Available online at www.sciencedirect.com

ScienceDirect

International Journal of Thermal Sciences 47 (2008) 1235–1242

International Journal of Thermal Sciences

www.elsevier.com/locate/ijts

An experimental study of HCCI-DI combustion and emissions in a diesel engine with dual fuel

Junjun Ma *, Xingcai Lü, Libin Ji, Zhen Huang

School of Mechanical & Power Engineering, Shanghai Jiaotong University, Shanghai, People's Republic of China

Received 23 March 2007; received in revised form 16 October 2007; accepted 16 October 2007

Available online 26 November 2007

Abstract

An experimental study of port injected n-heptane homogeneous charge compression ignition (HCCI) in combination with in-cylinder diesel fuel direct injection (DI) was conducted on a single cylinder diesel engine. By adjusting the quantities of premixed n-heptane as well as the direct injected diesel fuel, different premixed ratios were obtained over a wide range of load conditions while the test engine was kept at a constant speed. The effects of premixed ratio (r_p) and direct injection timing on HCCI-DI combustion characteristics and emissions were investigated. In order to evaluate the traits of HCCI-DI combustion, the conventional diesel and HCCI engine performances were presented to be compared with their results. Furthermore, special emphasis was put on the combined combustion process analysis. It can be found that the NOx emissions decreased dramatically with partial premixing and it exhibited a descending trend as a function of r_p increase when the r_p was lower than 0.3, but it showed a great tendency to increase when the premixed ratio was higher. The inherent trade-off of NOx and soot was not obvious since the soot emissions remained at the same level of the prototype diesel engine under the condition of lower r_p. Moreover, the influence of r_p on the CO and UHC emissions was assessed. The results also revealed that the HCCI-DI combustion could effectively improve the indicated thermal efficiency of diesel engine at low to medium loads.

Keywords: HCCI; n-Heptane; Combustion; Emission; Dual-fuel

1. Introduction

Since the environmental protection regulations and the need to reduce fuel consumption are getting more and more stringent, Homogeneous Charge Compression Ignition (HCCI), as a promising alternative combustion technology with high efficiency and lower NOx and particulate matter emissions, has been widely investigated over the recent years.

In some regards, HCCI combustion incorporates the advantages of both spark ignition (SI) engines and compression ignition direct injection (CIDI) engines. The lean homogeneous fuel/air mixture is essentially inducted into the cylinder without throttling losses and then compressed to autoignition which occurs simultaneously through the cylinder without discernable flame propagation. These features lead to very low NOx and PM emissions while maintaining high thermal efficiency.

However, regardless of the existing benefits, the control problems of ignition timing and burning rate are to be solved before the HCCI engines can be practically applied to commercial use. These two challenges are difficult to overcome firstly because HCCI is lack of ignition control mechanism like the spark and direct injection timing control. Secondly the HCCI combustion is dominated by the chemical kinetics based on the fuel properties therefore the occurrences of misfiring at low load and knocking at high load are usually noted which result in a limited operation range of HCCI engine.

Recently a lot of researches have been performed to investigate the potential control methods such as the inlet air heating [1,2], variable compression ratio (VCR) [3,4], variable valve actuation (VVA) [5,6] and EGR rates [7,8]. Moreover many studies also focused on the effects of different fuel physical and chemical properties, for instance the octane number and the cetane number, using the primary reference fuels and fuel addictives [9–12]. Within these attempts, attracting progresses have been reported, to some extent, about gaining control of

* Corresponding author. Tel.: +86 21 64078095; fax: +86 21 64078095.
E-mail address: junjun2323@sjtu.edu.cn (J. Ma).

1290-0729/$ – see front matter
doi:10.1016/j.ijthermalsci.2007.10.007

the HCCI combustion process. However due to the essence that there is no direct control method to determine the autoignition process, none of the achievements shows satisfactory results in thoroughly expending the HCCI engine operation range particularly at high loads.

Another concept to overcome the disadvantages of HCCI is to use HCCI / SI dual mode combustion [13,14]. Equipped with the VVA and spark ignition system, the HCCI/SI dual mode engine is able to operate in HCCI mode at low to medium loads and it can switch into SI mode to meet the large power output requirements. However the mode transition, especially from HCCI to SI, is not very stable and smooth so that more improvements would be needed on the control strategies to diminish the cycle-to-cycle variation [15–17]. Generally, this dual mode engine combines the HCCI and SI together to achieve the best performance.

Accordingly, the HCCI-DI compound combustion mode could be considered as a compromise between premixed HCCI and conventional CIDI. By adjusting the quantities of port injected fuel and direct injected fuel, various premixed ratios can be obtained. Several studies have revealed the advantages of similar combined combustion mode, Kim et al. investigated the effects of premixed gasoline fuel and direct injection timing on partial HCCI [18,19]. Suzuki et al. performed an experimental and computational study of the homogeneous charge intelligent multiple injection combustion system (HiMICS) [20]. Their results all showed the simultaneous reduction of NOx and soot emissions, at the expense of CO and UHC increase. Yet more understanding of the correlation between combustion characteristics and emissions of the compound HCCI-DI combustion is required. And experiments are needed to investigate the effect of premixed ratio on this combustion mode for all engine loads.

The objective of this study is to investigate the combustion and emission characteristics of HCCI-DI concept and the effects of premixed ratio and direct injection timing. Normal heptane was chosen as the premixed fuel because of its relatively low boiling point and excellent ignition ability. The commercial diesel fuel was injected near top dead center (TDC) as a complementation under the condition of misfire or knock to expend the engine operating range. The engine experiments were conducted over a load range from 0–100% while the engine speed was fixed at 1800 rpm. This work also examined the engine performance difference between conventional CIDI, HCCI-DI and the fully HCCI, as the premixed ratio varied from 0 to 1, and then provided suggestions for the optimum operation regime of HCCI-DI.

