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Advanced combustion operation in a single-cylinder engine

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A B S T R A C T

In this paper advanced combustion concepts such as HCCI and PCCI were studied in a single-cylinder engine. PCCI was achieved by the combination of part aspiration and part direct injection of DME in the experiments, which was a compromise to obtain HCCI in that only a portion of the fuel was premixed and the portion of combustion was still controlled by the injection timing. Basic investigations toward the PCCI and HCCI combustion in a DME engine were carried out. DICI operation was also conducted to make a comparison. Results showed that as for the PCCI combustion operation, $p_{\max}$, $(dp/d\theta)_{\max}$ and heat release rate were between the values of HCCI and DICI operation and they increased with a rise of premixed ratio. The combustion duration for the PCCI combustion was longer than those of HCCI combustion, but was shorter than that of DICI combustion. Furthermore, the combustion duration decreased and the brake thermal efficiency increased with an increase in premixed ratio. CO and HC emissions for the PCCI combustion operation were lower than those of the HCCI engine. In comparison to conventional DICI operation, NO_x emissions for the PCCI combustion operation decreased significantly. Experiments also indicated that the fuel injection timing had a great influence on the performance and emissions of a DME engine at a PCCI combustion mode.

1. Introduction

Now the internal combustion engines are spread to the extent that they represent one of the main causes of pollutant production. Nevertheless, it is well known that the stocks of fuels traditionally used in this kind of engine will be able to satisfy the world's needs for few more decades. This explains the massive research activity, all over the world, addressed to the utilization of innovative fuels and combustion concepts in order to either replace the traditional ones or obtain a more efficient and clean combustion. It has long been recognized that oxygenated alternative fuels are less polluting than diesel fuel. Dimethyl ether(DME) is one of the most attractive fuels to solve the exhaust emission problems of diesel engines. Numerous experimental studies showed that DICI DME engines can achieve high thermal efficiency, low emissions, as well as quiet and smokeless combustion [1–4]. For a DME engine equipped with an inline pump and conventional in-cylinder direct injection and combustion strategies, lower CO and HC emissions can be obtained. However, it is very difficult for NO_x emissions reduction from DME engine with DICI technology without after-treatment.

Advanced combustion operating regimes or modes, such as homogeneous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI), are currently of interest to reduce engine emissions, specifically for NOx and particulate matter (PM) [5–7]. HCCI and PCCI operations shift combustion toward an increased premixed combustion phase, resulting in a fuel-lean charge and lowered combustion temperature and, thus, resulting in engine operation away from in-cylinder conditions that favor NOx and PM formation.

However, there are still many challenges to overcome before full HCCI operation can be used reliably over the full engine operation range. The challenges can be briefly summarized as combustion control (start and rate) and high emissions of unburned HC and CO emission.

The PCCI has been used in advanced combustion literature with multiple meanings. PCCI commonly refers to an advanced combustion process that allows for a large premixed burn. In PCCI, part fuel is injected early into the cylinder or aspirated at the intake port, during which an ignition delay occurs until cylinder conditions are right for auto-ignition. During the ignition delay, atomized fuel mixes with air, creating a locally fuel-lean charge. If injection of fuel continues past the point of auto-ignition, the burn will transit from a premixed burn to a diffusion burn. PCCI can be seen as an intermediate step between conventional diesel combustion and HCCI. PCCI permits a practical route to approximate HCCI [8–11].

The conventional in-cylinder DICI combustion and HCCI combustion have opposite advantages and disadvantages. Taking advantage of the characteristics of HCCI combustion and in-cylinder

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Nomenclature

DME	dimethyl ether
HCCI	homogeneous charge compression ignition
DI CI	direct injection compression ignition
CO	carbon monoxide
HC	hydrocarbon
NO_x	nitrogen oxide
PM	particulate matter
BMEP	brake mean effective pressure
TDC	top dead center
$p_{\max}$	the maximum cylinder pressure
$\theta_{p\max}$	crank angle occurring $p_{\max}$
$(dp/d\theta)_{\max}$	the maximum pressure rise rate
$\theta_{dp\max}$	crank angle occurring $(dp/d\theta)_{\max}$
$T_{\max}$	the maximum value of in-cylinder mean temperature
EGR	exhaust gas recirculation

DICI combustion in a DME engine, PCCI combustion with port aspiration and direct injection of DME, was proposed in this paper. In this concept, a portion of fuel was aspirated into the combustion chamber via the air intake port to cause HCCI combustion on the compression stroke because of the high volatility of DME, and the remainder of the fuel was injected by a conventional inline fuel pump. As a result, PCCI combustion happened, which included HCCI combustion and in-cylinder spray combustion.

