when the RP is 0.46 and 0.52, the second stage of heat release for n-heptane/n-butanol occurs slightly earlier than for n-heptane/ethanol. This may due to the fact that ethanol's latent heat is higher than that of n-butanol. After the ethanol directly injected into the cylinder, the vaporized ethanol leads to a greater in-cylinder temperature decrease than for n-butanol. The combustion of n-heptane is similar to the HCCI mode, which is sensitive to variations in the in-cylinder temperature. In addition, the cetane number of ethanol is lower than that of n-butanol, the in-cylinder reactivity is lower when directly injected of ethanol. Meanwhile, due to lower LHV of ethanol, more ethanol should be injected into cylinder. The injection duration of ethanol would be longer which may lead ethanol difficult fully mixed with in-cylinder air. Finally these reasons lead to a slightly earlier heat release for the n-heptane/n-butanol than for the n-heptane/ethanol. When the premixed ratio is further increased to 0.63 and 0.71, the quantity of the directly injected fuel is seen to decrease. The

heat absorption by gasification decrease and the quantity of port injected n-heptane increases. The reaction activity of the in-cylinder n-heptane/air mixture is enhanced. The combustion is barely affected by the heat absorption from the direct injection fuel gasification. This causes when in high premixed ratio, the first and second stages of heat release of ethanol and butanol exhibit almost no difference.

Figure 3 shows the effects of the premixed ratio on the in-cylinder P_{max} and T_{max} of DFSC mode for different directly injected fuels. As seen in Fig. 3, as the RP increases, the maximum in-cylinder pressure and the maximum average temperature continuously increase. This is mainly because with increases to the premixed ratio, the combustion phase of the n-heptane is advanced and the concentration of n-heptane increases. More heat is released in the compression stroke, which increases the pressure and the cylinder temperature.

Figure 4 shows the relationship between the maximum pressure rise rates and the premixed ratio for different directly injected alcohols. With the increase in RP, the maximum cylinder pressure rise rate increases gradually. This is mainly because as the RP increases, the concentration of the port injected n-heptane increases. A higher concentration of n-heptane during compression leads to an increase in the maximum cylinder pressure rise rate. In addition, as the RP increases, the combustion phase is advanced. The peak value for the heat release rate curve is closer to TDC, which may also lead to an increase in the maximum pressure rise rate.

When the premixed ratio is less than 0.7 and for roughly the same RP, the difference in the maximum pressure rise rates between different directly injected fuels is not obviously. This may be due to the maximum pressure rise rate being achieved near TDC. For this condition, combustion is mainly achieved during the high temperature reaction of the port injected n-heptane. The pressure rise rate for these crank angles are mainly determined by the combustion of n-heptane, whereas the combustion of directly injected fuels have limited effects on the maximum pressure rise rate, which finally leads to a slight difference in the maximum pressure rise rate between different the directly injected fuels.

It is worth noting that the maximum pressure rise rate in the cylinder significantly increases when the premixed ratio is greater than 0.7. This is because in-cylinder severe burning is caused by the high concentration of n-heptane. For this condition, the in-cylinder pressure undulates violently, which leads to an irregular change in the maximum pressure rise rate when the premixed ratio is greater than 0.7 for the different directly injected fuels.

Figure 5 shows the effects of the premixed ratio on the emissions of DFSC mode for the different directly injected fuels. As seen in Fig. 5 (a), when the premixed ratio increases, the CO emissions first increase slightly and then decrease. The CO emissions are mainly determined by the degree of uniformity of the combustible mixture and the in-cylinder temperature. With the RP increases from 0.38 to 0.46, the combustion phase of the directly injected fuel advances which is not advantageous for the mixing of fuel and air. This leads to an increase in CO emissions. With further increases to the premixed ratio, the in-cylinder temperature increases which helps to oxidize the CO, resulting in CO emissions decreasing. Figure 5 (b) shows an increase in the premixed ratio and the HC emissions increase gradually. This is primarily because as the RP increases, the accumulation of mixed gas in the cylindrical boundary layer increases. Thus, HC emissions tend to increase.

In Fig. 5 (c), the NO_x emissions first increase and then decrease as the RP increases. The main reason for this is that as the premixed ratio increases, the in-cylinder combustion temperature gradually increases and part of the directly injected fuel will participate in the high temperature reaction for n-heptane combustion, which causes this part of the fuel to burn in a manner similar to diffusion combustion. For this condition, the combustion of a non-uniform mixture may lead to the formation of a local high temperature zone. This finally leads to an increase in the NO_x emissions. With further increases to the premixed ratio, the quantity of directly injected fuel is continuously reduced. This may inhibit the non-uniform combustion of directly injected fuel, which causes the combustion mode to gradually convert to the HCCI mode and inhibits NO_x formation.