2. Experimental apparatus and procedure

The research engine was based on a single-cylinder, direct-injection and four-stroke naturally aspirated diesel engine. The main engine specifications are listed in Table 1. Fig. 1 demonstrates the schematic diagram of the test bench system. An electronically controlled port injection system was employed to inject n-heptane in the intake manifold at the location of approximately 0.35 meters upstream to the inlet port and the diesel fuel was directly injected into the cylinder near the top dead center with the mechanical injection pump. The experimental apparatus also included an eddy current dynamometer and the coupled control system.

Table 1
Main engine specifications

Bore × stroke	98 (mm) × 105 (mm)
Displacement	0.792 (L)
Aspiration	naturally aspirated
Combustion chamber	ω type
Compression ratio	18.5
Injection nozzle (direct injection)	5 × 0.24 (mm)
Nozzle spray angle	154°
Needle open pressure	19 (MPa)

Table 2
Test conditions

Engine speed	1800 rpm
Port injection timing	340 °C A BTDC
Advance of direct injector opening	7 °C A BTDC
Intake air temperature	20 °C
Coolant temperature	85 °C
Lubricant oil temperature	90 °C
EGR rate	0%

The cylinder pressure was measured with a pressure transducer (Kistler model 6125A). The charge output from this transducer was converted to amplified voltage using an amplifier (Kistler model 5015) and then was recorded at 0.25 °CA resolution with the sampling signals from the shaft encoder. According to the in-cylinder gas pressure averaged from 50 consecutive cycles for each operating point, the heat release rate can be calculated by zero-dimension combustion model. The exhaust gas composition CO, UHC, and NOx emissions were measured by gas analyzer (AVL 4000). Also the equivalence ratio Φ could be obtained from the analyzer. Smoke opacity was measured by a smoke meter (AVL 439).

The premixed ratio r_p in this paper is defined using the following equation. Where m_p and m_d indicate the mass consumption rate of premixed n-heptane and directly injected diesel fuel respectively, h_{up} and h_{ud} are the lower heating values of n-heptane and diesel fuel. So $r_p = 1.0$ is equivalent to fully HCCI and $r_p = 0$ means the conventional CIDI.

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

The test conditions are summarized in Table 2 and the fuel properties of diesel and n-heptane are provided in Table 3. For all data presented, 0 °CA is defined as the top dead center (TDC) at compression stroke. To ensure the repeatability and comparability of the measurements for operating conditions, the temperatures of intake air, oil and coolant water were held accurately stable during the experiments. The EGR were not applied in the experiment and engine speed was kept at 1800 rpm.

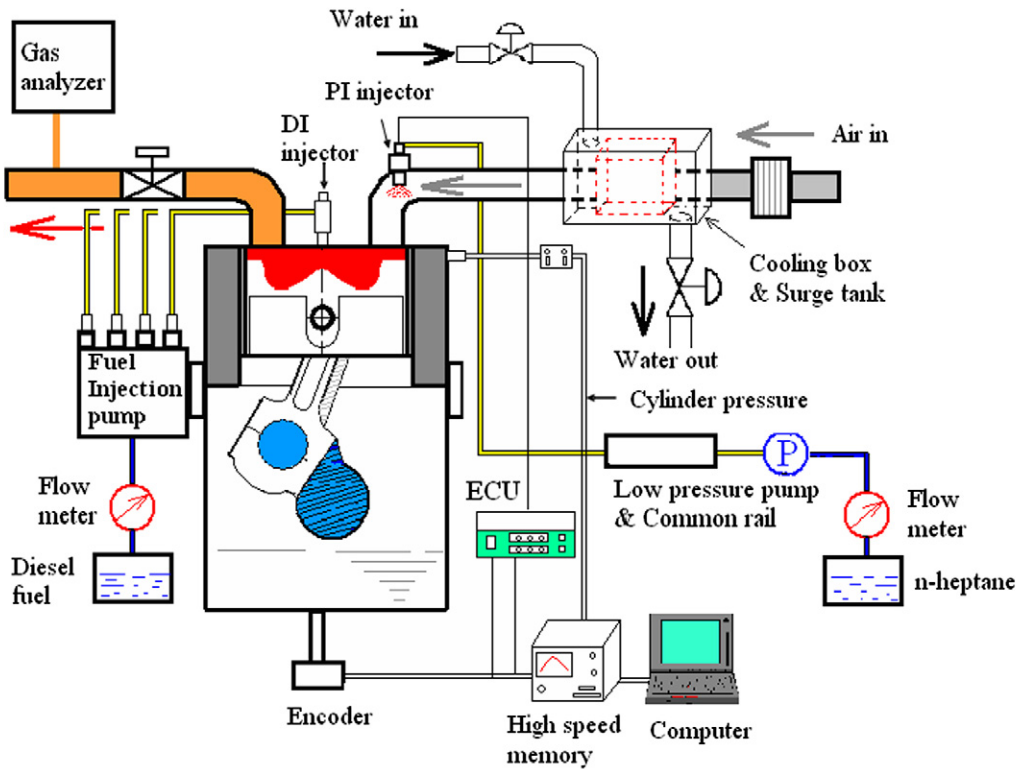


Fig. 1. The schematic diagram of the test bench system.

Table 3
Fuel properties of diesel and n-heptane

	Diesel	n-Heptane
Chemical formula	C10.8H18.7	n-C7H16
Molar mass (g/mol)	148.3	100.16
Density (g/cm^3 at 20 °C)	0.839	0.688
Boiling point (°C)	180–370	98
Lower heat value (MJ/kg)	42.9	44.5
Cetane number	51.7	56
Sulfur (mass ppm)	403	0

3. Results and discussion

3.1. Combustion characteristics of HCCI-DI

Fig. 2 shows the combustion characteristics of HCCI-DI with different premixed ratios. Many numerical and experimental results have verified that HCCI exhibits a two-stage combustion while the CIDI is mainly a diffusive combustion. In principle, the HCCI-DI may be a three-stage combustion consisting of cool flame region, the high temperature HCCI combustion region and the third diffusive combustion due to the directly injected diesel fuel. This kind of feature can be observed clearly in the heat release rate (HRR) curves of Fig. 2.