The goal of this study was to investigate the advanced combustion operating modes, such as HCCI and PCCI in a DME engine. This work also examined the engine performance differences between the PCCI, DICI and HCCI modes, and then provided suggestions for the optimum operation regime with the advanced combustion technology.

2. Experimental engine and apparatus

A TY1100 compression ignition engine was used in this study. It was a four-stroke, single-cylinder, naturally aspirated, direct injection compression ignition engine. Engine specifications are shown in Table 1. DME fuel used in this experiment was from Shandong Jiutai Chemical Corporation and its properties were exhibited in Table 2.

Fig. 1 shows a schematic diagram of experiment setup. DME, stored in a small tank, directly entered the intake line to form the premixed mixture. An electric balance was used to monitor the mass change of DME in this small tank. The heating device was mounted in front of the air intake system to keep the gaseous DME temperature at 50 C. The flux rate of premixed DME was controlled through the valve by the operator. The pump-line-nozzle fuel system was still applied to inject DME directly into the cylinder and an electrical fuel pump was added into this fuel supply system to prevent vapor lock in the fuel system for direct injection compression ignition combustion.

Table 1
TY1100 engine specifications.

Bore/mm	110
Stroke/mm	115
Cylinder number	1
Compression ratio	18
Rated speed/rpm	2300
Rated power/kW	11

Table 2
The properties of DME.

Parameter	Value
Chemical formula	CH_3–O–CH_3
Cetane number	55–66
Low calorific number	27 600 kJ/kg
Stoichiometric air-fuel ratio	8.9
Viscosity	0.15 cP
Oxygen	34.8%(V)
Density	660 kg/m^3
Auto-ignition temperature	235 °C
Boiling point	−25 °C

In order to state the experiments clearly and facilitate comparison, the premixed ratio k was introduced in this paper and was defined using the following equation.

$$k = \frac{m_p}{m_p + m_{dj}} \times 100\%$$

Where m_p and m_{dj} indicate the mass consumption rate of port aspirated DME fuel and directly injected DME fuel respectively. So $k = 100\%$ is equivalent to fully HCCI combustion and $k = 0\%$ indicates the conventional DICI combustion.

The main test conditions were summarized in Table 3. Because HCCI operation could not be achieved at the high speeds and loads owing to the limit of knock, the engine speed and load were fixed at 1400 r/min and 0.34 MPa respectively in order to make a comparison among the different combustion modes.

Under the operation, an effort has been made to keep speed and load constant, while the amount of port aspiration and direct injection of DME is adjusted through the controller. Thus, the engine output power was maintained at a constant level and the brake mean effective pressure (BMEP) was held constant in all tests.

To help understand the varying trends in emissions and the brake thermal efficiency, the cylinder pressure was measured and heat release rate was computed. The cylinder pressure was recorded by a piezoelectric transducer (Kistler type) with a resolution of 10 Pa, and the dynamic TDC was determined by motoring. The crank angle signal was obtained from an angle generating device mounted on the crankshaft. The signal of cylinder pressure was acquired for every 0.5 °CA and the acquisition process covered 254 completed cycles, the average value of these 254 cycles used as the pressure data used for the calculation of combustion parameters. The combustion duration was defined as the angle interval from 10% of total accumulated heat release to the angle of 90% of total accumulated heat release.

Gaseous emissions were measured by AVL exhaust gas analyzer, in which NOx was analyzed with a chemiluminescent detector(CLD), and CO was analyzed with a non-dispersive infrared(NDIR) analyzer. HC was analyzed with a flame ionization detector (FID). In addition, no other technology such as EGR or variable compression ratio and so on was adopted in order to make the basic comparison between HCCI, DICI and the compound combustion operation in a DME engine.

