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# INFLUENCE OF FUEL SUPPLY TIMING AND MIXTURE PREPARATION ON THE CHARACTERISTICS OF STRATIFIED CHARGE COMPRESSION IGNITION COMBUSTION WITH N-HEPTANE FUEL

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***Stratified charge compression ignition (SCCI) combustion was investigated on a single-cylinder, four-stroke diesel engine with n-heptane. A premixed atmosphere was realized by the early injection at the intake port, while the rest of fuel was directly injected into the cylinder. Premixed ratio (defined as the ratio of fuel amount in port injection divided by the total fuel amount in the charge) and fuel supply timing of direct injection were altered to explore their influences on SCCI combustion, emission, and engine performance characteristics. The results show that increased premixed ratio and earlier fuel supply timing can remarkably enhance the peak cylinder temperature and pressure while shortening the main combustion duration. NOx is strongly dependent on fuel supply timing, whereas premixed ratio and equivalence ratio play a secondary role. CO can be effectively reduced by increasing premixed ratio and advancing fuel supply timing. HC rises with increasing premixed ratio but is insensitive to the change of fuel supply timing. Finally, indicated thermal efficiency (ITE) drops with retarded fuel supply timing, and the maximum indicated mean effective pressure (IMEP) can be achieved by optimizing the premixed ratio and fuel supply timing.***

***Keywords***: Emission; Fuel supply timing; n-heptane; Premixed ratio; Stratified charge compression ignition



## INTRODUCTION

Homogeneous charge compression ignition (HCCI) is a promising combustion mode by which NOx and PM emission from a diesel engine can be simultaneously reduced because of its uniform mixture, lean burn, and the reduced combustion temperature. However, unlike the conventional SI and CI engines, HCCI engine lacks a direct control strategy for the auto-ignition timing and combustion rate, and the

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chemical kinetics dominant mechanism in HCCI tends to cause severe ‘‘knock combustion’’ at heavy load and combustion instability at light load. Therefore, the operation range of HCCI would only be restricted within a narrow region (Haraldsson, 2004; Stanglmaier & Roberts, 1999).

Stratified charge compression ignition (SCCI) might provide a novel combustion strategy to precisely control the auto-ignition timing. Different from the uniform charge in HCCI combustion, the temperature gradient or fuel distribution stratification is formed in SCCI combustion, and the local fuel-rich or hot place will serve as the ignition trigger. The effect of thermal stratification on smoothing heat release was investigated by Sjöberg et al. (2004). By reducing the coolant temperature and increasing the in-cylinder air swirl, in-cylinder thermal stratification was created, and knocking intensity could be reduced with a penalty of power loss. The effort of controlling ignition timing and combustion rate by stratified fuel distribution was also performed by many researchers (Aroonsrisopon et al., 2004; Berntsson & Denbratt, 2007; Lee et al., 2007). Lee et al. tried SCCI combustion on a gasoline engine, where intake temperature, negative valve overlap, and compression ratio were adjusted together to generate the stratified mixture. The stratified mixture was found to be able to operate over a wider load range than the homogeneous mixture (Lee et al., 2007). Some other researchers adopted double-stage fuel injection strategy in order to form the stratified mixture. The first stage injection, usually port injection, could obtain sufficient mixing time to form the uniform mixture like HCCI, while the second stage injection at top dead center (TDC) would generate a local fuel-rich mixture to facilitate the auto-ignition. These studies indicated that by this double-stage injection strategy, the ignition timing could be influenced; therefore, the combustion stability was improved and the operation range could be extended. (Aroonsrisopon et al., 2004; Berntsson & Denbratt, 2007).

However, for most of the study mentioned above, high-octane gasoline-like fuel was chosen for testing, and the potential of diesel-like fuel for SCCI combustion has not been widely investigated. Some researchers studied the dual-fuel SCCI combustion, where the diesel-like fuel was partly applied. Inagaki et al. (2006) investigated dual-fuel stratification combustion mode in a single-cylinder diesel engine. Iso-octane was injected into port and diesel was injected directly into the cylinder. The load range could be extended to 1.2 MPa IMEP with little soot and NOx production. In addition, Kim and Chang studied the effect of various premixed fuel on the dual-fuel SCCI combustion mode with diesel fuel in the direct injection (Chang et al., 2003; Kim & Chang, 2006; Kim et al., 2004). As a result, the single-stage combustion was found when gasoline was employed as the premixed fuel in SCCI combustion, rather than the three-stage combustion with diesel or n-heptane as the premixed fuel (Kim & Chang, 2006).

In this study, n-heptane was selected to be injected into both the intake port and the combustion chamber. The proportion of fuel injected at each stage was adjusted to alter the charge stratification extent. Moreover, fuel supply timing of direct injection varied within the range from 10°CA BTDC to 26°CA BTDC in order to optimize this SCCI combustion strategy.

## EXPERIMENTAL DESCRIPTION

### Experimental Apparatus

The experiment was conducted on a single-cylinder, direct-injection and four-stroke naturally aspirated diesel engine. In order to inject a portion of fuel into the port, several modifications were made to the intake port system, and an electrically controlled injection system was mounted at a location 0.35 meters upstream to the intake port. The rest of fuel was directly injected into the cylinder via the nozzle in prototype engine. The engine specification can be found in Table 1. The other experimental apparatus comprises an eddy current dynamometer and a coupled control cabinet. A schematic diagram of the test bench is shown in Figure 1.

The cylinder pressure was measured by a pressure transducer (Kistler Model 6125A), then the output charge signal, which could be amplified using a charge amplifier (Kistler model 5015A). Finally, the pressure data were recorded in a high-speed memory at a 0.25°CA resolution. NOx, CO, and HC emission were measured with a gas analyzer (AVL Digas 4000), the measurement range and accuracy for which are shown in Table 2.