It is worth noting that the premixed ratio has little impact on the ignition timing of low temperature cool flame which occurs approximately at 25° BTDC, but the negative temperature coefficient (NTC) region of the chemical controlled intermediate temperature reactions shortens distinctly as the r_p increases. Consequently, the start of second stage combustion is found to be advanced and the maximum heat release rate of HCCI combustion phase increases when the premixed ratio is increased. Also the peak values of cylinder pressure and temperature rise in proportion to the increase of r_p. The ignition delay of directly injected diesel fuel is not much affected by r_p because the previous HCCI combustion raised the in-cylinder pressure and temperature,which resulted in a minimal diffusive ignition delay that was not evident to be observed.

The pressure rise rate $\mathrm{d}P/\mathrm{d}\phi$ is usually adopted as an index to describe the intensity of combustion roughness. In this paper, the “knock combustion” is defined as the maximum $\mathrm{d}P/\mathrm{d}\phi$ exceeds 1.2 MPa/°CA. Fig. 3 gives the profiles of maximum cylinder pressure $P_{\max}$ and the maximum pressure rise gradient under the same total fuel equivalence ratio with various premixed ratios. As it is mentioned above, $r_p = 0$ represents the CIDI operating condition.

It can be seen from Fig. 3 that in comparison with CIDI, the peak cylinder pressure of HCCI-DI is much higher and it increases monotonically with the premixed ratio. The maximum $\mathrm{d}P/\mathrm{d}\phi$ displays a slight decrease with the premixed ratio lower than 0.2 and it quickly rises when the premixed ratio is higher ($r_p \geqslant 0.2$). The reason is that when r_p is lower than 0.2, the premixed charge is very lean and there is no high temperature reaction of HCCI, therefore the $\mathrm{d}P/\mathrm{d}\phi_{\max}$ occurs during the diffusive combustion stage. Under the condition of higher premixed ratio, the hot ignition of HCCI starts simultaneously in the chamber so that the $\mathrm{d}P/\mathrm{d}\phi_{\max}$ changes to occur in this close to constant volume combustion stage.

Fig. 3 also reveals that with the directly injected diesel, the test engine run in HCCI-DI mode could eliminate the misfire

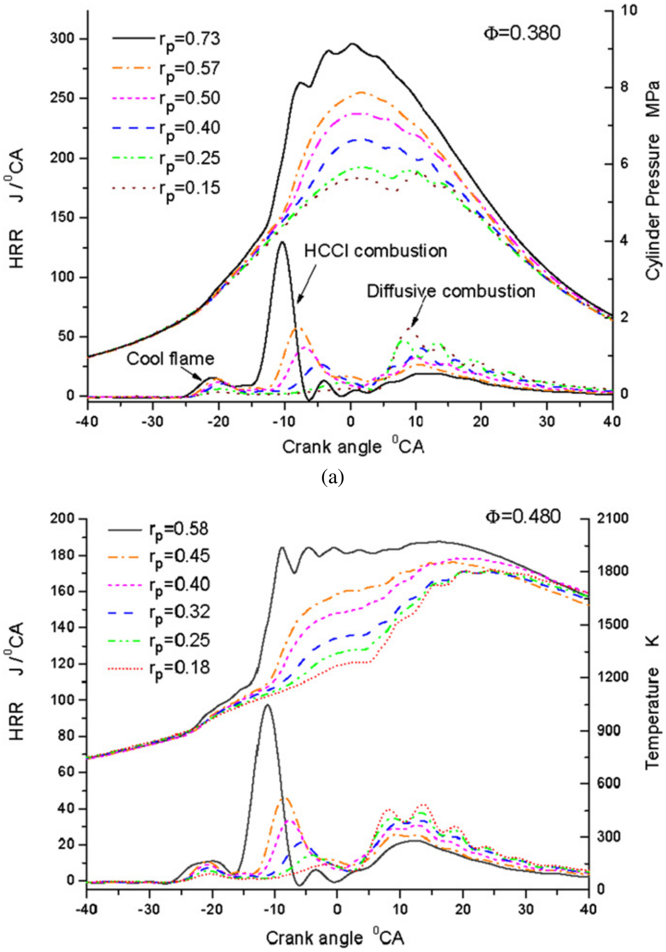


Fig. 2. Effects of premixed ratio on HCCI-DI combustion characteristics: (a) $\Phi = 0.380$, (b) $\Phi = 0.480$.

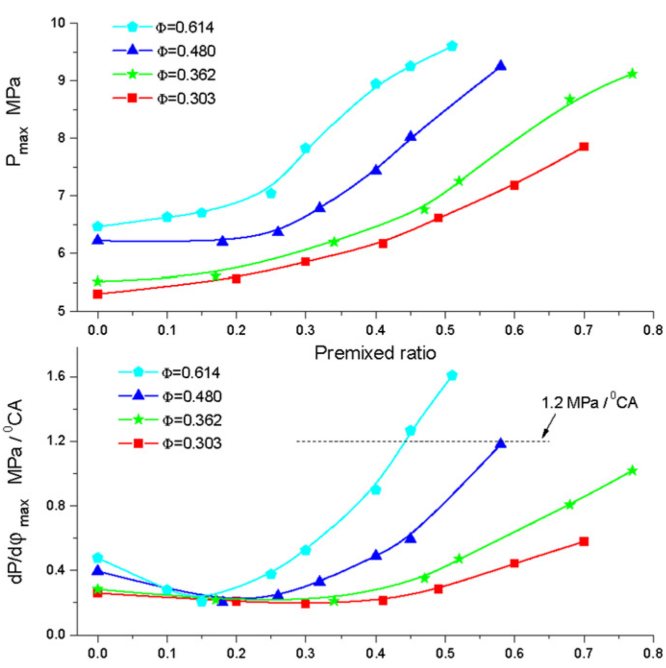


Fig. 3. Effects of premixed ratio on $P_{\max}$ and $\mathrm{d}P/\mathrm{d}\phi_{\max}$ of HCCI-DI.