3. Experimental results and discussion

3.1. Combustion characteristic

Based on the analysis of the measured indicator diagrams, it could be found that there were the great distinctions regarding with the values and positions of $p_{\max}$ (maximum cylinder pressure) and $(dp/d\theta)_{\max}$ (maximum pressure rise rate)under the various premixed ratios. Fig. 2(a) shows the $p_{\max}$ and $\theta_{p\max}$(crank angle occurring $p_{\max}$) under the various premixed ratios. The $p_{\max}$ of HCCI ($k = 100\%$) operation was 5.49 MPa and its position was 2 °CA BTDC.

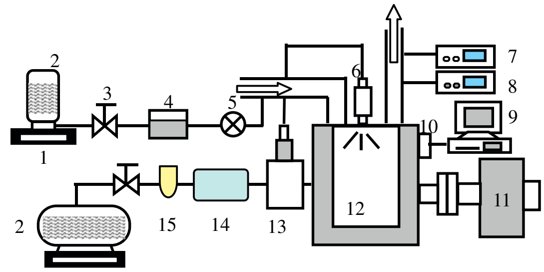

Fig. 1. The schematic diagram of the experiment setup.

Table 3
Test conditions.

Engine speed	1400 r/min
BMEP	0.34 MPa
Coolant temperature	80 C
Lubricant oil temperature	90 C
EGR	no

The p_{max} of PCCI combustion operation was 4.71–5.03 MPa and the position was 7–14 °CA ATDC. The p_{max} of CIDI($k = 0\%$) operation was 4.71 MPa and its position was 15 °CA ATDC. It could also be demonstrated that at the same loads, p_{max} dropped and θ_{pmax} occurred slightly later with a decrease in premixed ratio for the PCCI combustion operation.

The pressure rise rate($dp/d\theta$) is usually utilized as an index to describe the level of pressure oscillations. The faster the pressure rise is, the larger the pressure oscillation is, thus affecting the stability of engine work. Fig. 2(b) exhibits the $(dp/d\theta)_{max}$ and θ_{dpmax} under the various premixed ratios. It also reveals that in comparison to HCCI and DICI combustion, the value of $(dp/d\theta)_{max}$ for the PCCI combustion was between the two modes and increased with an increase in premixed ratio. Like the varying trend of θ_{pmax}, θ_{dpmax} was retarded somewhat with a decrease in premixed ratio for the PCCI combustion operation.

The maximum in-cylinder mean temperatures(T_{max}) for the various premixed ratios are shown in Fig. 3. From it, it is obvious that the maximum in-cylinder mean temperatures for the HCCI and PCCI combustion were lower than that for DICI operation. Because of part premixed combustion, the temperature of gases in the combustion chamber increased, thus reducing the ignition delay of the injected DME fuel.

Fig. 4 gives the heat release curves for various premixed ratios. A conventional direct injection combustion ($k = 0\%$) includes a premixing combustion and a diffusion combustion. However, the shape of the heat release curve for the PCCI combustion cases is not typical of DICI combustion and is more like the HCCI case. A two-stage heat release process existed for the PCCI combustion and HCCI combustion in our experiments. The heat release rate for the first stage was relatively low and that for the second stage was relatively high. However, for the PCCI combustion mode, the heat release curve shifted backward. It was worth noting that DME premixed ratio had a great impact on the elapsed time between the first stage and the second stage and this time decreased distinctly as the DME premixed ratio increased. Consequently, the start of second heat release stage was found to be advanced and the maximum heat release rate of the second heat release stage rose with an increase in DME premixed ratio.

The combustion durations for the various premixed ratios are also shown in Fig. 5. The combustion duration for HCCI combustion was the shortest and that for DICI combustion was the longest. The combustion duration also decreased with an increase in the amount of DME premixed ratio because of higher heat release during the cool flames when increasing the aspirated DME, thus decreasing the delay time of the main injected fuel.

