COMBUSTION CHARACTERISTICS OF DIFFERENT PREMIXED METHANOL CHARGE COMPRESSION IGNITION COMBUSTION MODES

by

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This paper proposes two dual fuel combustion modes for a Diesel engine based on two alternative fuels and explores the influence of engine compression ratio on combustion and fuel economy characteristics under heavy loads. The results show that reducing the compression ratio can reduce the pressure rise rate of the combustion mode of methanol premixed charge induced ignition, owing to a decrease in the brake thermal efficiency.

Key words: *methanol, Fischer-Tropsch diesel, polyoxymethylene dimethyl ethers, premixed charge induced ignition, combustion*

Introduction

With the increasing shortages of oil resources, research on improving fuel economy as well as seeking appropriate alternative energy sources has become important. Oxygen compounds, such as alcohols [1] and ethers [2], can effectively reduce the emissions of Diesel engines, particularly soot emissions, and improve its brake thermal efficiency and have therefore been a hot topic of research for several years.

Methanol is widely recognized as a promising alternative fuel owing to a high octane rating, good anti-knock performance, widespread availability, and low production costs [3-7]. However, the low cetane rating of methanol can lead to difficulties in directly using it within compression ignition engines. Other materials and technologies have therefore been tested to make use of methanol as a fuel in Diesel engines, most often, the port premixed charge compression ignition combustion mode. Song *et al.* [7] and Zhang *et al.* [8] used methanol as the port injection fuel and the results showed a significant reduction in smoke emission and NO_x emissions. Zhang *et al.* [8] also observed a modest reduction in particulate mass and concentration of emissions. Li *et al.* [9, 10] took advantage of the low-reactivity of methanol and achieved methanol/diesel reactivity-controlled compression ignition. The results showed that several parameters, including the mass fraction of premixed methanol, start of injection (SOI), and initial in-cylinder temperature, can affect the combustion and emissions of a Diesel engine. Moreover, a high methanol fraction and advanced SOI resulted in improved fuel efficiency and lower emissions.

Polyoxymethylene dimethyl ethers (PODEn) are another promising oxygen-containing alternative fuel for Diesel engines and represents a mixture of ethers with a universal

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chemical formula of CH_3O (CH_2O) nCH_3 where *n* denotes the number of CH_2O groups. In general, only PODE3-5 are used in Diesel engines [11-14]. Wang *et al.* [15] observed the low-temperature heat release characteristics of PODE homogenous charge compression ignition and improved combustion when an engine is operating in this combustion mode. Liu *et al.* [16, 17] found that a blend of $PODE_{3-4}$ and diesel can be used to achieve soot-free combustion. Since PODE has a high cetane rating and strong ignition performance, it can be used as an ignition fuel alongside methanol as the main fuel in Diesel engines.

As a product of the coal chemical industry, Fischer-Tropsch diesel (F-T diesel) has a higher cetane rating than regular diesel fuel, with almost no sulfur, which can improve the performance of the exhaust after treatment with a catalyst and exhaust gas recycling system. Many scholars have carried out studies on the use of F-T diesel in Diesel engines. Results show that compared to traditional diesel, F-T diesel can significantly reduce CO and HC emissions, particularly during the cold start stage, without reducing the brake thermal efficiency [18-20].

Although a great deal of research has previously been performed on the use of PODE, F-T diesel, and methanol in Diesel engines, most research has combined them with a traditional petroleum fuel to create dual fuels. In this paper, two dual fuel combustion modes, premixed methanol charge induced ignition by F-T diesel or by PODE, are investigated, and could potentially be used to apply two distinct alternative fuels to the Diesel engine. The cetane rating of F-T diesel and PODE fuel is about 1.5 times higher than that of 0# diesel fuel, which will increase the ignition energy of the engine and cause the fuel to burn faster. The latent heat of vaporization of methanol is four times higher than that of diesel. A lot of heat in the cylinder is absorbed by the methanol vaporization process thereby reducing the temperature inside the cylinder, however, when the methanol premix is ignited, the rate of combustion is high, which helps to maintain a constant volume within the engine and dramatically improves the brake thermal efficiency of the engine. The influence of engine compression ratio on the combustion and economy characteristics under heavy load are explored.

Experimental apparatus

Fuel and the supply system

The physical and chemical parameters of the F-T diesel, PODE, and diesel fuel used in the experiments are presented in tab. 1.