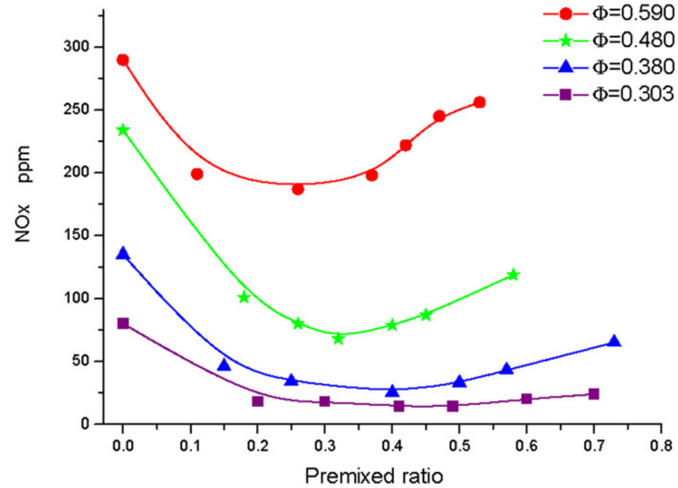


Fig. 4. Effects of premixed ratio on NOx emissions of HCCI-DI.

under the condition of low load and low premixed ratio, and severe knock could be avoided by using proper r_p at high loads.

3.2. Emission characteristics of HCCI-DI

HCCI combustion is well known for its ultra low NOx and soot emission. Thus it is considered in this study that the NOx and soot emissions are mainly due to the diffusive combustion of direct injection. On the other hand, the CO and UHC emissions mainly originate from the HCCI combustion since the prototype engine run in CIDI mode has a CO emission within 0.02% (volumetric) while the UHC emission is below 32 ppm.

Fig. 4 depicts the NOx versus premixed ratio with different equivalent ratios. It is evident that NOx is reduced dramatically with partial premixing and it exhibits a descending trend at first as a function of r_p which is approximately up to 0.3, but it shows a great tendency to increase when the premixed ratio was higher. This is because of the trade-off relationship between the reduction of NOx during HCCI combustion and the increase of NOx during diffusive combustion. On one hand, the *n*-heptane undergoing HCCI combustion produces nearly no NOx so that increasing premixed ratio would reduce the overall NOx. On the other hand, the heat release prior to the diffusive combustion raised the in-cylinder temperature that offset the NOx decrease. Therefore it should be notified that an optimal solution of premixed ratios for all operating conditions is quite possible.

To understand the effects of r_p on NOx more clearly, NOx emissions as a function of indicated mean effective pressure (IMEP) are shown in Fig. 5. Three representative premixed ratios were selected and the results of conventional CIDI were provided as a comparison.

The curves in Fig. 5 coincide well with the results in Fig. 4. The HCCI-DI is capable to decrease NOx effectively for all loads. In particular, the advantage is more distinct at higher loads (IMEP $>$ 0.5 MPa) and then a small premixed ratio ($r_p = 0.3$) is preferred to obtain the optimal performance.

Fig. 6 presents the smoke opacity to specify the effects of premixed ratio on soot emissions of HCCI-DI. The r_p appears

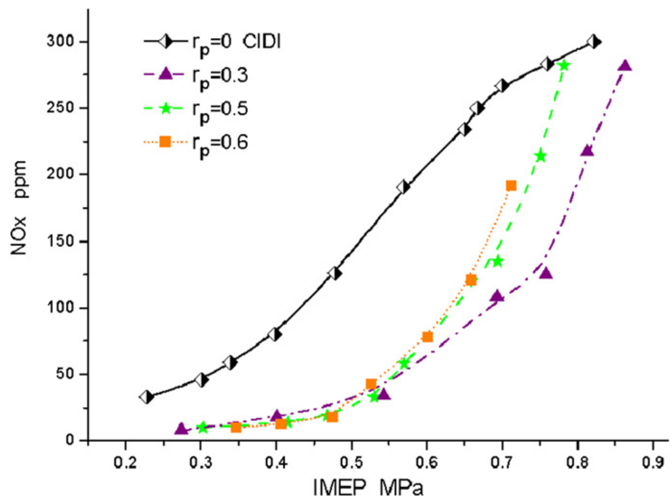


Fig. 5. NOx emissions of HCCI-DI mode vs. CIDI mode for all loads.

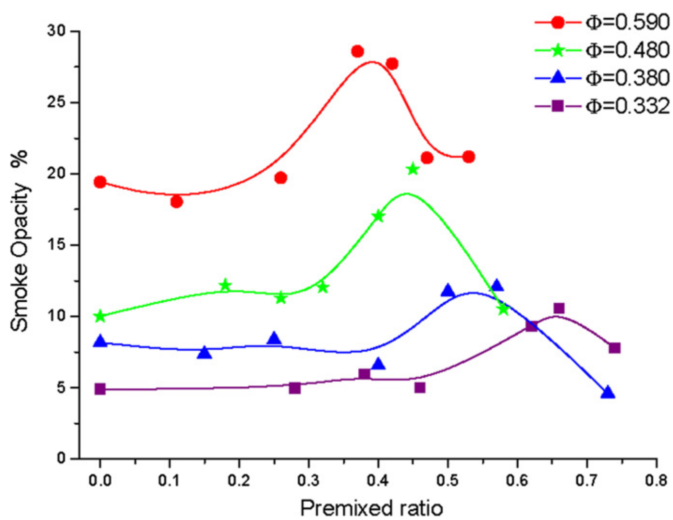


Fig. 6. Effects of premixed ratio on soot emissions of HCCI-DI.