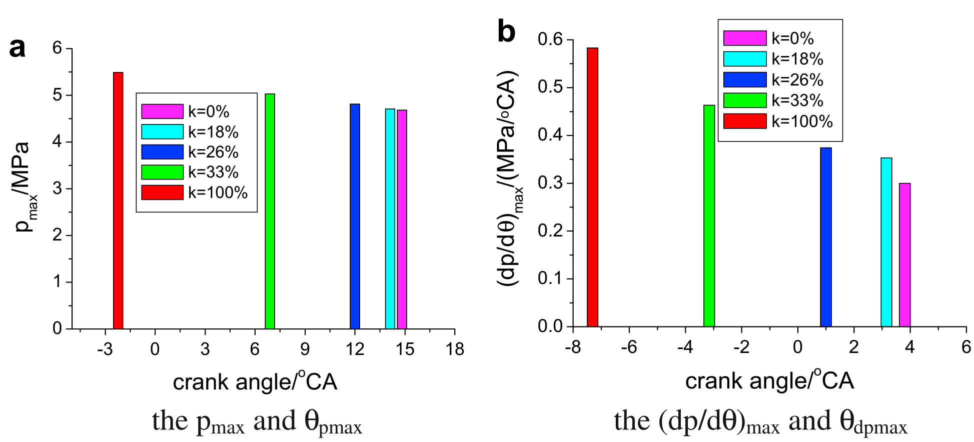

Fig. 2. The p_{max} ,$(dp/d\theta)_{max}$, θ_{pmax} and θ_{dpmax} for the various premixed ratios.

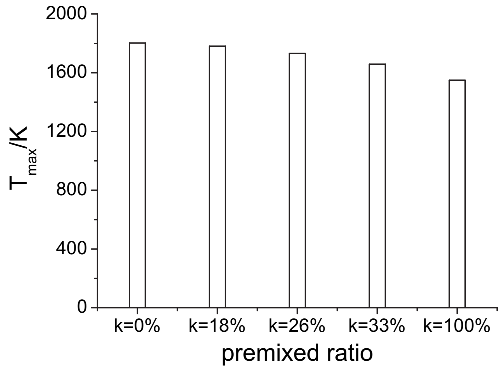

Fig. 3. The T_{max} for the various premixed ratios.

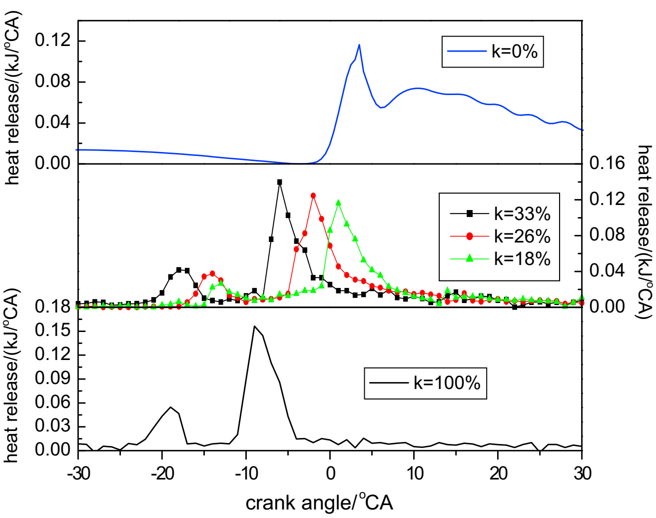

Fig. 4. The heat release rate curve for the various premixed ratios.

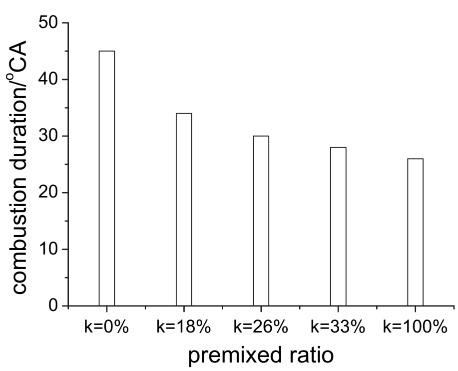

Fig. 5. The combustion duration for the various premixed ratios.

3.2. Brake thermal efficiency

Fig. 6 gives the brake thermal efficiency for the various premixed ratios. Due to the shortest ignition delay and fastest burning rate, the brake thermal efficiency for HCCI operation is highest for all operation modes. Compared with HCCI, the combustion duration elongated since a portion of fuel was injected into cylinder for the PCCI combustion, so the brake thermal efficiency decreased a little. Furthermore, it can also be seen that the brake thermal efficiency decreased a little with a decrease in premixed ratio.