### Experimental Procedure

In the experiment, the engine was operated at a fixed speed 1800 rpm. The proportion of premixed fuel could be controlled by the electronic injection system from 0 to 100%, which means that the combustion mode varies from conventional DI engine to fully HCCI engine. In this paper, the index used to indicate the homogenous degree of the mixture is named premixed ratio $r_p$, and is calculated by Eq. (1), where $m_p$ and $m_d$ represent the mass consumption rate of premixed and directly injected fuel, respectively.

$$r_p = \frac{m_p}{m_p + m_d} \tag{1}$$

In order to investigate the influence of fuel supply timing of direct injection on SCCI combustion characteristics, the fuel supply timing was adjusted through mechanically altering the relative position of fuel supply pump with respect to the camshaft. The fuel supply timing was varied from 10°CA BTDC to 26°CA BTDC in this study. The intake gas temperature was held within $25 \pm 2$°C, and coolant water temperature

**Table 1** Engine specifications

| | |
|---|---|
| Cylinder number | 1 |
| Bore × stroke | 98 mm × 105 mm |
| Displacement | 792 cm$^3$ |
| Combustion chamber | $\omega$ type |
| Compression ratio | 18.5 |
| Injection nozzle number and diameter | 5 × 0.24 mm |
| Nozzle spray angle | 154° |
| Needle open pressure | 19 Mpa |

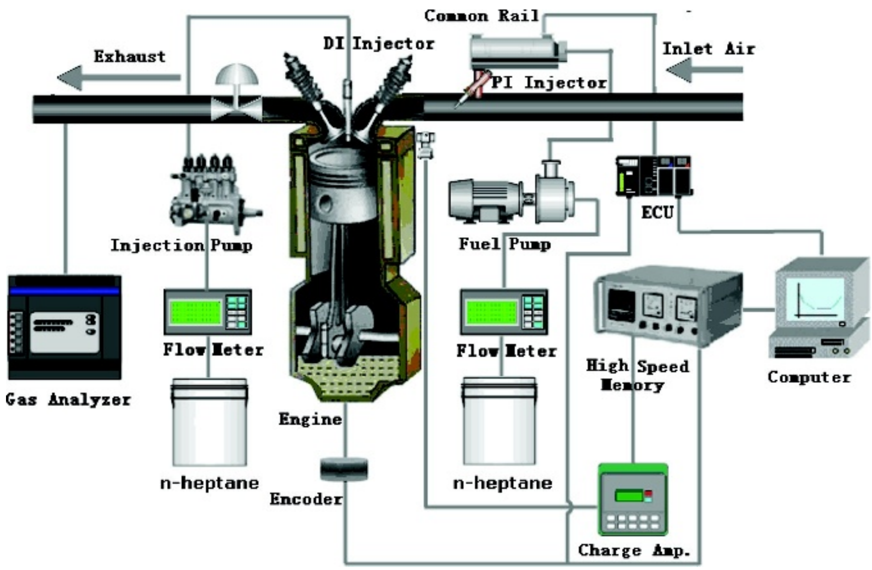



**Figure 1** The schematic diagram of the experimental bench.

**Table 2** Emission measurement range and accuracy

| Gaseous emission | Measurement range | Relative accuracy |
|---|---|---|
| CO (vol%) | 0–10 | ±5% |
| HC (ppm) | 0–20,000 | ±5% |
| NOx (ppm) | 0–5000 | ±5% |

**Table 3** Test conditions

| | |
|---|---|
| Engine speed | 1800 rpm |
| Advance of direct injector opening | 10–26°CA BTDC |
| Intake air temperature | $25 \pm 2$°C |
| Coolant temperature | 80°C |
| EGR | None |

**Table 4** Properties of n-heptane

| | |
|---|---|
| Chemical formula | n-$C_7H_{16}$ |
| Molar mass | 100.16 g/mol |
| Density | 0.688 g/$cm^3$ at 20°C |
| Boiling point | 98°C |
| Lower heat value | 44.5 MJ/kg |
| Cetane number | 56 |
| Concentration | 97% |

remained exactly at 80°C. The test condition is listed in Table 3, and the physical and chemical properties of n-heptane are shown in Table 4.

## Heat Release Analysis

The heat release rate, bulk gas temperature, and IMEP in this study are calculated from an in-house zero-dimensional model. The experimentally acquired cylinder pressure is first filtered by weighted-averaging adjacent points with a specified filter width. A weighted coefficient vector is assigned to make sure that the polynomial curve fits the raw data.

The heat release algorithm used in this model is based on Heywood (1988), and a crank angle based differential form of the first law is derived by applying the energy conservation equation to the engine cylinder. In this differential equation, only in-cylinder internal energy, cycle work, and heat loss between combustion gases and cylinder wall are taken into account, as shown in Eq. (2), where $\frac{dQ_{ch}}{d\theta}$ is the chemical energy release rate from the fuel, $mC_v\frac{dT}{d\theta}$ is the term to evaluate the rate of change in the in-cylinder internal energy, $P\frac{dV}{d\theta}$ is used to describe the cycle work rate, and $\frac{dQ_{loss}}{d\theta}$ is the heat loss rate. The heat and mass transfer as a result of blow-by and crevice flow are not considered for simplicity. Therefore, constant mass in cylinder is assumed from IVC and EVO. The calculation of heat loss rate between combustion gas and cylinder wall is based on Newton's cooling law, and the heat loss coefficient is obtained by empirical correlations developed by Woschni (1967).

$$\frac{dQ_{ch}}{d\theta} = mC_v\frac{dT}{d\theta} + P\frac{dV}{d\theta} - \frac{dQ_{loss}}{d\theta} \tag{2}$$

The gases during the compression, combustion, and expansion processes are assumed to possess the ideal-gas characteristics, so the instantaneous bulk gas temperature can be calculated through the ideal gas state equation, based on the measured cylinder pressure and calculated cylinder volume. At each crank angle, the in-cylinder gases species are estimated by assuming complete combustion of the burned gases. The constant volume-specific heat for each individual species is estimated as the function of temperature, $c_v = a_0 + a_1T + a_2T^2 + a_3T^3$. Then, the specific heat for the entire mixture in cylinder can be obtained as the mass-weighted average of the specific heat for each species.