Table 1. Combustion parameters of Diesel engine operating on various fuel types

Fuel types	Diesel	F-T diesel	PODE	Methanol
Lower heating value, [$MJkg^{-1}$]	42.5	44.2	19.1	19.66
Heat of vaporization, [$kJkg^{-1}$]	270	–	–	1109
Cetane rating	44-55	74	78.6	3
Oxygen content, [%]	0	0	47	34.8
Viscosity (60 °C), [mm^2s^{-1}]	2.77	1.32	1.05	1

The ignition fuel, either PODE or F-T diesel, was supplied by a direct injection system. The injection pressure was 20 MPa and the real maximum injection pressure was 25.3 MPa. Methanol was supplied by an independent methanol injection system on the intake port and the injection pressure was fixed at 0.35 MPa and regulated by an alcohol injection controller, which can be used to alter the methanol injection period and pulse width.

Test engine and facility

The test engine was a CY25 type single-cylinder Diesel engine. The main engine parameters are listed in tab. 2.

Table 2. Main parameters of the test engine

Item	Parameter
Combustor type	ω
Bore×stroke	115 × 115 mm
Displacement	1.25 L
Compression ratio	17:1
Rated power	12.5 kW/(2200 rpm)
Maximum torque	63.8 Nm/(1600 rpm)
Fuel supply advance angle	20° CA bTDC

Figure 1 shows the layout of the engine test rig. The combustion pressure was measured by a cylinder pressure sensor (6125C11, Kistler, Switzerland) and amplified by the charge amplifier (4618A2, Kistler, Switzerland). The combustion characteristics were analyzed using a DEWE-800-CA-SE combustion analyzer. To reduce measurement error, the number of sampling cycles was greater than 100 and the sampling interval was 0.1° CA. The volume of combustion chamber can be changed by machining the upper edge of combustion chamber, so as to change the compression ratio.

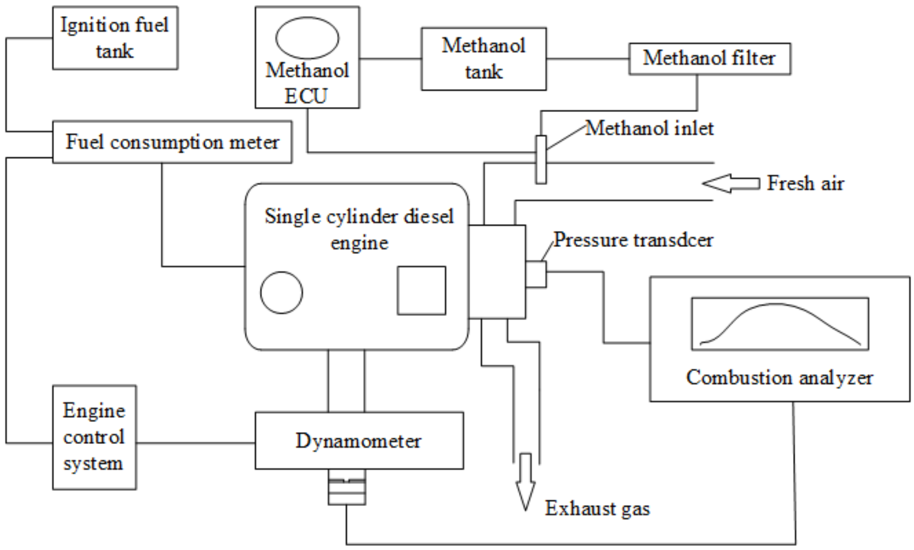

Figure 1. Schematic diagram of experimental set-up

Test procedure

During the experiment, the engine speed was maintained at 1200 rpm and the load was fixed at 0.6 MPa. The methanol mass ratio (MMR) was varied by simultaneously opening the throttle valve to allow PODE or F-T diesel into the cylinder and adjusting the methanol injection pulse width. The working conditions were held constant while the amount of methanol injection was gradually reduced and the ignition throttle opening was increased to reduce the MMR to PODE or F-T diesel. The fuel consumption and other data were recorded under the same working conditions each time. Then, compression ratio of the engine were varied and the previous operations were repeated. Fuel consumption and other data could measure under the various compression ratios, however, MMR values under different compression ratios are uncertain. During the experiment, variation of the compression ratio was achieved by changing the thickness of the cylinder gasket, thereby varying the clearance volume of the cylinder.