to have little influence on soot emissions over a wide range of premixed ratio. The soot emissions remain almost at a constant level despite of the increase premixed ratio until r_p reaches a certain value. After that, the soot emissions indicate an obvious trend of increasing then decreasing as the premixed ratio further increases. Similar results were also observed by Simescu et al. [21] and Zhang [22]. The explanation to this observed soot increase may take into account several factors. As it is mentioned above, though HCCI combustion may not produce any soot, it elevated the temperature and decreased the in-cylinder oxygen concentration so that it played a similar role of internal EGR which had a positive impact on soot formation. Moreover, the active radicals and products formed during the HCCI combustion may also be considered as a factor for their attribution of soot precursors. Besides, the outer layer of the directly injected oil spray ignites instantly under the condition of high in-cylinder temperature and pressure, causing the poor evaporation and oxidation of the inner core of the spray. The inner core of the oil spray may undergo decarburizing reactions thus resulting in more soot. Only when the premixed ratio is high

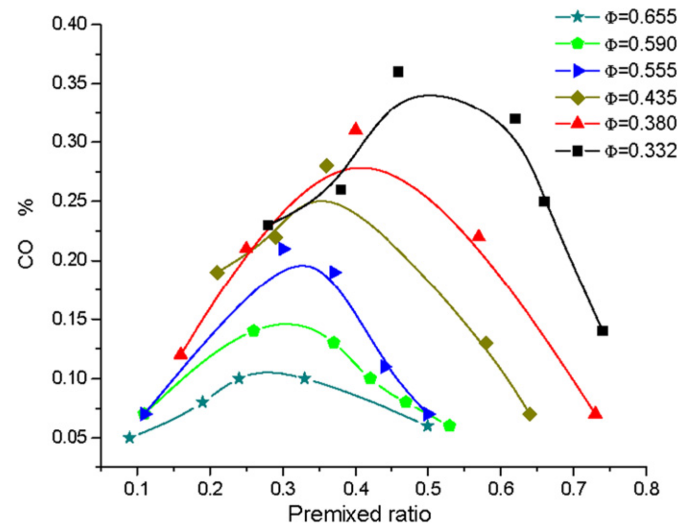


Fig. 7. Effects of premixed ratio on CO emissions of HCCI-DI.

enough that HCCI occupies a dominant status, the suppression effect of HCCI on soot emissions begins to appear.

Although the soot emission of HCCI-DI basically maintains the same level as conventional CIDI, it is worth noting that the inherent trade-off between NOx and soot could be solved in the aspect of optimal operating. Using the proper premixed ratios, for example $r_p = 0.3$, HCCI-DI could reduce NOx significantly while keep the soot emissions at the original level without much increase.

The much higher CO and UHC emissions also are the major problems of HCCI. Recently Amann et al. [23] evaluated the HCCI engine performances in comparison with modern advanced gasoline and diesel engines by calibrating HCCI operation factors, and found that on a break-specific basis the HCCI combustion emitted higher levels of CO and UHC emissions than the other combustion concepts. In general, it is widely accepted that CO emissions are controlled primarily by the fuel/air equivalence ratio and the reaction from CO to CO2 is sensitive to the bulk gas temperature. However UHC emissions come from the wall quench layer of the combustion chamber, ring-crevice storage and the absorption–desorption of fuel from oil layers. So it is expected that, as for the HCCI-DI combustion, the CO and UHC emissions will be equivalent to HCCI because the formation is of the same fundamental reasons, considering that CIDI combustion emits relatively very low CO and UHC emissions.

The CO emission of HCCI-DI as a function of premixed ratio is indicated in Fig. 7. CO increases at first with the premixed ratio up to a certain value and then the CO emission starts to decrease. This trend is mainly due to the premixed equivalence ratio and was also observed by Simescu et al. [21]. According to Dec's computational study [24], with the equivalence ratio of the mixture below 0.15, conversion of CO to CO2 could not go to completion. The results presented in Fig. 7 are consistent with Dec's calculations as it can be seen that under a certain total equivalence ratio, when the premixed ratio exceeds the critical value resulting in the CO emission starts to decrease, the corresponding premixed equivalence ratio is kept at approxi-

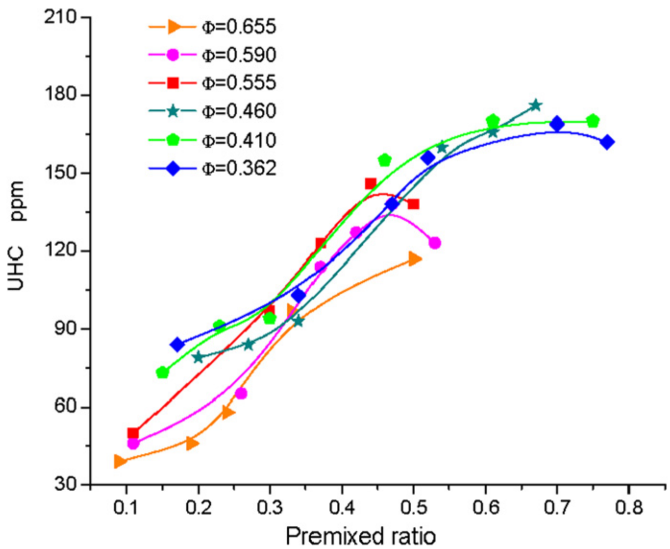


Fig. 8. Effects of premixed ratio on UHC emissions of HCCI-DI.

