Research on Combustion Characteristics and Performances of an HCCI Engine Fuelled with Methanol

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Keywords: HCCI; Methanol; Combustion characteristics; Performance

Abstract. The impacts of engine operating parameters, including intake charge temperature, fuel-air equivalence ratio and engine speed, on methanol HCCI combustion and performances were investigated on a modified diesel engine. The results show that the intake charge temperature is the most sensitive parameter of methanol HCCI combustion. The engine speed scopes are changed with boundary conditions, and the optimum speed increases gradually with equivalence ratio. But, the equivalence ratio just has a little influence on thermal efficiency. Moreover, CA50 is more sensitive to the boundary conditions than CA10, and the appropriate combustion phase for methanol HCCI engine may be within 8°CA for CA50 and within 11°CA for combustion duration.

Introduction

HCCI engine, combining the advantages of spark ignition engines and compression ignition engines, has the abilities to improve thermal efficiency and emissions especially NOx and soot simultaneously. Despite these benefits, HCCI combustion also has to face many challenges, such as controlling the ignition timing, reducing HC and CO emissions, extending the operation range [1]. The combustion process of HCCI engines is dominated by the chemical kinetic reactions of the fuel-air mixture. From the macroscopic view, it is mainly affected by properties of fuel and the histories of in-cylinder pressure and temperature of air-fuel mixture. So the choose of fuel is important for HCCI engine to get a good performance [2].

HCCI engines displayed strong compatibility on fuel, many fuels, once used in traditional engines, can be used in HCCI engines, but it would display different combustion and emission properties when fuelled with different fuel [3]. Many fuels have been studied on HCCI combustion, such as traditional fossil fuel, e.g. gasoline and diesel [4, 5], pure hydrocarbon, e.g. n-heptane, iso-octane, hydrogen, nature gas, acetylene [6-8]. In addition, many dual-fuels used on HCCI engines were also investigated, such as gasoline/ethanol [9].

Nowadays, oxygenated fuels such as ethanol, dimethyl ether and methanol are being increasingly concerned, because, owing to the oxidation of oxygen, they have the potential to reduce the engine-out HC and CO emissions [10, 11]. Of all the oxygenated fuels, methanol contains the highest ratio of oxygen and mainly produced by coal with relative rich reserves, so it can be acted as alternative fuel for the traditional engine fossil oil that be about to dried up [12]. The methanol has been used as additive on HCCI engine fuelled with n-heptane [13]. And some researchers have investigated the cyclic variations of methanol HCCI combustion by statistical method at a constant engine speed [11]. In this paper, our studies mainly focus on the impacts of the intake charge temperature, engine speed and mixture concentration on the combustion of methanol HCCI engine on a modified four-stroke diesel engine.

Test Apparatus and Procedure

Experimental setup. A two-cylinder, four-stroke diesel engine of CT2100Q is converted into a single cylinder HCCI operation. Its second cylinder, the specifications are given in Table 1, was modified into HCCI combustion mode, while the first cylinder remained diesel mode. The test system is shown in Fig.1.

Tab.1 Specifications of HCCI cylinder.

Bore×stroke[mm×mm]	100×105	IVO[℃A]	17
Displacement volume[mL]	825	IVC[℃A]	43
Compression ratio	17:1	EVO[℃A]	47
Ratio of crank radius to connecting rod	0.32	EVC[℃A]	17