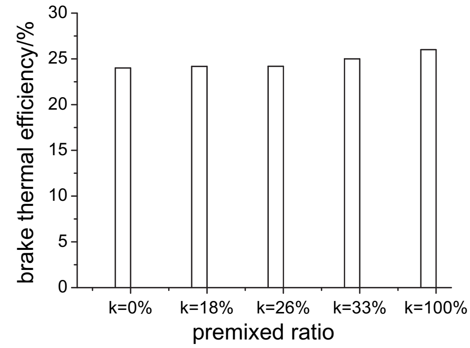

Fig. 6. The brake thermal efficiency for the various premixed ratios.

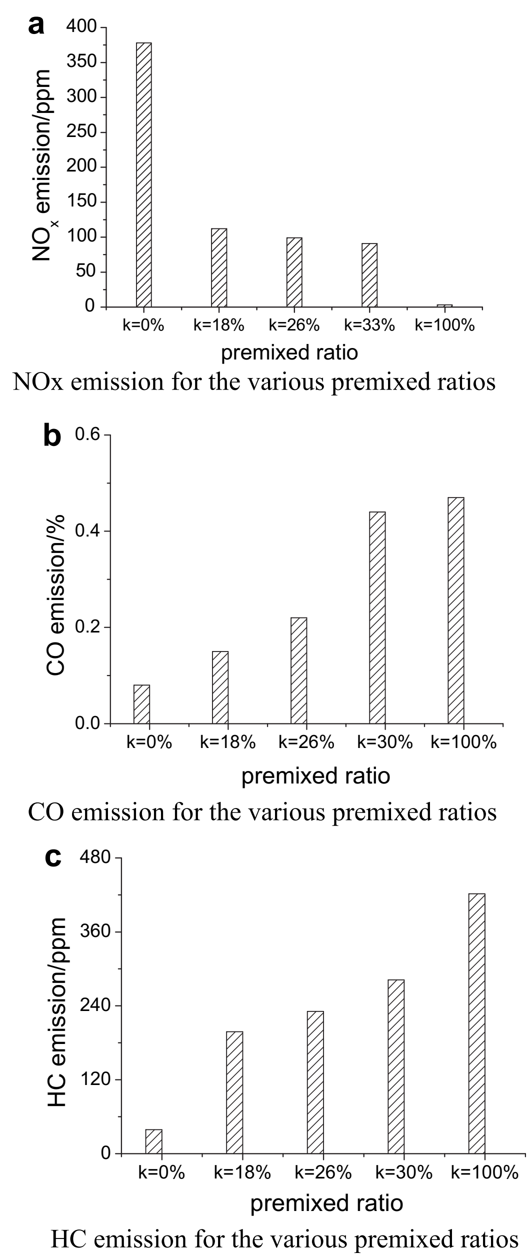

Fig. 7. Emissions for the various premixed ratios.