As seen in Fig. 5 (d), the increase in the premixed ratio causes soot emissions to decrease rapidly when using

n-heptane/n-butanol and n-heptane/n-amyl alcohol. The main reason for this is that as premixed ratio increases, the quantity of direct fuel injection decreases, meaning more fuel is injected into the cylinder using port injection. The fuel experience better mixing with the air and become more homogenous, resulting in a decrease in soot emissions. Additionally, when ethanol is directly injected, smoke emissions are maintained at a lower level. This is mainly due to the higher molecular weight of oxygen and the shorter carbon chains associated with the ethanol, which can inhibit the formation of soot precursors, lowering soot emissions than that of n-butanol and n-amyl alcohol. It is worth noting that for the range of premixed ratios greater than 0.35, soot emissions are maintained at a low level.

4.2 Effects of the overall LHVs per-cycle on the DFSC fueled using n-heptane/alcohols fuels

The overall LHVs per-cycle are defined as the sum of LHVs per-cycle for the port and directly injected fuels. To a certain extent, the overall LHVs per-cycle provide a characterization of the engine load. To further investigate the combustion and emissions from the DFSC mode with direct injection of ethanol, n-butanol and n-amyl alcohol, respectively, under different loads, the premixed ratio is kept at 0.52 and six overall LHVs (0.67, 0.80, 1.01, 1.12, 1.28 and 1.42 kJ/cycle) are selected pre-cycle. The effects of the overall LHVs per-cycle on the combustion and emission properties for different port injected alcohol-like fuels in the DFSC mode are discussed in detail.

Figure 6 shows the effects of the overall LHVs per-cycle on the in-cylinder gas pressure and the heat release rate of the DFSC mode for different directly injected fuels. From Fig. 6, it can be seen that as the overall LHVs per-cycle increase, the in-cylinder pressure curves continue to increase and the in-cylinder pressure curves for the n-heptane/n-butanol are higher than that of n-heptane/ethanol. However, this difference decreases as the overall LHVs per-cycle increase. This is mainly due to the higher concentration of n-heptane enhancing the high temperature reaction.

It is interesting to see that as overall LHVs per-cycle increase, the intervals between the combustion of the directly injected fuel and the high temperature reactions for the n-heptane increase. Some reasons for this may be that

as the port injected n-heptane increases, the low-temperature and high-temperature reactions for the n-heptane are significantly improved as the overall LHVs per-cycle increase. Thus, the in-cylinder reaction activity is enhanced. Meanwhile, the quantity of the directly injected fuel also increases for the same RP. The high latent heat of the direct injected fuels reduces the in-cylinder gas temperature, which leads to an increase in the ignition delay for the direct injected fuels. For these two reasons, the combustion phase of the third reaction remains fairly constant when the overall LHVs per-cycle increases. However, the combustion phase for the first two stages is advanced because of a higher quantity of port injected n-heptane per-cycle. Consequently, the intervals between the combustions of the directly injected fuels and the high temperature reactions of the n-heptane are increased.

Figure 7 shows the effects of the overall LHVs per-cycle on the in-cylinder Pmax and Tmax for the DFSC mode for different directly injected fuels using the same premixed ratio. There is an increase in the overall LHVs per-cycle and the in-cylinder Pmax and Tmax increases. This trend is attributed to more fuel being injected into the cylinder per-cycle, especially for n-heptane, which leads to the advancement of the combustion phase and promotes in-cylinder combustion.

Figure 8 shows the effects of the LHVs per-cycle on the maximum pressure rise rate for the DFSC combustion mode. When the overall LHVs per-cycle are below 1.2 kJ/cycle, the maximum pressure rise rate for the DFSC combustion mode remains at a low level. However, when the overall LHVs per-cycle are greater than the 1.2 kJ/cycle, the maximum pressure rise rate increases sharply. This is because as the overall LHVs per-cycle increase, the heat release rate for port injected n-heptane increases and the concentration of n-heptane results in severe burning, which can lead to a higher maximum pressure rise rate. Comparing Figs. 4 and 8, it can be found that the in-cylinder maximum pressure rise rate always increases with the injection quantity for port fueled n-heptane. This trend shows that the in-cylinder maximum pressure rise rate is mainly controlled by the high temperature reaction of n-heptane, meaning that an appropriate high temperature reaction rate for n-heptane and the combustion phase can optimize the

242 maximum pressure rise rate.