## Definition of Combustion Parameters

As SCCI combustion concept is a compromise between the HCCI and standard DI combustions, three-stage combustion could be found in diesel SCCI combustion (Simescu et al., 2003), as shown in Figure 2. In order to investigate the influences of premixed ratio and fuel supply timing on the SCCI combustion phase, several basic combustion parameters are illustrated in Figure 2. First, the start timing of low temperature reaction (LTR) and that of high-temperature reaction (HTR) are defined: the start timing of LTR is when the value of HRR exceeds 0.5 J/°CA; while for HTR, the start timing is when the derivative of HRR exceeds 0.5 J/(°CA)$^2$. Additionally, the main combustion duration, which involves the

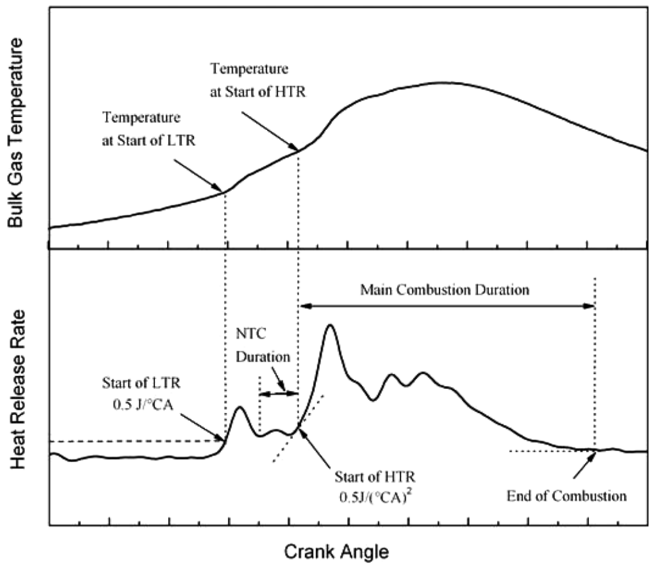



**Figure 2** Definitions of combustion parameters.



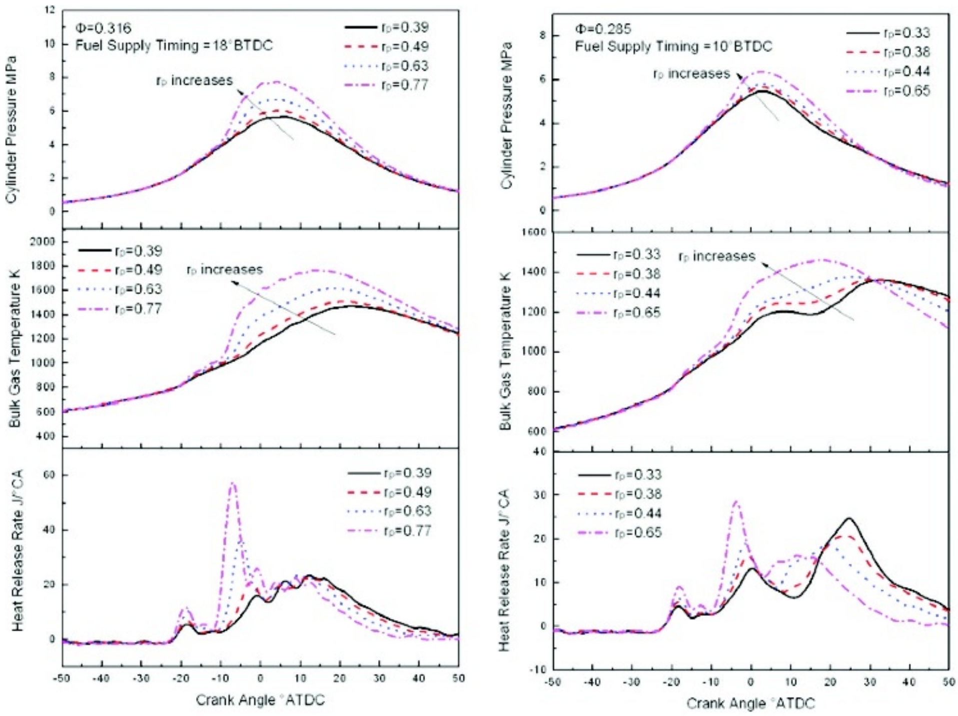
HTR and diffusive combustions, is defined as the crank angle between the start timing of HTR and the end of diffusive combustion. The crank angle, which corresponds to 90% of the accumulated HRR, is considered as the end of combustion.

## RESULTS AND DISCUSSION

### Combustion Analysis

Figure 3 shows the influence of premixed ratio on HRR, in-cylinder pressure, and bulk gas temperature at the fixed equivalence ratio and fuel supply timing. The general features of SCCI combustion shown in Figures 3a and 3b consist of rising cylinder pressure and bulk gas temperature with the higher premixed ratio. These changes can be attributed to the combustion changes, which are illustrated by HRR sweeps. The increased premixed ratio does not only raise the maximum HRR of HTR but also advances the start timing of HTR, which is generally considered as the ignition timing of main combustion. Meanwhile, the diffusive combustion region also approaches HTR with the increase of premixed ratio; thus, the main combustion is close to constant-volume burn and achieves a higher peak pressure and temperature.

However, the start timing of LTR is hardly influenced by premixed ratio, as shown in Figure 4a. The low-temperature reactions of different premixed ratios in the condition of Figure 3b almost all begin at 21°CA BTDC. The only impact of increased premixed ratio lies in the elevated peak HRR magnitude of LTR. Nevertheless, HTR start timing advances as premixed ratio increases, as shown in Figure 4b. Figure 5 may well explain these results. As is shown, although the premixed ratio and fuel supply timing vary greatly, the temperature at the start timing of LTR and HTR merely changes within a certain range. The LTR start temperature




(a) Φ=0.316

Fuel Supply Timing=18°CA BTDC

(b) Φ=0.285

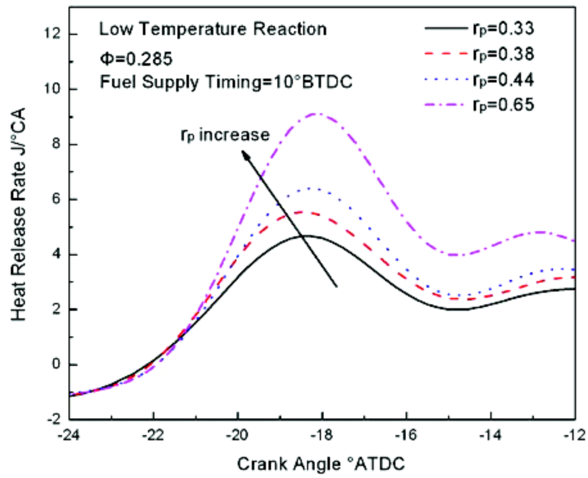
Fuel Supply Timing =10°CA BTDC

**Figure 3** Effect of premixed ratio on cylinder pressure, bulk gas temperature, and heat release rate with fixed equivalence ratio and fuel supply timing.