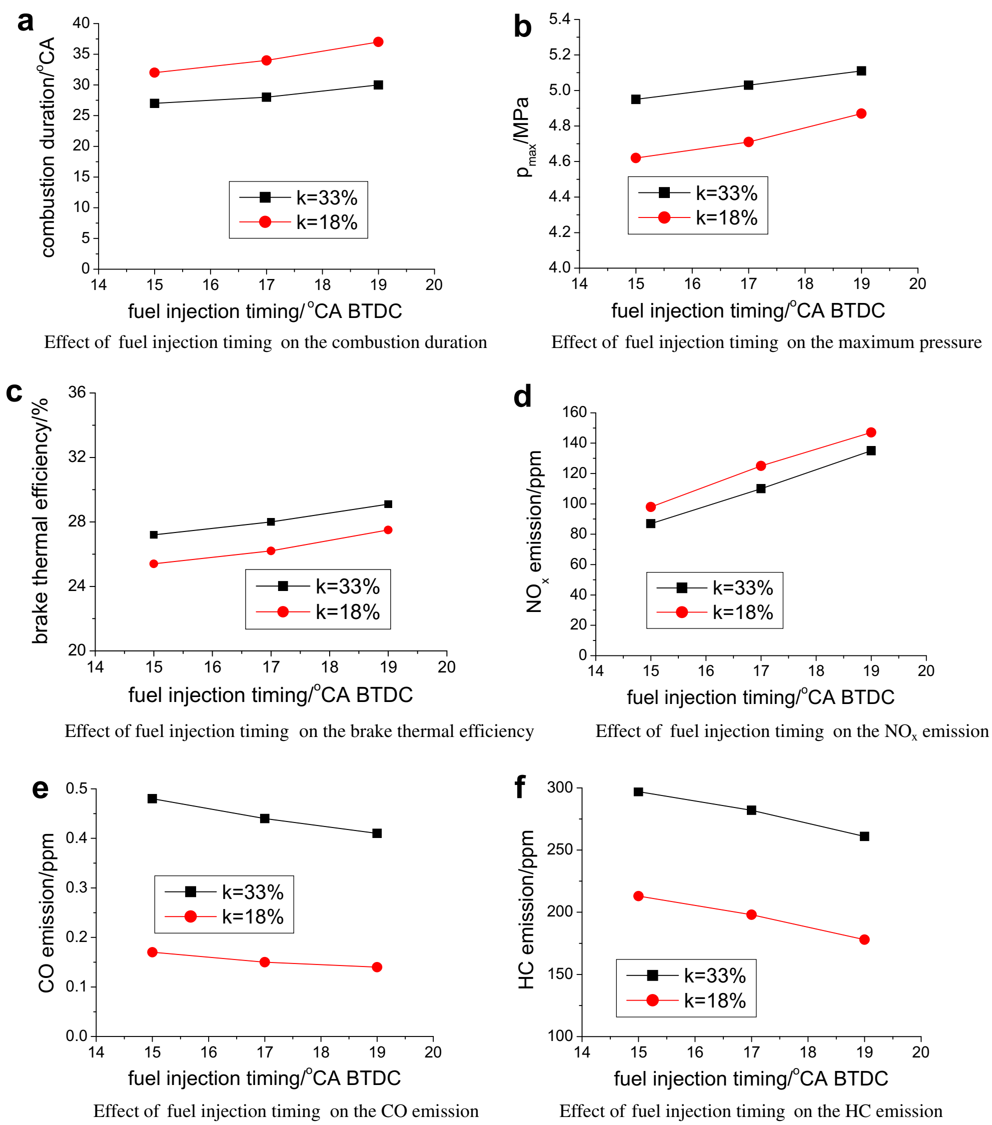

Fig. 8. Effects of fuel injection timing on combustion and emission characteristics for the PCCI combustion.

3.3. Emissions

Fig. 7(a) shows NO_x emission for the various premixed ratios. It was obvious that NO_x emission for DICI operation was highest and that for HCCI operation was lowest. NO_x emissions for PCCI combustion operation were between them. In addition, NO_x emission for PCCI combustion showed a descent tendency with an increase in premixed ratio.

In case of DICI mode, the diffusion burning is dominant and most of the NO_x is formed around the stoichiometric zone of the diffusion flame and the NO_x emission is basically controlled by thermal NO_x mechanism. However, for HCCI operation, the higher excess air ratio results in lower overall and local temperature, suppressing the formation of thermal nitric oxides. NO_x emissions are very low for the DME HCCI engine. However, HCCI operation is achievable in a narrow power output range. Similar results were also observed by Refs. [12–14]. In the PCCI mode, the NO_x emission could be controlled by thermal NO_x mechanism and N_2O mechanism, depending on the level of premixed ratio. As for the PCCI combustion, premixed combustion by a lean fuel/air mixture introduced via the air intake port slightly shortened the ignition delay of injected DME fuel, reduced the heat release in the premixed combustion phase, and thus suppresses the NO*x* formation. Meanwhile, the lower cylinder mean temperature resulted in lower NOx emissions. On the other hand, the gas after premixed homogeneous charge combustion can function as internal EGR and reduce the charge oxygen concentration by dilution, and therefore, suppress NO*x* formation.

HCCI combustion [5–10] typically resulted in higher CO and HC emissions than conventional diesel or spark-ignited combustion. In general, it was widely accepted that CO emissions were controlled primarily by the excess air ratio and high CO was due to a lack of full oxidation and an incomplete combustion. HC emissions came from the wall quench layer of the combustion chamber, ring-crevice storage and the absorption of fuel from oil layers. So it was expected that, as for the PCCI combustion, the CO and HC emissions would be similar to those from HCCI because the formation was due to the same fundamental reasons.