Data analysis

The MMR is defined as the ratio of methanol to total mass of fuel delivered to the engine per engine cycle and can be calculated:

$$\text{MMR} = \text{BM}(\text{BM} + \text{BP})^{-1} \tag{1}$$

where BM and BP refer to the fuel consumption of methanol and PODE or F-T diesel, respectively.

The heat release rate can be calculated by the following formula:

$$\frac{\mathrm{d}Q}{\mathrm{d}\theta} = \frac{r}{r-1} p \frac{\mathrm{d}V}{\mathrm{d}\theta} + \frac{1}{r-1} V \frac{\mathrm{d}p}{\mathrm{d}\theta} \tag{2}$$

where r is the isentropic index, which varies with temperature and composition of the working medium (ranging from 1.2 to 1.34), and its fixed value has little effect on the calculation results, V – the instantaneous in-cylinder volume, and p – the in-cylinder pressure.

Results and discussion

Changes in the compression ratio can affect the eddy current intensity, excess air coefficient, and residual exhaust gas coefficient, thereby influencing the combustion process. Here, after the compression ratio is reduced, the combustion characteristics are analyzed. The compression ratio of the original engine is 16.9. In contrast, the compression ratio can be reduced to 15.4 with the dual-fuel combustion modes and the oil delivery advance angle was 20 bTDC.

Fuel economy

Figure 2 shows effect of reducing the compression ratio on the brake thermal efficiency for both ignition modes. As the compression ratio is reduced, the brake thermal efficiencies of the two ignition modes decrease. The PODE ignition mode resulted in a decrease of 7% and the maximum decrease of the F-T diesel mode was 38%. The brake thermal efficiency is not only influenced by the compression ratio but also closely related to the capacity of the combustion cycle in the cylinder. The reduction of the compression ratio leads to a decrease in the average temperature inside the cylinder, the combustion cycle capacity is reduced, and the brake thermal efficiency decreases. From fig. 2, it can be observed that at a low compression ratio, the start of combustion (SOC) phase lags behind and almost approaches TDC, such that the main combustion phase appears as the piston moves downwards, and the brake thermal efficiency clearly decreases.

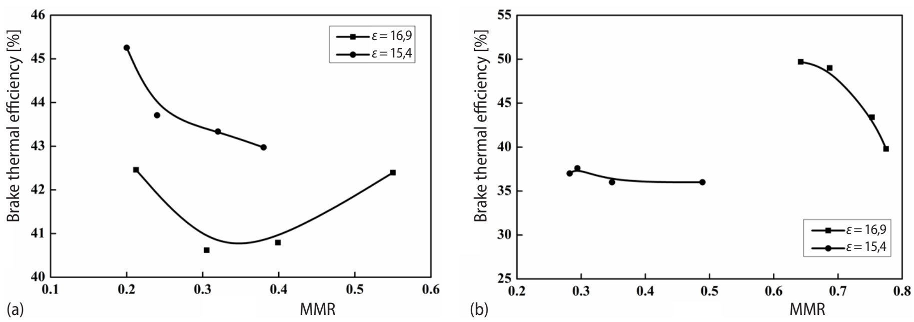

Figure 2. Effect of reducing the compression ratio on the brake thermal efficiency of the two dual-fuel ignition modes; (a) premixed methanol charge induced ignition by PODE, (b) premixed methanol charge induced ignition by F-T diesel

Combustion

Figure 3 shows the variation of the maximum cylinder pressure and the peak pressure rise rate (PPRR) with MMR for the two dual-fuel ignition modes. When the compression ratio is reduced from 16.9 to 15.4, the maximum pressure decreases in both cases, and the decrease of the PODE ignition mode is obvious for an MMR of less than 0.30. Reduction of the compression ratio can reduce the PPRR of the two ignition modes and the PPRR of the F-T diesel ignition mode is decreased by more than 17%.