243 Combining Figs. 7 and 8, it can be seen that no matter what the overall LHVs are, the differences between the
244 in-cylinder Pmax or the maximum pressure rise rates for different alcohol-based fuels are not obvious. The reason for
245 this is that Pmax and the maximum pressure rise rate occurs during the high temperature reaction for n-heptane,
246 which exhibits an HCCI-like combustion, resulting in the in-cylinder pressure rising rapidly. Otherwise, the alcohol
247 fuels directly injected into the cylinder mainly burn after TDC. At this time, the piston is already moving downward,
248 meaning that the in-cylinder pressure and the rise rate cannot be very high.

249 Figure 9 shows the effects of the overall LHVs per-cycle on the combustion phase and the burn duration for the
250 DFSC mode. As the overall LHVs per-cycle increase, CA10 and CA50 advance at first and then roughly remain
251 constant. However, the trends for CA90 are complicated. The advance in CA10 occur because as the overall LHVs
252 increase, the concentration of n-heptane increases. The in-cylinder reaction activity is enhanced, which may shorten
253 the ignition delay. However, more fuel charges need to burn, which may delay the CA10. Therefore, CA10 advances
254 at first and then remains constant. CA50 exhibits a similar trend as CA10. When the overall LHVs is lower, with an
255 increases in the quantity of PI n-heptane, the peak value of heat release rate occurs earlier, at this time, the in-cylinder
256 reaction activity is low and the high temperature flame of n-heptane has smaller influence on DI alcohol fuels, so that
257 CA50 is not sensitive to the change of the overall LHVs. When the overall LHVs are higher, the advance of CA10
258 can promote CA50. Besides, the temperature of n-heptane high temperature combustion increases, more DI alcohol
259 fuels burn at this stage, CA50 can be substantial advanced.

260 The trend for CA90 with the overall LHVs is more complex. First, as the overall LHVs increase, the quantity of
261 the directly injected fuel increases, leading to a longer reaction time. This is the main reason for the delay in the
262 occurrence of CA90. Second, the reaction temperature of the in-cylinder mixture increases as the overall LHVs
263 increase, which can promote combustion. Third, the gasification of the directly injected fuel also has a great influence

on CA90. In addition, CA10 also has great effect on CA90.

Figure 10 shows the effects of the overall LHVs per-cycle on the emissions for the DFSC combustion mode. In Fig. 10 (a), as the overall LHVs increase, the CO emissions decrease. As the overall LHVs per-cycle increase, the in-cylinder temperature increases, which benefit the CO conversion to CO₂. In Fig. 10 (b), HC emissions remain at a stable level. This is mainly due to an increase in the concentration of the premixed n-heptane as a greater fuel charge is trapped in the crevices. However, the increase in the in-cylinder temperature can promote the oxidation of HC. These two reasons lead to the HC emissions remaining at a stable level.

In Fig. 10 (c), NO_x emissions increase initially and then decrease. At first, the increase in NO_x is due to the rise of the in-cylinder temperature. Then, as the overall LHVs increase, more alcohol fuel is injected into the cylinder, which reduces the reaction temperature due to its high latent heat. In Fig. 10 (d), the soot emissions increase gradually when the overall LHVs per-cycle increases. This is due to the increase in the quantity of the DI fuel and the proportion of the non-homogeneous charge increasing gradually. However, it is worth mentioning that soot emissions always remain at low levels. In addition, soot emissions for ethanol are always slightly lower than that from n-butanol and n-amyl alcohol.

4.3 The combined effects of the premixed ratio and the overall LHVs per-cycle on the combustion and emissions for the DFSC mode

For further analysis of the premixed ratio and the LHVs per-cycle on the combustion and emissions of DFSC mode fueled with n-heptane/alcohols, the experimental conditions for the premixed ratio are increased from 0.4 to 0.8 and LHVs per-cycle are increased to between 0.4 to 1.2 kJ/cycle for direct injection of ethanol and n-butanol, respectively. These are chosen for the plotting of contour maps for the main parameters of combustion and emissions. Direct injection of ethanol is shown in red and n-butanol is shown in blue.