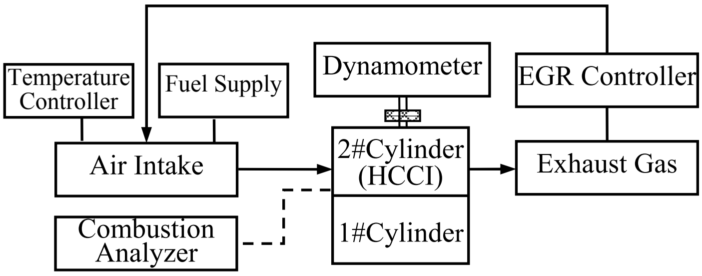

Fig.1 Test system

The electric heater with temperature controller is used to control the intake charge temperature of HCCI. The injector in the inlet pipe is used to provide fuel to get the homogeneous mixture. And the injection signal was acquired and transmitted to the electronic control unit (ECU) by a Hall sensor attached to the camshaft. The ECU has the ability to control the injection timing and duration, then to control fuel-air equivalence ratio. An electric whirlpool dynamometer is used to control the engine speed and load. The cylinder pressure is measured by a piezo-electric pressure transducer (Kistler 6052A), with a timing resolution of 0.25°CA. The output signal from the transducer is converted into an amplified voltage by a 5019B amplifier, and lastly recorded by the CB566 combustion analyzer.

Before HCCI tests, the first cylinder is fuelled with diesel to warm up the engine. After the engine is warmed up, and the engine speed and intake charge temperature are adjusted to the desired value, the diesel supply for the first cylinder is stopped and the HCCI fuel for the second cylinder is injected at the same time. Then, the HCCI tests can be started.

Definition of parameters. According to the value of in-cylinder pressure that is averaged from 60 consecutive cycles, the heat release rate (HRR) and cumulative heat release rate (CHRR) are calculated by zero-dimensional model.

The TDC of compression stroke is defined as 0°CA, and the phase of 10% and 90% of CHRR, named as CA10 and CA90, respectively, are regarded as the start of combustion (SOC) and the terminal point, and the time difference of them is defined as the combustion duration. Furthermore, the phase of 50% of CHRR, named as CA50, is an important parameter too. The cycle-to-cycle variation is represented by COV_{Pmax}, the coefficient of variation for the peak combustion pressure. The formula of COV_{Pmax} is given as follows.

$$COV_{P\max} = \frac{\sigma_{P\max}}{\overline{P}_{\max}} \times 100\% \tag{1}$$

Where, σ_{Pmax} is the standard deviation of the peak combustion pressure, $\overline{P}_{\max}$ is the mean peak pressure. The indicated thermal efficiency (ITE) is computed by the following formula.

$$ITE = \frac{P_i \cdot V_h}{H_u \cdot m_{cyc}} \times 100\% \tag{2}$$

Where, P_i is the indicated mean effective pressure (IMEP), V_h is displacement volume, H_u is low heating value of methanol, and mcyc is fuel mass per-cycle.

Combustion Characteristics

Cylinder pressure and HRR. Fig.2 shows the cylinder pressure and HRR with intake charge temperatures (T_{in}), fuel-air equivalence ratios (ER) and engine speeds (n) under different conditions. It can be found that the peak cylinder pressure and the peak HRR increase, and the phases of them also advance with the increase of intake charge temperature, fuel-air equivalence ratio and engine speed.

Apparently, the intake charge temperature is the most sensitive parameter for cylinder pressure and HRR, which can be seen from the curves that intake charge temperature goes up from 135°C to 160°C, the peak pressure increases by up to 22.53 bar and its phase advances about 15°CA. The reason is that the higher intake charge temperature can improve the mixing of fuel and air, and lift the temperature of mixture which enhances the combustion reaction. For methanol, the influence may be enhanced due to its high latent heat of vaporization.

The fuel-air equivalence ratio has a remarkable influence on the peak cylinder pressure and HRR but slightly on the phase of them. It is observed that the highest peak heat release rate corresponds to the richest mixture and the lowest peak heat release rate corresponds to the leanest mixture. For richer mixture, larger amount of fuel is auto-ignited at several locations, so the heat release rate is higher. In Fig.2, it also can be found that the peak cylinder pressure increases with the engine speed. There are mainly three reasons: a higher engine speed can boost charge-flow intensity and improve the mixing of fuel and air; the main stage of heat release is close to TDC, so that the most heat is released into the cylinder at small volume; and the heat-conduct loss is reduced because of shorter combustion time.