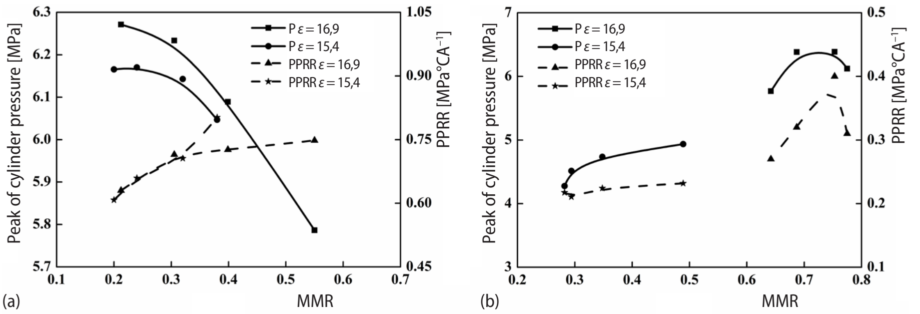

Figure 3. Variation of the peak of cylinder pressure and the PPRR with MMR for both ignition modes; (a) premixed methanol charge induced ignition by PODE, (b) premixed methanol charge induced ignition by F-T diesel

In this experiment, since the compression ratio is reduced by increasing the volume of the combustion chamber, this results in a smaller compression ratio, higher heat dissipation rate, and lower temperature inside the cylinder. The MMR increase will cause the temperature in the cylinder to increase and delay the SOC. However, with the increase of MMR, the proportion of premixed combustion in the cylinder increases correspondingly, which is beneficial to speeding up the combustion reaction. The combustion rate and phase will determine changes in maximum cylinder pressure and rate of pressure rise. At a speed of 1200 rpm, the time it takes for the reaction to occur in the cylinder is longer. The reduced compression ratio leads to a higher heat release rate and lower temperatures. Therefore, the maximum cylinder pressure and pressure rise rate decrease with the decreasing compression ratio. For the F-T diesel ignition mode, due to the high proportion of methanol and poor atomization of the ignition fuel, the peak values of cylinder pressure and pressure rise rate clearly decrease as the compression ratio decreases.

The effect of reducing the compression ratio on the peak heat release rate (PHRR) and the SOC phase for both ignition modes are shown in fig. 4. The PPRR increases as the com-

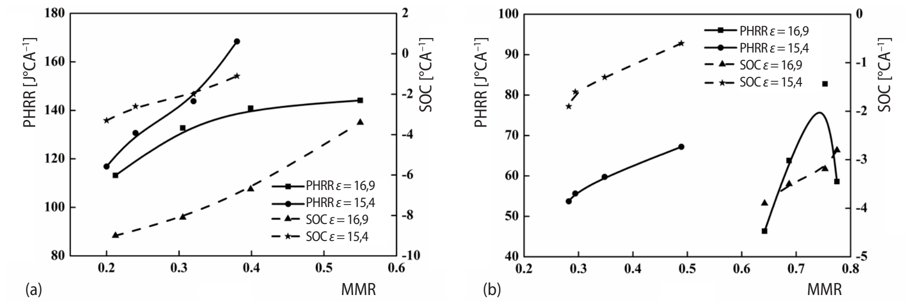

Figure 4. Effect of reducing the compression ratio on the PHRR and the burning SOC for the two ignition modes; (a) premixed methanol charge induced ignition by PODE, (b) premixed methanol charge induced ignition by F-T diesel

pression ratio is reduced. The maximum increase in PPRR is observed in the case of the PODE ignition mode and is 20%. The SOC of the two ignition modes is delayed when the compression ratio is decreased. The maximum delay for the PODE ignition mode is 5.7° CA, and for F-T diesel is 7° CA. The SOC is mostly determined by the ignition delay period of the ignition fuel. As the compression ratio decreases, the temperature and pressure in the cylinder also decrease. Under these conditions, the critical temperature of the ignition fuel increases, and the delay period is prolonged. In addition, the amount of combustible mixture increases, which leads to a higher burning rate and higher PHRR.

Conclusions

The fuel economy and combustion characteristics of premixed methanol charge induced ignition by F-T diesel and PODE combustion modes were studied. By changing the engine compression ratio, the following conclusions can be observed.

- The methanol premixed charge induced ignition combustion mode is better suited to the original 16.9 compression ratio.
- When the compression ratio is decreased to 15.4, the pressure rise rate of both the studied ignition modes decreases and the SOC phase changes. It is worth emphasizing that as the brake thermal efficiency was decreased, the PODE ignition mode decreased by 7%, and the largest decline of 38% was observed with the F-T diesel mode.

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Conflicts of Interest

The authors declare no conflict of interest.

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