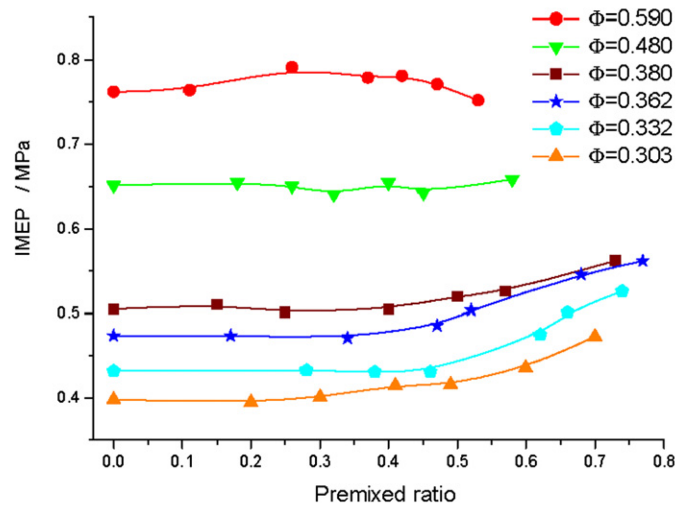


Fig. 9. Effects of premixed ratio on indicated mean effective pressure of HCCI-DI.

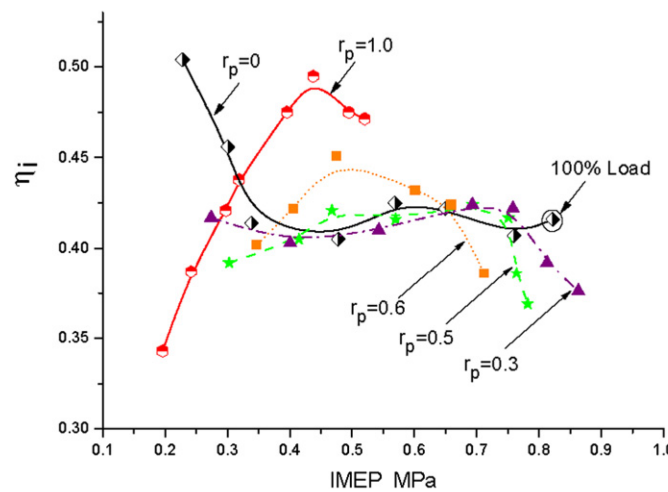


Fig. 10. Comparison of indicated thermal efficiency between HCCI-DI, HCCI and CIDI.

mately 0.176. (It is a little higher than 0.15 because heat transfer and boundary layer in the real engine may increase the critical value of the low limit.) And the lowest CO emission, maintaining at about 0.05% in volume, is mostly from the boundary layer where the further oxidation of CO is hard to proceed.

Similar to the spark ignition engines, the unburned hydrocarbons of HCCI-DI also originate mainly from crevices and quenching effect. It can be seen in Fig. 8 that the UHC does not exhibit the same trend as CO and it increases almost linearly with the premixed ratio. The main factor is that the premixed fuel/air mixture trapped in the crevice and boundary layer is hard to be oxidized during the low temperature HCCI combustion stage. It is worth noting that the highest UHC emission of n-heptane/diesel dual fuel HCCI-DI combustion is within 180 ppm, and based on the previous experimental work by Lü et al. [25], the UHC level of n-heptane port injection HCCI combustion is about 200 ppm or even higher with low equivalence ratio. So it is inferred that the heat release of the third stage CIDI combustion is good for the further oxidation of UHC produced in HCCI combustion stage.

3.3. *Effects of premixed ratio on the engine performance of HCCI-DI*

The indicated mean effective pressure (IMEP) and indicated thermal efficiency (η_i) are discussed in this study to evaluate the engine performance of HCCI-DI. Fig. 9 demonstrates the IMEP versus premixed ratio at various equivalence ratios. With an equivalence ratio below 0.480, the IMEP shows a trend of enhance as the premixed ratio increases. For the equivalence of 0.480, the plot of IMEP is quite flat and at the condition of even higher load, for instance when the equivalence is 0.590, the IMEP changes to decrease with the premixed ratio.

It can be concluded that for low to medium loads, high premixed ratios could improve the power output performances. However, for high load with a large premixed ratio, a deterioration may occur because of the over advanced combustion timing resulted from the excessive premixed equivalence ratio. This conclusion is consistent with the consensus developed over the years that HCCI is particularly suitable for partial loads. As to HCCI-DI, this feature still exists, whereas the upper limit of engine operation range has been largely expended.

To verify this, Fig. 10 providing comparison of indicated thermal efficiency between HCCI-DI, HCCI and CIDI, is in good agreement with the information revealed in Fig. 9. In Fig. 10, $r_p = 1.0$ and $r_p = 0$ represent fully HCCI and CIDI, respectively.

Restrained by the misfire and knock, although the η_i of fully HCCI is quite reasonable, the operation range in terms of IMEP is limited. Engine run in HCCI-DI mode can approach the 100% load of the prototype engine and has no restriction of the ignition problem at idle speed. The deterioration of η_i at high load is denoted, as the plots in Fig. 10 all exhibit a descending trend. This is of the same reason as for the deterioration of IMEP.

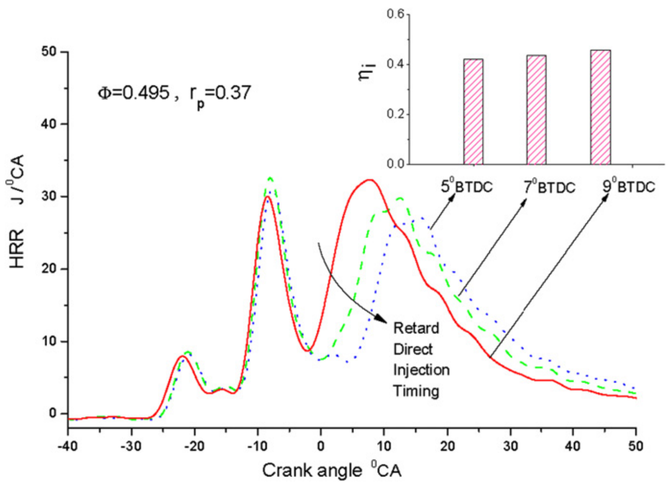


Fig. 11. Effects of direct injection timing on combustion characteristics of HCCI-DI.