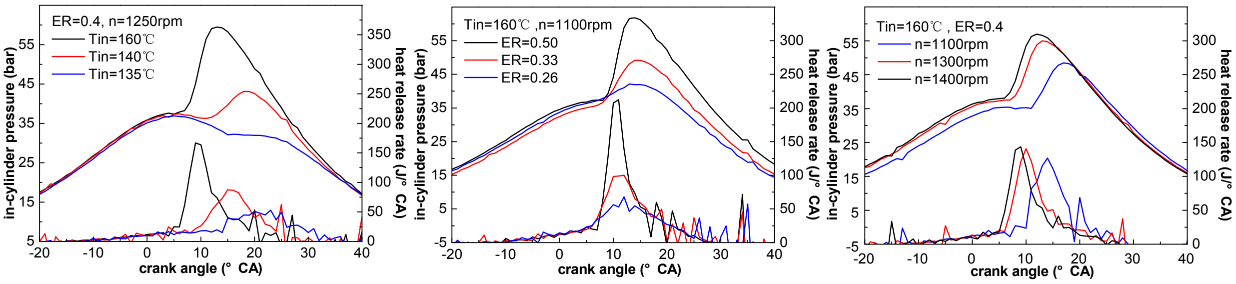

Fig.2 Impacts of T_{in}, ER and speed on cylinder pressure and HRR

Ignition timing and combustion duration. Fig.3 shows CA10, CA50 and the combustion duration with intake charge temperatures, equivalence ratios and engine speeds under different conditions. From the figure, it can be observed that with the increase of the intake charge temperature and engine speed, the ignition timing is advanced and the combustion duration is reduced. The reason is that higher intake charge temperature can increase mixture temperature, which enhances the combustion reaction. And why the same tendency appears in engine speed is that the heat loss of high speed is less than that of low speed, which makes mixture temperature rise quickly in compression stroke. With the increase of the equivalence ratio, the ignition timing (CA10 and CA50) is delayed and the combustion duration is shortened.

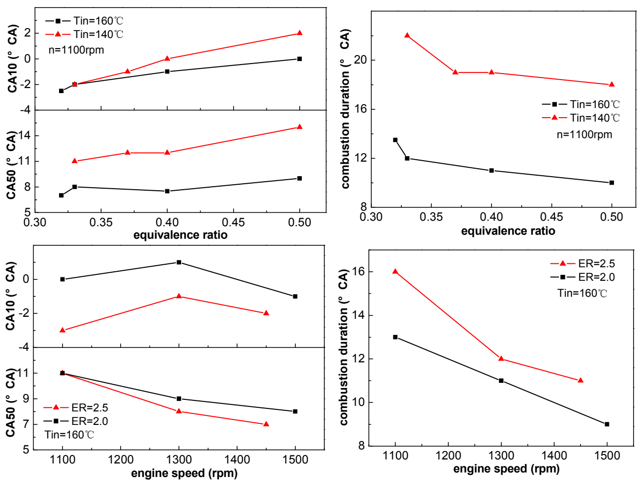

Fig.3 Impacts of T_{in}, ER and speed on the ignition timing and combustion duration.

When the intake charge temperature is kept at 140°C, CA50 almost appears after 11°CA, and the longest combustion duration is up to 22°CA, much longer than that when the intake charge temperature is up to 160°C. This is mainly because a low intake charge temperature represses the chemical reaction to delay the ignition timing and to slow the reaction rate; furthermore, when auto-ignition begins too late, the fast moving of piston in expansion stroke leads to the increase of cylinder volume that results in the decrease of cylinder pressure and temperature, so the combustion reaction rate is reduced.