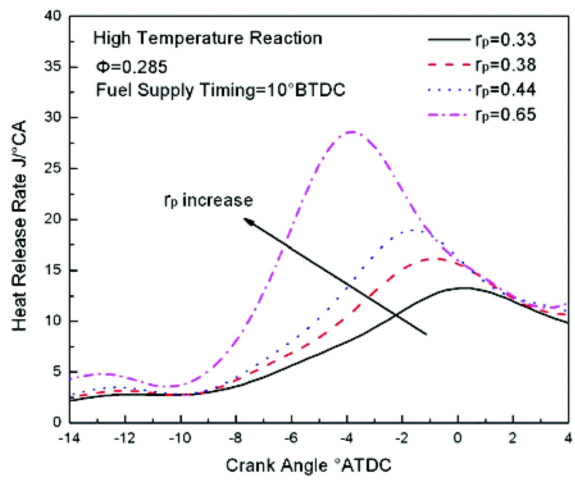


changes within 750 K to 800 K, while HTR start temperature varies within 950 K to 1020 K. Similar results were also found in previous studies by Inagaki et al. (2006) and Lü et al. (2005). According to Lü's study, the decomposition temperature of ketohydroperoxide species is the key factor to initialize LTR. Because this decomposition temperature is located within a specified range, the first stage combustion starts as long the mixture temperature reaches this critical temperature by compression. Therefore, premixed ratio plays a small role on the start timing of LTR. On the contrary, the premixed ratio has an obvious influence on the ignition timing of main combustion. Because higher premixed ratio can produce more heat release in LTR, the in-cylinder temperature tends to reach the start temperature of HTR earlier and the main combustion phase advances.

Figure 6 shows the effect of fuel supply timing on the SCCI combustion characteristics at the fixed equivalence ratio and premixed ratio. As is shown in the figure, when the fuel supply timing is retarded, the peak in-cylinder pressure and temperature are decreased. With regard to the HRR, the sweep demonstrates more obvious three-stage combustion, as the direct injected fuel is inducted into the cylinder later. Correspondingly, the peak HRR of diffusive combustion occurs at a retarded time




Influence of premixed ratio on the low temperature heat release rate




Influence of premixed ratio on high temperature heat release rate

**Figure 4** Effect of premixed ratio on low-temperature and high-temperature heat release rates with fixed equivalence ratio and fuel supply timing. (Enlargement from Figure 3b.)



and the magnitude decreases. LTR and HTR stages are influenced slightly by fuel supply timing. In addition, as shown in Figure 7, the main combustion duration is prolonged with retarded fuel supply timing, especially at low premixed ratios, which might help avoid the occurrence of knock and therefore extend load range. However, as the premixed ratio increases, the effect of fuel supply timing on the main combustion duration is weakened.

Figure 8 illustrates the trends of peak cylinder pressure and temperature versus fuel supply timing, equivalence ratio, and premixed ratio. At the fixed equivalence ratio, as shown in Figure 8a, the peak cylinder pressure and temperature both fall with retarded fuel supply timing. In addition, increased premixed ratio produces

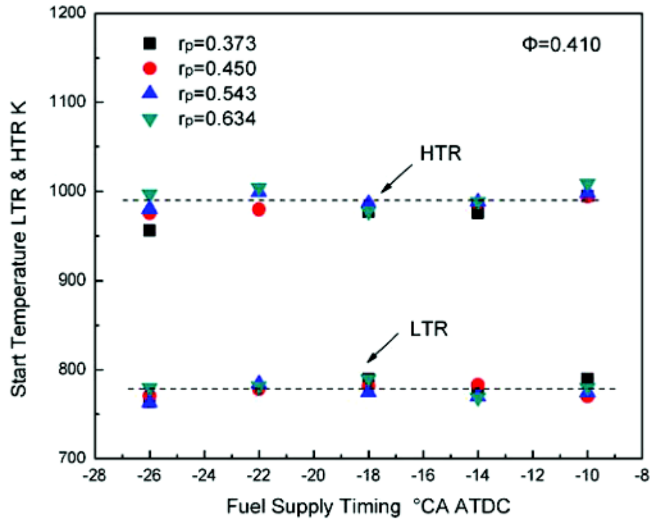
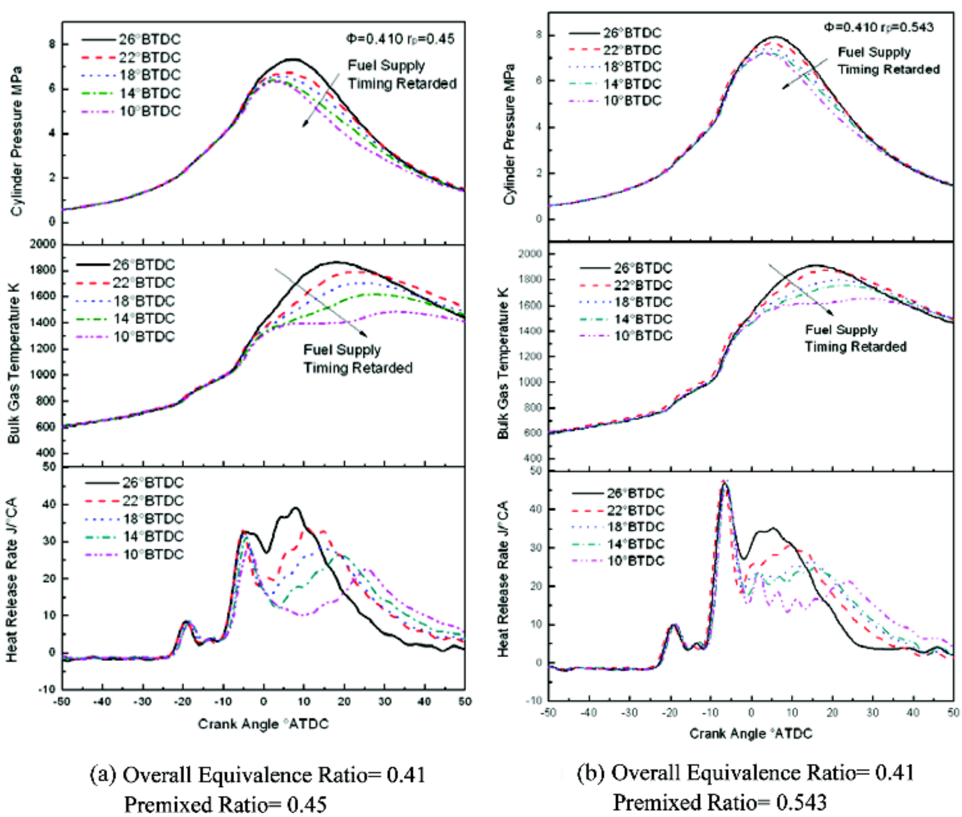