CO and HC emissions for DME PCCI combustion were much higher than those for DICI combustion, however, they were much lower than those for HCCI combustion. Obviously, the relatively high levels of CO and HC after the premixed homogeneous charge combustion are expected to be further oxidized during the period of the in-cylinder spray combustion. Further, the high CO emission in HCCI and PCCI combustion mode is also expected that the considerable level of CO could be generated due to the incomplete combustion near cylinder wall. From Fig. 7(b) and (c), it was also obtained that CO and HC emission increased with an increase in premixed ratio.

Like DME DICI operation, smokeless combustion was realized at all test conditions for the DME PCCI combustion and the HCCI engine.

3.4. Influence of fuel injection timing

The fuel injection timing plays an important role in the combustion and emissions. The state of charge into which the fuel injected changes as the fuel injection timing is varied, and thus ignition delay will vary. If the fuel is injected earlier, the initial temperature and pressure are lower, so the ignition delay will increase. If injection starts later (when the piston is closer to the TDC) the cylinder temperature and pressure are initially slightly higher, a decrease in ignition delay proceeds. Hence, the fuel injection timing variation has a strong effect on the exhaust emissions, especially on the NOx emissions, because of the changing of the maximum temperature in the engine cylinder [15,16]. The three values of f fuel injection timing (15 °CA BTDC, 17 °CA BTDC and 19 °CA BTDC) were studied for PCCI operation in the single-cylinder engine.

Fig. 8(a) shows the combustion duration of the PCCI combustion versus the fuel injection timings. The figure shows that the combustion duration increases with the advancing of the fuel injection timing. Advancing the fuel injection timing angle would bring the injected fuel spray into a relatively low gas temperature environment and increase the period of the ignition delay. Thus, the whole combustion duration increased. Fig. 8(b) shows the maximum cylinder gas pressure (p_{max})of the PCCI combustion versus the fuel injection timing. p_{max} increased with the advancing of the fuel injection timing, and this can also be explained by the high premixed burning rate due to the long ignition delay and more mixture prepared within the ignition delay period. Meanwhile, early injection timing will bring the premixed combustion phase closer to the TDC, resulting in a high value of cylinder pressure.

Fig. 8(c) gives the brake thermal efficiency of the PCCI combustion versus fuel injection timing. It can be seen from this figure that the brake thermal efficiency increases with advancing the fuel injection timing. The increase in thermal efficiency with advancing the fuel injection timing was also related to the increase in the gas temperature with early injection timing.

Like the varying trend of conventional DICI combustion, NO_x emissions(shown in Fig. 8(d)) decreased dramatically with the retarding of the direct injection timing. Because more fuel burns after TDC, retarding the injection timing decreases the peak cylinder pressure and peak temperatures. CO and HC emissions increased with the retard of direct injection timing in the experiment, as shown as in Fig. 8(e) and (f).

4. Conclusion

Studies on combustion and emission characteristics of the compound combustion were carried out in a single-cylinder DME engine. The main results were summarized as follows:

1. The combustion duration for PCCI combustion was between HCCI and DICI combustion and it shortened with an increase in premixed ratio. p_{max}, $(dp/d\theta)_{max}$ and peak heat release for DME compound combustion were between HCCI and DICI combustion and they increased with an increase in premixed ratio. θ_{pmax} and θ_{dpmax} for DME compound combustion were also between HCCI and DICI combustion and they retarded with an increase in premixed ratio.
2. Although NOx emissions for PCCI combustion operation were higher than that for HCCI operation, they were still much lower than that at the DICI mode. HC and CO emissions were also very high for the compound combustion, however, they were much lower than that for HCCI.
3. The fuel delivery advanced angle had a great effect on the performance and emission of a DME engine with the PCCI combustion technology. The combustion duration, p_{max}, brake thermal efficiency and NO_x increased with the advancing of the fuel delivery advanced angle from 19 °CA BTDC to 15 °CA BTDC. CO and HC reduced with the advancing of the fuel delivery advanced angle.

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