Cyclic variation. Fig.4 displays the traces of COV_{Pmax} versus the engine speed and the equivalence ratio. As shown in figure 4 (a), an engine speed has no regular influences on the cyclic variation. Under the shown conditions, 1500rpm brings the lowest cyclic variation when ER is 0.5. Whereas, when ER is 0.4, the optimum engine speed appears at 1300rpm, and when the ER is 0.37, the optimum engine speed locates around 1200rpm. In a whole, the engine speed with the smallest COV increases gradually with the equivalence ratio. A higher engine speed may cause a larger intake flow fluctuation to induce a bigger variation of the equivalence ratio then deteriorate the cyclic variation; but a higher engine speed also brings into a more intense flow, beneficial to the combustion. For rich mixture, the latter has a larger impact on combustion than that of former, so the best engine speed trends gradually to be higher. On the contrary, as the mixture becomes leaner, the impact of the former strengthens to make the variation bigger in the high engine speed. Also it should be noticed that when the engine speed is very low, the combustion duration is too long to intensify the cyclic variation.

The figure 4 (b) reveals that the cyclic variation in richer mixture is much less than that in leaner mixture. The reason mainly associates with the combustion duration. The lowest cyclic variation happens at ER of 0.4 and engine speed of 1100rpm. Cyclic variation at the intake charge temperature of 140°C is bigger than that of 160°C, especially for lean mixture, which is mainly caused by late ignition timing and long combustion duration when intake charge temperature is 140°C.

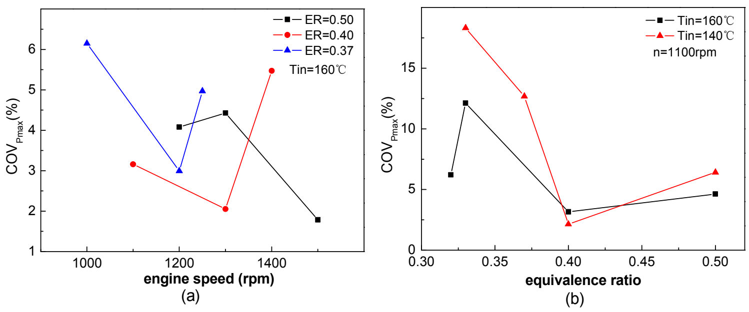

Fig.4 Impacts of T_{in}, ER and speed on cyclic variation

Engine Performances

Indicated mean effective pressure (IMEP). Fig.5 shows the impacts of the intake charge temperature, fuel-air equivalence ratio and engine speed on IMEP. It can be found that IMEP decreases with engine speed while increases with the intake charge temperature and fuel-air equivalence ratio. Even though a higher engine speed brings advanced ignition timing and short combustion duration to increase the peak cylinder pressure, IMEP still drops, the reason is that the indicated work is reduced due to the main stage of heat release is too early. The IMEP increase with the intake charge temperature at extent of 135°C to 160°C, the reason is that the mixtures has poor combustion under low intake charge temperature condition and reduces the total heat release. Higher equivalence ratio means more energy is released in a unit volume, which benefits to increase IMEP.

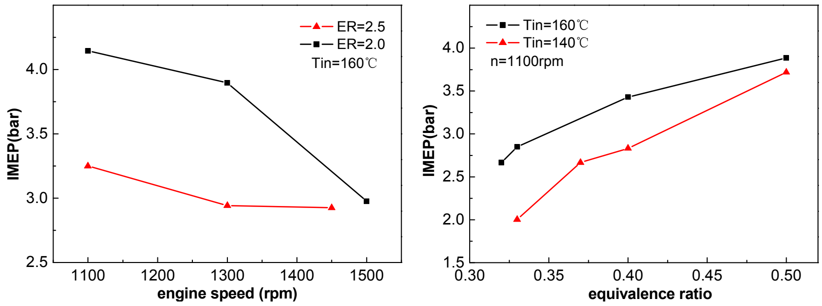

Fig.5 Impacts of T_{in}, ER and speed on IMEP

Indicated thermal efficiency (ITE). Fig.6 gives the ITE under the conditions of different engine speeds, equivalence ratios and intake charge temperatures. From the figure, it can be known that ITE is greatly affected by intake charge temperature and engine speed, compared to by equivalence ratio. The ITE increases and decreases greatly with the increases of intake charge temperature and engine speed, which is owing to its impacts on IMEP. Although IMEP improves with the increase of the equivalence ratio, the increased extent is less than the heat loss from reducing of burning ratio caused by the poor homogeneity of the mixture. So ITE decreases slightly with the increase of the fuel-air equivalence ratio.