**Figure 5** Starting temperatures of LTR and HTR with various premixed ratios and fuel supply timing.




(a) Overall Equivalence Ratio= 0.41
Premixed Ratio= 0.45

(b) Overall Equivalence Ratio= 0.41
Premixed Ratio= 0.543

**Figure 6** Effect of fuel supply timing on cylinder pressure, bulk gas temperature, and heat release rate with fixed equivalence ratio and premixed ratio.

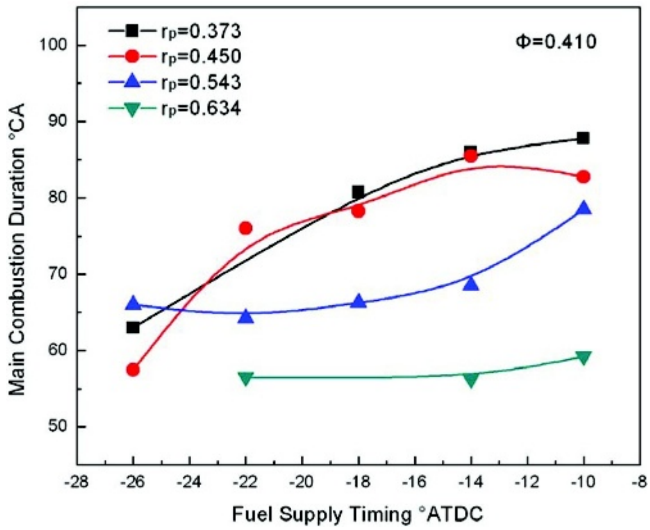
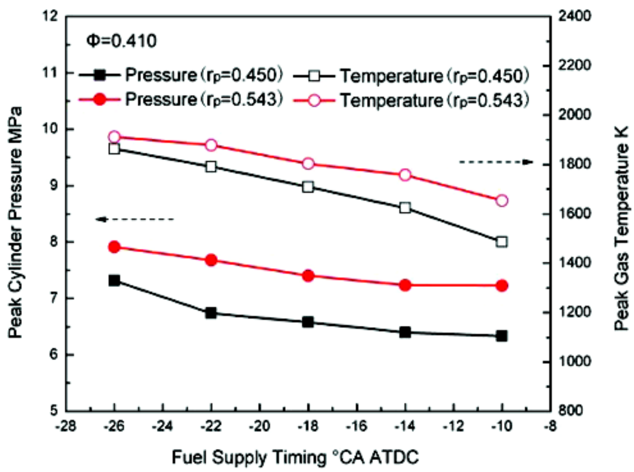



**Figure 7** Main combustion duration as a function of fuel supply timing at various premixed ratios.

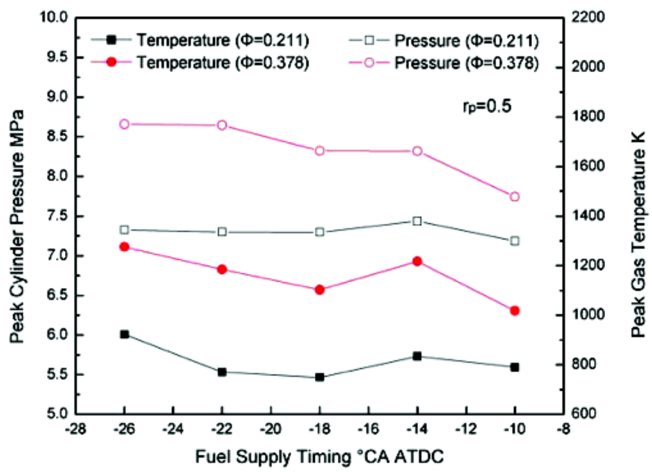
higher peak cylinder pressure and temperature, particularly at retarded fuel supply timing. Figure 8b shows the peak cylinder pressure and temperature at the fixed premixed ratio. At higher equivalence ratio, the peak cylinder pressure and temperature drop rapidly with retarded fuel supply timing; on the contrary, low equivalence ratio could generate a flat curve with retarded fuel supply timing, possibly because the fuel lean environment in this case, regardless of the fuel supply timing change. In our colleagues' study on the same test bench (Ji et al., 2009), where iso-octane was chosen as the direct injection fuel, fuel supply timing was found not to play a monotonic effect on the peak in-cylinder temperature, because as the fuel supply timing was further advanced to 35°CA BTDC, the peak in-cylinder temperature would start decreasing. To better illustrate the fuel effect on SCCI combustion, even further advanced fuel supply timing should be investigated for the n-heptane SCCI combustion in the future.

## Emission Analysis

In this section, the influence of several operating parameters, including fuel supply timing, premixed ratio, and equivalence ratio on SCCI emission characteristics are analyzed. Figure 9 shows the influence of fuel supply timing on emissions at various premixed ratios. As is shown in Figure 9a, fuel supply timing plays a dominant role on NOx emission. As the supply timing is retarded, NOx emission can be finally reduced to a minimum level, which is lower than 40 ppm regardless of premixed ratio. It is well known that NOx is formed through the chemical reaction between oxygen and nitrogen molecules or atoms in the high-temperature burned gases. As the fuel supply timing is retarded, the diffusive combustion mainly occurs late in the expansion stroke; thus, the peak cylinder temperature falls and NOx emission reduces.