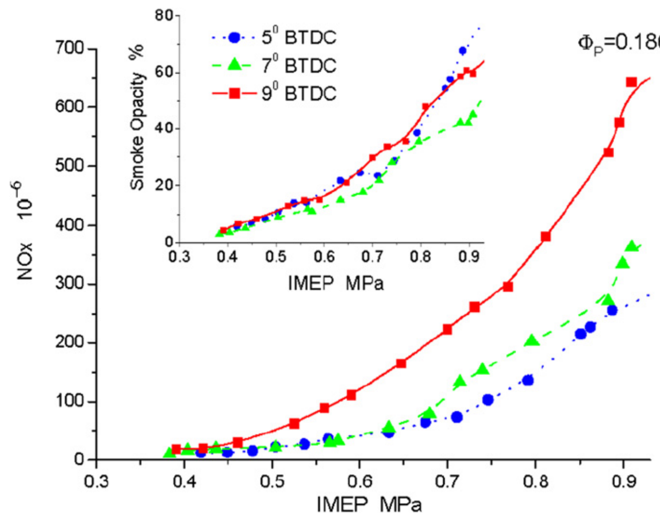
3.4. *Effects of direct injection timing on HCCI-DI*

The premix injection timing had no significant effect on the HCCI-DI since the *n*-heptane was port injected during the intake stroke after the intake valve opening event. However, direct injection timing plays an important role in CIDI combustion phase and it must have impacts on the charge stratification of HCCI-DI combustion [26]. In this experimental study, the effects of direct injection timing were investigated. And three different advanced angles of direct injector opening, 5° BTDC, 7° BTDC and 9° BTDC were used.

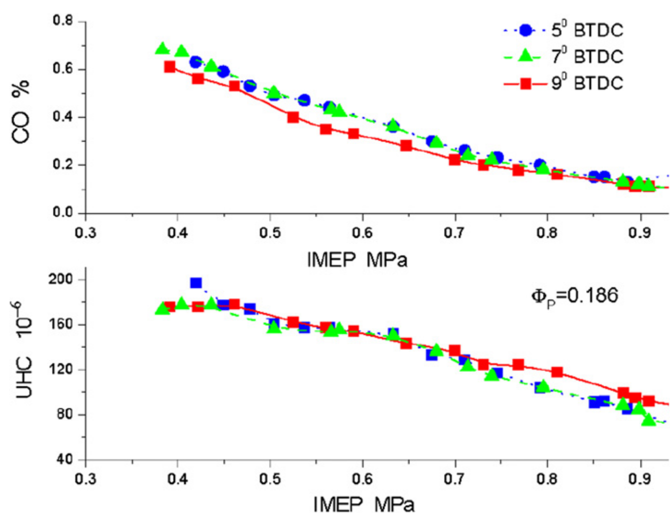
Fig. 11 presents the effects of direct injection timing on combustion characteristics of HCCI-DI under the condition of same equivalence ratio and premixed ratio. As the direct injection timing is retarded, the ignition timing of the third stage diffusive combustion is delayed when the prior two stages of HCCI combustion remain the same. Accordingly, the interrupt of combustion interference is prolonged which may lead to increasing duration of combustion and heat transferring loss. So it can be seen in the right upper corner of Fig. 11 that the indicated thermal efficiency slightly increases with advance of direct injection timing.

Fig. 12 shows the effects of direct injection timing on emission characteristics of HCCI-DI with the fixed premixed equivalence ratio $\Phi p = 0.186$. Fig. 12(a) reveals that NOx emissions decrease dramatically with the retard of direct injection timing, especially from 9° BTDC to 7° BTDC. Due to the possible reasons discussed in Section 3.2, the soot emission in terms of opacity depicts a relatively complex trend as there is no obvious reduce along with the retard of direct injection. There are no remarkable differences of CO and UHC emissions in Fig. 12(b) as the direct injection timing varies by four crank angles.

Consequently, taking into account the NOx and soot emissions as well as the indicated thermal efficiency, 7° BTDC is considered as the optimal advance of direct injector opening for the test engine in this study. Therefore it is used as the baseline for all the data presented in this paper.



(a)



(b)

Fig. 12. Effects of direct injection timing on emission characteristics of HCCI-DI: (a) NOx & soot, (b) CO & UHC.

4. Conclusions

(1) The HCCI-DI combustion with *n*-heptane/diesel dual fuel is a three-stage combustion process consisting of cool flame, HCCI combustion and diffusive combustion. With the increase of premixed ratio, the NTC is shortened, the peak in-cylinder pressure and temperature increases and the highest heat release rate of HCCI combustion phase rises.

(2) NOx emissions decrease firstly at low premixed ratios and exhibit a trend of increasing at higher premixed ratios. Generally the NOx of HCCI-DI combustion could be dramatically reduced in comparison with the prototype diesel engine.

(3) The premixed ratio has no significant effect on soot emission and the soot emission could remain at the same level but then have a peak value with a certain higher premixed ratio relating to the equivalence ratio.

(4) The change of carbon monoxide with premixed ratio is mainly depending on whether the premixed equivalence

ratio exceeds the critical value while the unburned hydrocarbon increases almost linearly with the premixed ratio mainly due to the incomplete oxidation in the boundary layer and the crevices.