Fig. 11 (a) and (b) show the combined effects of the premixed ratio and the LHVs per-cycle on CA10 and CA50 for the DFSC mode, respectively. From Fig. 11 (a), it can be found that when the premixed ratio is constant, if the LHVs per-cycle are increased, the CA10 continues to advance. When the LHVs per-cycle are constant, the CA10 increases with the increases of premixed ratio. The difference between directly injected ethanol and n-butanol is smaller when the LHVs per-cycle are higher than that for the 1.6 kJ/cycle. The CA10 of n-butanol was significantly more advanced than that of ethanol when the LHVs per-cycle are lower than 1.6 kJ/cycle. Fig. 11 (b) shows the combined effects of the premixed ratio and LHVs per-cycle on the CA50 for the DFSC mode. The difference is insignificant when both the LHVs per-cycle and the premixed ratio are high. For other conditions, the CA50 of the directly injected n-butanol is advanced compared to that of ethanol.

Fig. 11 (c) shows the combined effects of the premixed ratio and the LHVs per-cycle on the combustion duration for the DFSC mode. When the LHVs per-cycle is fixed, the combustion duration can be gradually shortened by increasing the premixed ratio for n-heptane/ethanol. However, for directly injected n-butanol, when the LHVs per-cycle are higher than 0.6, the combustion duration is gradually shortened as the premixed ratio is increased. When the LHVs per-cycle are lower than 0.6, combustion duration changes slightly as the premixed ratio is increased. From a comparison of directly injected ethanol and n-butanol, it can be found that when the LHVs per-cycle are higher than 0.6 and the premixed ratio is higher than 0.65, there is not an obvious difference between using ethanol and n-butanol. However, for other conditions, the combustion duration of the ethanol is longer than that of n-butanol.

Figure 12 shows the combined effects of premixed ratio and LHVs per-cycle on the main gas emissions for the DFSC mode. Fig. 12 shows that for different direct injection of alcohol-like fuels, there are several common features in the gas emissions for the DFSC mode. The CO and NO_x emissions decrease with the increase in the LHVs per-cycle. Soot emissions reach a maximum at a low premixed ratio and high LHVs per-cycle. At approximately the

same loads, the CO, NO_x and soot emissions from the ethanol are a little higher than that for the n-butanol. The higher molecular oxygen and the higher latent heat of the ethanol help to inhibit the emissions. In addition, it is worth noting that for direct injection of ethanol and butanol, soot and NO_x emissions are maintained at a low level.

Fig. 13 shows the combined effects of the premixed ratio and the LHVs per-cycle on the indicated mean effective pressure (IMEP) and indicated thermal efficiency of DFSC mode. In figure 13 (a), it can be found that under current operating conditions, the loads can be reached in the range from 1 to 6 bar. When in high LHVs per-cycle and with the increase of premixed ratio, the engine loads increase first and then decrease. When in high LHVs per-cycle, with the increase of premixed ratio, the engine loads are decreased gradually. This is mainly determined by the reaction characteristics of direct injected alcohols and in-cylinder reaction activity.

In figure 13 (b), for port injection of ethanol and n-butanol, when in low premixed ratio and in medium and low LHVs per-cycle, the indicated thermal efficiency would be higher. Under this condition, the amount of port injected n-heptane is suitable to control the combustion phase. It can also be found in figure 13 (b) that in most loads, when directly injected of n-butanol, the indicated thermal efficiency would be higher than that of ethanol. The main reason is that for n-butanol, which has higher cetane number and lower latent heat makes the CA50 earlier and the combustion duration shorter. So that the indicated thermal efficiency would be higher.

5. Conclusions

An experimental study was conducted on the combustion and emissions characteristics of DFSC in a single-cylinder engine using port injection of n-heptane combined with in-cylinder direct injection of ethanol, n-butanol and n-amyl alcohol. The combustion parameters, main emissions and the heat release features are analyzed in detail. From this research, the following conclusions are reached:

- (1) The heat release may be divided into three stages, which are the low temperature and high temperature reaction of n-heptane and the directly injected fuel combustion stage.
- (2) The amount of port injected n-heptane plays a key role in P_{max} , T_{max} and the maximum pressure rise rate for the DFSC mode.
- (3) When in the high overall LHVs per-cycle, the CO emissions decrease with the increase of the premixed ratio. By contrast, the CO emissions increase with the increase of premixed ratio when the overall LHVs per-cycle is in medium and low levels.
- (4) NO_x and soot emissions are kept at low levels for the current experimental conditions. The soot emissions reach a maximum value for a low premixed ratio and high overall LHVs per-cycle. In particular, the higher latent heat of vaporization, the lower cetane values and the shorter carbon chains associated with the ethanol lead to lower NO_x and soot emissions than observed for n-butanol and n-amyl alcohol.
- (5) Compared to ethanol, under the current conditions, when directly injection of n-butanol, the indicated thermal efficiency would be higher. This is mainly due to the higher cetane number, lower latent heat of n-butanol which lead to more suitable combustion phase. It is worth noting that current experiments are not optimized. For optimized direct injection pressure and timing, the indicated thermal efficiency of ethanol would be higher.