(a) Different premixed ratios




(b) Different equivalence ratios

**Figure 8** Peak cylinder temperature and pressure as a function of fuel supply timing.

The premixed ratio plays a secondary role on NOx formation, although this role is a little complicated. When fuel supply timing is advanced prior to 18°CA BTDC, NOx emission shows no apparent relationship with premixed ratio. This scenario might be attributed to the following reason: the excessively advanced fuel supply timing is more likely to superimpose diffusive combustion onto HTR combustion, thus forming a single major heat release region. The approximate constant volume combustion and rapid heat release tend to produce similar peak combustion temperature, as shown in Figure 8a. Therefore, the premixed ratio does not influence NOx emission evidently with excessively advanced fuel supply timing. However, as the fuel supply timing is retarded after 18°CA BTDC, NOx emission increases with premixed ratio. At this time, diffusive combustion is separated from HTR, and peak

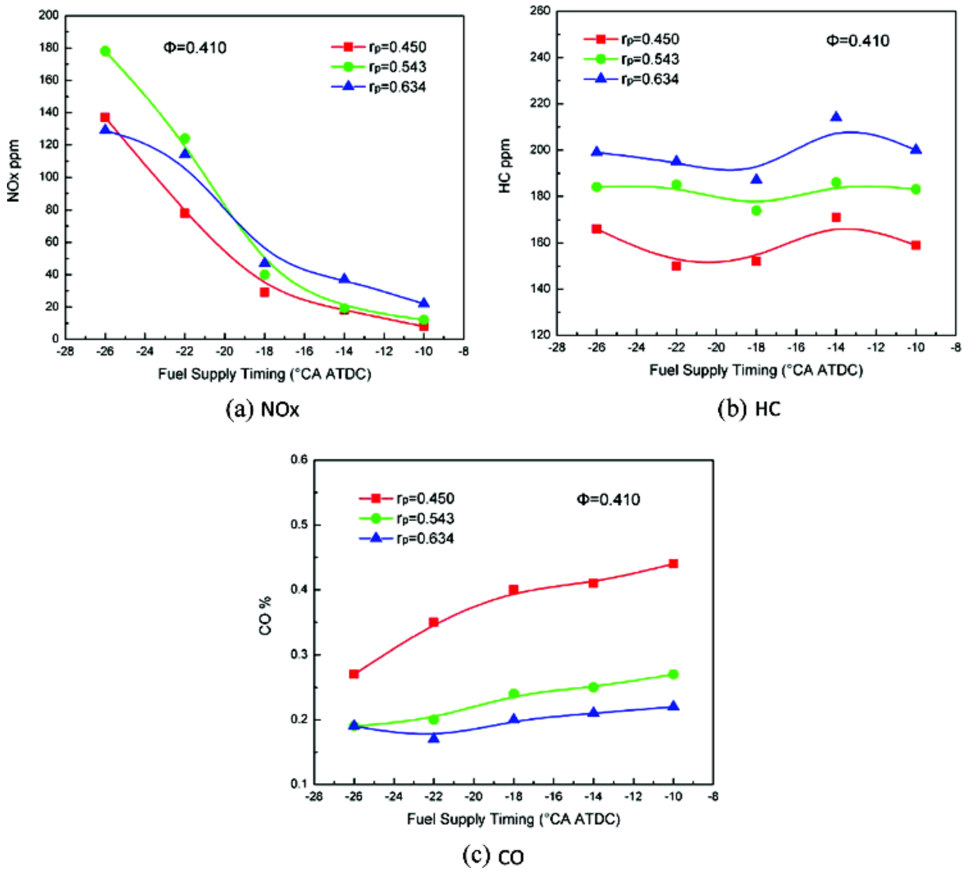



**Figure 9** Emission characteristics at various fuel supply timing and premixed ratios but fixed equivalence ratio.

cylinder temperature, which is determined by HTR, increases steadily with premixed ratio; hence, NOx emission rises with increasing premixed ratio at retarded fuel supply timing.

The elevated HC and CO emission in the HCCI engine, in contrast to the conventional DI engine, is one obstacle for practical application. Moreover, increased HC and CO emission also means deteriorated fuel consumption efficiency. HC emission has a complicated mechanism because several factors can simultaneously contribute to the HC generation. In SI engine, HC generally originates from flame quenching area in combustion chamber, crevice storage, and the absorption/desorption effect from oil layers and chamber deposits, whereas HC generated in CI engine is principally due to undermixing and overleaning effect of the injected fuel (Heywood, 1988). Because the SCCI combustion incorporates the characteristics of both SI and CI engines, the HC emission of SCCI engine is also expected to demonstrate a combination of their mechanisms. However, the result in this study (see Figure 9b) shows that in SCCI combustion, premixed ratio plays a more important role than the fuel supply timing in HC emission, which means the HC emission generated from the SI engine mechanism overwhelms that from the DI engine mechanism. As is shown in Figure 9b, HC emission increases with premixed ratio because of more premixed charge trapped in the dead zone of the combustion chamber. On the other hand, generally, HC emission is insensitive to the change of fuel supply timing, considering the measuring error of the gas analyzer (±5% relative accuracy). This might be explained by the magnitude difference of HC emission between SI and CI combustion. Lu et al. (2008) found that the HC emission from DICI engine was one order of magnitude smaller than that from homogeneous charge combustion. In this study, the fuel supply timing mainly affects the phasing of diffusive combustion, but does not have any obvious effect on the premixed combustion. Therefore, the change in HC formation in the diffusive combustion might be covered by the much higher HC generation in the premixed combustion.

Engine-out CO emission is mainly influenced by fuel/air ratio and combustion temperature, as rich fuel/air mixture is likely to cause incomplete combustion and produce more CO emission, while decreased combustion temperature may freeze the CO oxidation reaction. Because both conventional DI engine and HCCI engine are normally operated in the fuel lean condition, the variance of CO emission can be mainly explained by the combustion temperature change. As is seen in Figure 9c, CO emission increases steadily as the fuel supply timing is retarded or premixed ratio drops. This result is consistent with the peak cylinder temperature change shown in Figure 8a, because either retarded fuel supply timing or decreased premixed fuel leads to a lower peak temperature.

Figure 10 illustrates the emission characteristics as a function of fuel supply timing at different equivalence ratios. NOx emission gradually decreases with retarded fuel supply timing, and at the supply angle of 10°CA BTDC, NOx emission of different equivalence ratios nearly achieve the same low level. However, at early fuel supply timing, high equivalence ratio can enhance NOx emission remarkably, probably due to the soaring cylinder temperature at high load.