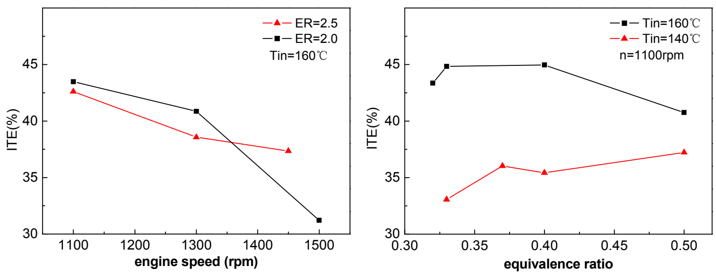

Fig.6 Impacts of T_{in}, ER and speed on ITE.

Both the ignition timing and the combustion duration play important roles on engine performances. Fig.7 reveals the impacts of CA50 and the combustion duration on ITE. Some discovers can be drawn from the figure. Under the condition of 1100rpm, the maximum ITE occurs when CA50 is 8°CA and the combustion duration is about 11°CA. And under the condition of 1250rpm, the maximum ITE happens when CA50 is 6.5°CA and the combustion duration is 9°CA. When CA50 appears after 8°CA, combustion duration is so long that in-cylinder pressure and temperature are reduced which result in a low ITE. Therefore, it can be concluded that the appropriate combustion phase should probably be within 8°CA for CA50 and within 11°CA for the combustion duration.

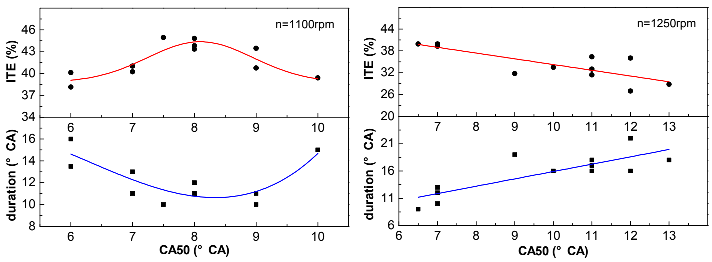

Fig.7 Impacts of CA50 and the combustion duration on ITE.

Conclusions

With the increase of intake charge temperature, fuel-air equivalence ratio and engine speed, both peak cylinder pressure and HRR increase, and the phases of them also advanced. The intake charge temperature is the most sensitive parameter of the three for in-cylinder pressure and HRR.

With the increase of intake charge temperature and engine speed, the ignition timing is advanced and the combustion duration is shortened. With the increase of the equivalence ratio, the ignition timing is delayed but the combustion duration is shortened; and CA50 is more sensitive to the boundary conditions than CA10.

The engine speed scopes are changed with boundary conditions. The optimum speed increases with equivalence ratio. Leaner mixture is easier to cause a larger cyclic variation.

The IMEP increases with the increase of intake charge temperature and fuel-air equivalence ratio; the ITE increases and deceases with the increase of intake charge temperature and engine speed, respectively, but little impacted by equivalence ratio.

Both the ignition timing and the combustion duration have important impacts on ITE. For HCCI combustion of methanol, the appropriate combustion phase should be probably within 8°CA for CA50 and within 11°CA for combustion duration.

Acknowledgments

This study was supported by the project of Natural Science Foundation of Shaanxi province (2012JQ7031).

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