(5) The engine performance of HCCI-DI is comparative with the prototype diesel engine. The IMEP increases with the increase of premixed ratio at low to medium loads. The indicated thermal efficiency shows deterioration at high load with large premixed ratios.

(6) The direct injection timing has distinct effects on the combustion and emission characteristics of HCCI-DI combustion. So that premixed ratio together with the direct injection timing are very important for HCCI-DI optimal operating.

Acknowledgements

This work was supported by the National Basic Research Program of China (Grant No. 2007CB210007) and Shanghai Nature Science Foundation (Grant No. 06ZR14045).

References

[1] Göran Haraldsson, Per Tunestål, Bengt Johansson, HCCI closed-loop combustion control using fast thermal management, SAE 2004-01-0943.

[2] Uwe Wangner, Razvan Anca, Amin Velji, Ulrich Spicher, An experimental study of homogeneous charge compression ignition (HCCI) with various compression ratios, intake air temperatures and fuels with port and direct fuel injection, SAE 2003-01-2293.

[3] Thomas W. Ryan III, Timothy J. Callahan, Darius Mehta, HCCI in a variable compression ratio engine-effect of engine variables, SAE 2004-01-1971.

[4] Jari Hyvonen, Goran Haraldsson, Bengt Johansson, Operating range in a multi-cylinder HCCI engine using variable compression ratio, SAE 2003-01-1829.

[5] Petter Strandh, Johan Bengtsson, Rolf Johansson, Bengt Johansson, Variable valve actuation for timing control of a homogeneous charge compression ignition engine, SAE 2005-01-0147.

[6] Fredrik Agrell, Hans-Erik Ångström, Bengt Eriksson, Jan Wikander, Johan Linderyd, Transient control of HCCI through combined intake and exhaust valve actuation, SAE 2003-01-3172.

[7] Xing-cai Lü, Wei Chen, Zhen Huang, A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR, Part 1: The basic characteristics of HCCI combustion, Fuel 84 (2005) 1074–1083.

[8] Xing-cai Lü, Wei Chen, Zhen Huang, A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR, Part 2: Effect of operating conditions and EGR on HCCI combustion, Fuel 84 (2005) 1084–1092.

[9] Salvador M. Aceves, Daniel Flowers, Joel Martinez-Frias, Robert Dibble, Fuel and additive characterization for HCCI combustion, SAE 2003-01-0184.

[10] Shigeyuki Tanaka, Ferran Ayala, James C. Keck, John B. Heywood, Two-stage ignition in HCCI combustion and HCCI control by fuels and additives, Combustion and Flame 132 (2003) 219–239.

[11] Daisuke Kawano, Hiroyoshi Naito, Hisakazu Suzuki, Effects of fuel properties on combustion and exhaust emissions of homogeneous charge compression ignition (HCCI) engine, SAE 2004-01-1966.

[12] Lucien Koopmans, Elna Strömberg, Ingemar Denbratt, The influence of PRF and commercial fuels with high octane number on the auto-ignition timing of an engine operated in HCCI combustion mode with negative valve overlap, SAE 2004-01-1967.

[13] Gregory M. Shaver, Matthew J. Roelle, J. Christian Gerdes, Modeling cycle-to-cycle dynamics and mode transition in HCCI engines with variable valve actuation, Control Engineering Practice 14 (2006) 213–222.

[14] Jari Hyvönen, Göran Haraldsson, Bengt Johansson, Operating conditions using spark-assisted HCCI combustion during combustion mode transfer to SI in a multi-cylinder VCR-HCCI engine, SAE 2005-01-0109.

[15] Lucien Koopmans, Hans Ström, Staffan Lundgren, Ove Backlund, Demonstrating a SI–HCCI–SI mode change on a Volvo 5-cylinder electronic valve control engine, SAE 2003-01-0573.

[16] Nebojsa Milovanovic, Dave Blundell, Stephen Gedge, Jamie Turner, SI–HCCI–SI mode transition at different engine operating conditions, SAE 2005-01-0156.

[17] Hongming Xu, Simon Rudolph, Zhi Liu, Stan Wallace, Steve Richardson, An investigation into the operating mode transitions of a homogeneous charge compression ignition engine using EGR trapping, SAE 2004-01-1911.

[18] Dae Sik Kim, Myung Yoon Kim, Chang Sik Lee, Effect of premixed gasoline fuel on the combustion characteristics of compression ignition engine, Energy & Fuels 18 (2004) 1213–1219.

[19] Dae Sik Kim, Myung Yoon Kim, Chang Sik Lee, Combustion and emission characteristics of a partial homogeneous charge compression ignition engine when using two-stage injection, Combustion Science and Technology 179 (2007) 531–551.

[20] Takashi Suzuki, Haruyuki Yokota, Yugo Kudo, Hiroshi Nakajima, Toshiaki Kakegawa, A new concept for low emission diesel combustion, SAE 970891.

[21] Stefan Simescu, Scott B. Fiveland, Lee G. Dodge, An experimental investigation of PCCI-DI combustion and emissions in a heavy-duty diesel engine, SAE 2003-01-0345.

[22] Long Zhang, A study of pilot injection in a DI diesel engine, SAE 1999-01-3493.

[23] Manfred Amann, Janet Buckingham, Evaluation of HCCI Engine potentials in comparison to advanced gasoline and diesel engines, SAE 2006-01-3249.

[24] John E. Dec, A computational study of the effects of low fuel loading and EGR on heat release rates and combustion limits in HCCI engines, SAE 2002-01-1309.

[25] Xing-cai Lü, Wei Chen, Yu-chun Hou, Zhen Huang, Study on the ignition, combustion and emissions of HCCI combustion engines fueled with primary reference fuels, SAE 2005-01-0155.

[26] Magnus Sjöberg, John E. Dec, Smoothing HCCI heat-release rates using partial fuel stratification with two-stage ignition fuels, SAE 2006-01-0629.