At the equivalence ratio of 0.213, the HC emission achieved the minimum value at the fuel supply timing of 14°CA BTDC. However, when equivalence ratio increases, the cylinder temperature is enhanced rapidly, and the unburned fuel remaining in the dead zone is more likely to be oxidized, so the effect of fuel supply timing is weakened.

As with the HC emission, CO emission is reduced dramatically as the equivalence ratio increases, because the increased in-cylinder temperature. In addition, it is interesting that the CO emission of the equivalence ratio 0.213 is lower than that of the equivalence ratio 0.310, because the former condition is expected to result in lower combustion temperature and the CO oxidation should slow down.

## Performance Analysis

The indicated thermal efficiency (ITE) as a function of fuel supply timing is shown in Figure 11. Generally, the ITE exhibits a descending trend as the fuel supply timing retards. It is notable that when fuel supply timing is further retarded after 14°CA BTDC, the ITE deteriorates more rapidly. In addition, when fuel supply timing varies between 22°CA BTDC and 14°CA BTDC, the ITE changes little with premixed ratio. At late fuel supply timing (10°CA BTDC), the operation with a

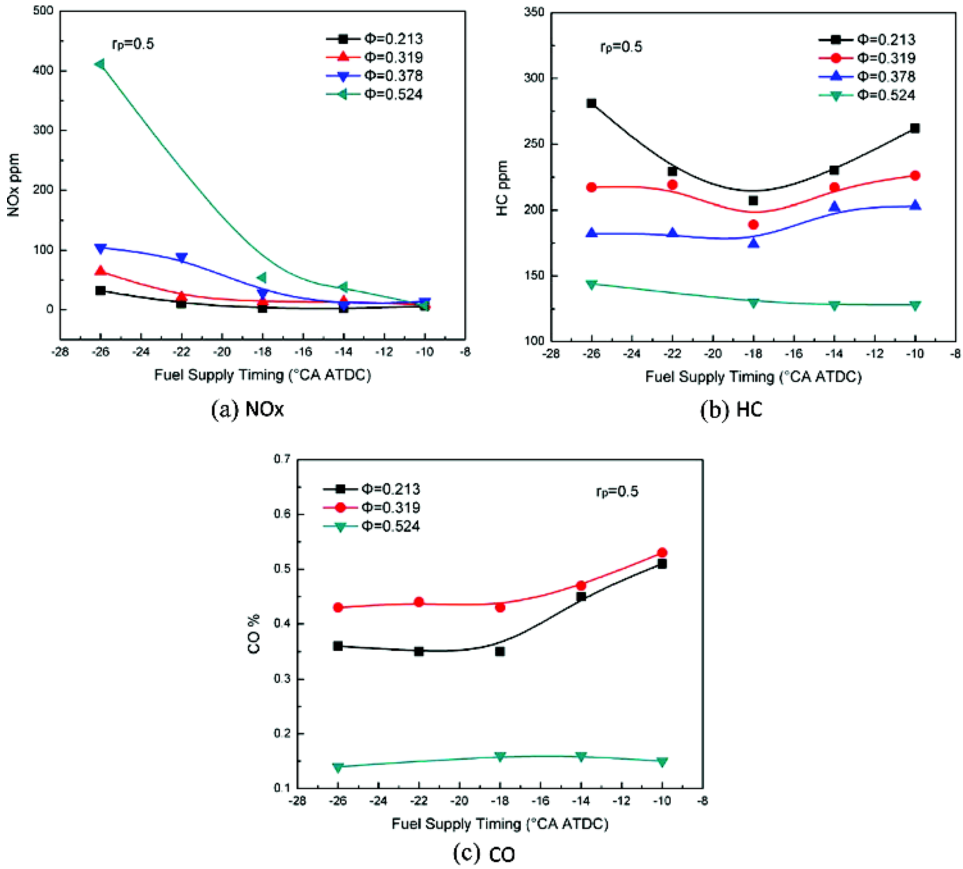



**Figure 10** Emission characteristics at various fuel supply timing and equivalence ratios but fixed premixed ratio.

higher premixed ratio possesses a superior thermal efficiency because the low-quality diffusive combustion in expansion stroke is limited. On the contrary, when fuel supply timing is excessively advanced (26°CA BTDC), lower premixed ratio produces elevated thermal efficiency, possibly because the proportion of premixed combustion is reduced and negative work loss in compression stroke falls.

Figure 12 shows the indicated mean effective pressure (IMEP) trend with fuel supply timing at various premixed ratios. It can be seen that IMEP exhibits a descending trend with the retarded fuel supply timing, for all the premixed ratios except 0.634. At the premixed ratio of 0.634, when fuel supply timing is as early as 26°CA BTDC, the IMEP drops rapidly, possibly owing to the overly advanced combustion phasing. On the other hand, throughout the range of fuel supply timing investigated in this study, the IMEP achieves the maximum value when premixed ratio is 0.543. This might be attributed to the tradeoff between negative work loss in compression stroke and late combustion loss in expansion stroke. In this sense, through altering fuel supply timing and premixed ratio, load range can be extended to a higher level than HCCI and DI engine.

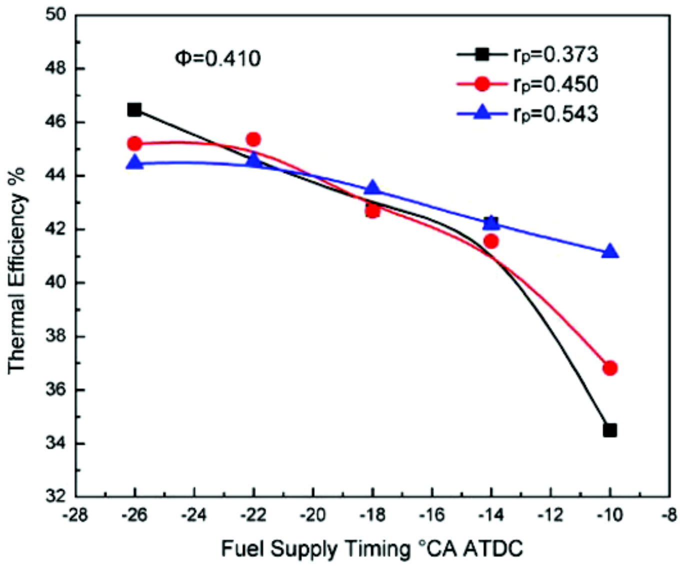



**Figure 11** Indicated thermal efficiency as a function of fuel supply timing at different premixed ratios but fixed equivalence ratio.