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Enabling Dual Fuel Sequential Combustion using Port Fuel Injection of High Reactivity Fuel Combined with Direct Injection of Low Reactivity Fuels

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Table 1. Engine specifications

Displaced volume	0.792 L	Exhaust Valve Open	114°CA ATDC
Stroke	105 mm	Exhaust Valve Close	348°CA BTDC
Bore	98 mm	Inlet Valve Open	344°CA ATDC
Compression ratio	18.5:1	Inlet Valve Close	128°CA BTDC
Number of Valves	4	Needle Open Pressure	24 Mpa

Table 2. Accuracies of the measurements

Measured parameters	Unit	Measurement range	Accuracy
Engine speed	r/min	0-10,000	±1
Engine torque	Nm	0-200	±0.1
Cylinder pressure	MPa	0-25	±0.0005
Intake pressure	kPa	0-500	±0.5
Intake air temperature	°C	0-200	±0.1
Exhaust gas temperature	°C	0-800	±1
Coolant temperature	°C	0-150	±0.1
Lubricant oil temperature	°C	0-150	±0.1
CO emissions	%	0-4.0	±0.01
HC emissions	ppm	0-10,000	±1
NOx emissions	ppm	0-4000	±1
Soot emissions	m ⁻¹	0-99.99	±1%
Fuel consumption	kg/h	0-10	±1%

Table 3. Fuel properties

	N-heptane	Ethanol	N-butanol	N-amyl alcohol
Molecular formula	n-C ₇ H ₁₆	C ₂ H ₆ O	n-C ₄ H ₁₀ O	n-C ₅ H ₁₂ O
Molecular weight	100.2	46	74.12	88
Density g/cm ³ @20°C	0.682	0.746	0.8098	0.8114
Purity %	>97.33	>99.9	>99.5	>99.5
Lower heat value (MJ/kg)	44.57	26.8	33.04	35.2
Latent heat (kJ/kg)	316	862	584	427
Research octane number	0	110	87	-
Cetane number	56	8	~25	-
Carbon content % (mass)	84	52	65	68
Hydrogen content % (mass)	16	13	13	14
Oxygen content % (mass)	0	35	22	18
Stoichiometric ratio	15.3	9	11.1	11.7

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Fig. 1 Schematic of the engine and instrumentation setup

Fig. 2 Effects of the premixed ratio on the in-cylinder gas pressure and the heat release rate curves for the DFSC mode with directly injected fuels

Fig. 3 Effects of the premixed ratio on the in-cylinder Pmax and Tmax for the DFSC mode using directly injected fuels

Fig. 4 Effects of the premixed ratio on the maximum pressure rise rate during the DFSC mode with directly injected fuels

Fig. 5 Effects of the premixed ratio on the emissions of DFSC mode for different directly injected fuels

Fig. 6 Effects of the overall LHVs per-cycle on the in-cylinder gas pressure and heat release rate curves for the DFSC mode for different directly injected fuels

Fig. 7 Effects of the overall LHVs per-cycle on the in-cylinder Pmax and Tmax for the DFSC mode

Fig. 8 Effects of the overall LHVs per-cycle on the maximum pressure rise rate for the DFSC combustion mode

Fig. 9 Effects of the overall LHVs per-cycle on the combustion phase

Fig. 10 Effects of overall LHVs per-cycle on the emissions of DFSC combustion mode

Fig. 11 Combined effects of the premixed ratio and the LHVs per-cycle on the combustion parameters of the DFSC

Fig. 12 Combined effects of the premixed ratio and the LHVs per-cycle on the main gas emissions for the DFSC

Fig. 13 Combined effects of the premixed ratio and the LHVs per-cycle on the IMEP and indicated thermal efficiency of DFSC mode

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(a) CA10 (°CA)

(b) CA50 (°CA)

(c) Combustion Duration (°CA)

Fig. 11 Combined effects of the premixed ratio and the LHV's per-cycle on the combustion parameters of the DFSC

Fig. 12 Combined effects of the premixed ratio and the LHV's per-cycle on the main gas emissions for the DFSC

(a) IMEP (bar)

(b) Indicated Thermal Efficiency (%)

Fig. 13 Combined effects of the premixed ratio and the LHV per-cycle on the IMEP and indicated thermal efficiency of DFSC mode