Figure 13 depicts the relationship between the peak pressure rising rate and fuel supply timing at different premixed ratios. The result indicates that the peak pressure rising rate changes very little with fuel supply timing. It is reasonable because the peak magnitude of pressure rising rate basically occurs during HTR stage, but is not influenced evidently by diffusive combustion. In contrast, increasing premixed ratio can raise peak pressure rising rate obviously, which may increase the possibility of knock combustion.

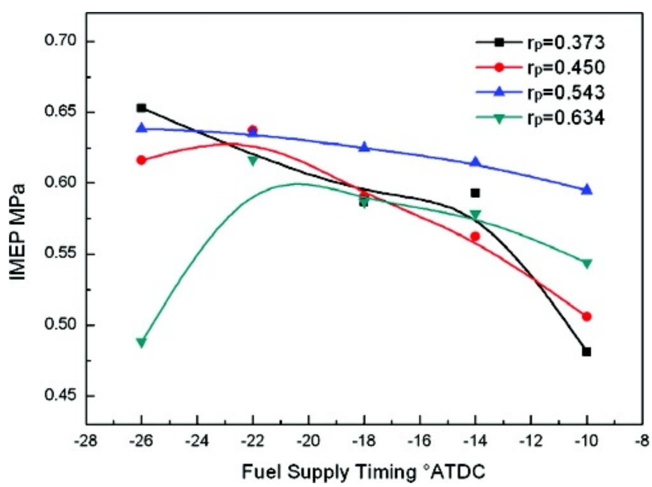



**Figure 12** IMEP versus various premixed ratios and fuel supply timing at fixed equivalence ratio.

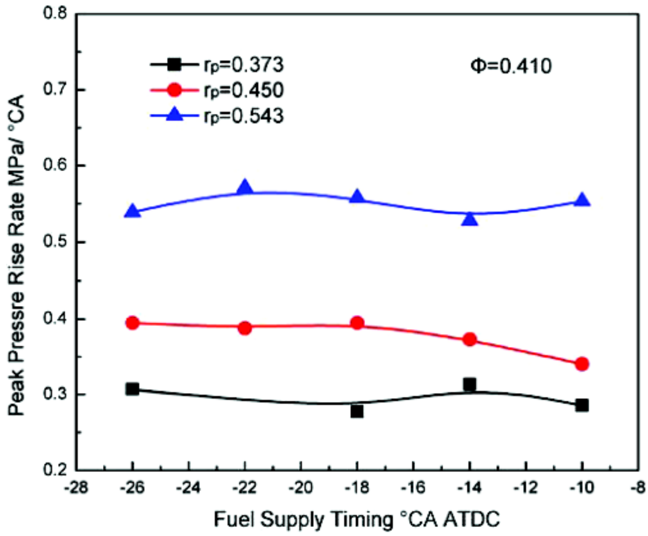



**Figure 13** Peak pressure rise rate as function of fuel supply timing at different premixed ratio but fixed equivalence ratio.



## CONCLUSION

Experiments were conducted on a modified single-cylinder diesel engine to explore the combustion, emission, and performance characteristics of SCCI combustion. Several experimental parameters, such as premixed ratio, fuel supply timing of direct injection, and equivalence ratio were adjusted in order to explore the combustion phase control. The main results concluded from this study are listed as following:

- Premixed ratio and fuel supply timing are influential factors on SCCI combustion. Increased premixed ratio and advanced fuel supply timing can both elevate the cylinder temperature and pressure while the diffusive combustion moves closer to the premixed combustion. The main combustion duration can be shortened by advancing fuel supply timing and increasing premixed ratio. The HRR curve exhibits obvious three-stage combustion as the fuel supply timing is sufficiently retarded.
- NOx emission has a strong dependency on the fuel supply timing. As fuel supply timing is retarded, NOx emission decreases dramatically regardless of equivalence ratio and premixed ratio. Equivalence ratio plays a secondary effect on NOx emission because NOx tends to increase at larger load, when combustion temperature is higher.
- The increased premixed ratio and equivalence ratio contribute to reduce the CO emission because of the higher cylinder temperature in these conditions. As fuel supply timing retards, CO emission shows an ascending trend because the cylinder temperature decreases and the CO oxidation reaction slows down. The CO emission shows little relationship with fuel supply timing at high equivalence ratio, when the in-cylinder temperature is sufficiently high.

- Equivalence ratio and premixed ratio pose an obvious influence on HC emission. In this study, the rising equivalence ratio would reduce the HC emission, owing to the increased cylinder temperature; on the other hand, higher premixed ratio can lead to the increasing HC emission, probably because more premixed charge is trapped in the dead zone of the combustion chamber. However, HC emission is insensitive to the change of fuel supply timing.
- Engine-indicated thermal efficiency decreases with the retard of fuel supply timing. Furthermore, the indicated thermal efficiency of lower premixed ratio condition tends to fall more rapidly as the fuel supply timing is excessively retarded.
- At the modest premixed ratio and fixed equivalence ratio investigated in this study, the IMEP shows a monotonic decrease as the fuel supply timing retards; on the other hand, an optimal premixed ratio exists to achieve the maximum IMEP, due to the tradeoff between negative work loss compression stroke and the late combustion loss in expansion stroke.

## ACKNOWLEDGMENT

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## NOMENCLATURE

| | |
|---|---|
| BTDC | before top dead center |
| CA | crank angle |
| CI | compression ignition |
| CO | carbon monoxide |
| DI | direct injection |
| HCCI | homogenous charge compression ignition |
| HRR | heat release rate |
| HTR | high-temperature reaction |
| IMEP | indicated mean effective pressure |
| ITE | indicated thermal efficiency |
| LTR | low-temperature reaction |
| NOx | nitric oxide |
| PM | particulate matter |
| $r_p$ | premixed ratio |
| SCCI | stratified charge compression ignition |
| SI | spark ignition |
| TDC | top dead center |
| UHC | unburned hydrocarbon |

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