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Combustion and emission characteristics of a turbocharged diesel engine using high premixed ratio of methanol and diesel fuel

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HIGHLIGHTS

- High premixed ratio of methanol was introduced on a turbocharged diesel engine.
- Ignition delay was prolonged but the combustion duration was shortened.
- The trade-off relationship between NO_X and soot emissions disappeared.
- DOC could effectively reduce the increased HC, CO, HCHO and proportion of NO_2 in NO_X.

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ABSTRACT

The combustion and emission characteristics of a dual fuel diesel engine with high premixed ratio of methanol (PR_m) were investigated. Experiments were performed on a 6-cylinder turbocharged, inter-cooling diesel engine. Methanol was injected through the intake port and ignited by direct injected diesel in the cylinder, the maximum PR_m was over 70%. The experimental results showed that with high PR_m, the maximum in-cylinder pressure increased from medium to high engine load but varied little or even decreased at low engine speed and load. High PR_m prolonged the ignition delay but shortened the combustion duration and decreased the in-cylinder gas temperature at ignition timing. Hydrocarbons (HC), carbon monoxide (CO), formaldehyde emissions and the proportion of nitrogen dioxide (NO_2) in nitrogen oxides (NO_X) increased significantly with the increase of PR_m while NO_X and dry soot emissions were significantly reduced, which meant the trade-off relationship between NO_X and soot emissions disappeared. The increased HC, CO and formaldehyde emissions could be effectively reduced by diesel oxidation catalyst (DOC) when the exhaust gas temperature reached the light off temperature of the DOC. After DOC, the NO_2 proportion in NO_X was greatly reduced to less than that of baseline engine at methanol premixed mode but increased slightly at pure diesel mode. The maximum PR_m was confined by in-cylinder pressure at high engine speed and load. But at low engine speed and load, it was confined by the high emissions of HC, CO and formaldehyde even after DOC.

1. Introduction

Diesel engines are widely used in agriculture, transportation and industry due to their high combustion efficiency, reliability, adaptability and cost-effectiveness. However, diesel vehicles are one of the main contributors of today's major concerns of energy shortage and environmental pollutions. In order to alleviate the pressure of energy shortage and meet the tightening emission regulations, especially to simultaneously reduce nitrogen oxides (NO_X) and particulate matter (PM), using oxygenated alternative fuels is one of the effective methods. Ren et al. [1] investigated the combustion and emissions of a diesel engine fuelled with diesel-oxygenate blends. Six oxygenated fuels which reflect ethers, esters and alcohols were selected. The results showed that the smoke decreased with the increase of the oxygen mass fraction in the blends regardless of the types of oxygenate additives and there was no increase of NO_X emission. For the benefits of saving fossil fuels and reducing emissions, oxygenated alternative fuels has gained more and more attentions recently. For example, methanol [2–4], ethanol [5,6] biodiesel [7,8] and dimethyl ether [9,10]

Abbreviations: PR_m, premixed ratio of methanol; HC, hydrocarbons; CO, carbon monoxide; NO_X, nitrogen oxides; NO, nitrogen monoxide; NO_2, nitrogen dioxide; N_2O, nitrous oxide; PM, particulate matter; DOC, diesel oxidation catalyst.

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are the main representatives that have been widely studied. Among these fuels, methanol is a promising fuel as it can be easily synthesized from abundantly available materials such as coal, nature gas and biomass [11].

In previous studies, there are three main applications of methanol in diesel engines including direct mixture [3,4], direct injection [12] and port injection [2,13]. Because of the miscibility problem, decreasing of heating value and some other issues, generally, the blending ratio of direct mixture cannot exceed 20%. And, the direct injection is hard to apply on the engine. On the contrary, port injection is very easy to realize, even on in-use diesel vehicles. Furthermore, the quantity of methanol injected can be adjusted according to the engine's operation conditions, and the premixed ratio of methanol (PR_m) may reach a much higher level.

In the previous studies of premixed methanol, the PR_m was relatively low, usually less than 30%. With dwindling of oil resources, the energy crisis is getting worse, especially in the countries with limited oil resources like China. Seeking for the possibilities to reduce the fuel consumption is of great importance. Based on that, in this study, a much higher PR_m was introduced. All previously reported studies on premixed methanol were conducted on naturally aspirated diesel engines [13,14]. Since the turbocharged, inter-cooling diesel engines are with higher thermal efficiency and widely used for heavy-duty vehicles, especially for urban buses in recent years. It is necessary to carry out experiments on this type of diesel engines. Meanwhile, in order to reduce the increased hydrocarbons (HC), carbon monoxide (CO) and formaldehyde emissions, commercial diesel oxidation catalyst (DOC) was used [14,15]. In this study, a special DOC was used to evaluate the effects on reducing HC, CO and formaldehyde emissions.

Because of its high latent heat of vaporization, methanol can reduce the intake air temperature, thus the in-cylinder combustion could be influenced especially at low engine speed and load. In this study, the low engine speed and load operation condition was selected and the effects of engine speed on the combustion and emissions were also investigated.

2. Experimental setup and procedure

2.1. Test engine and fuels

The baseline engine used was a 6-cylinder, turbocharged, intercooling, heavy-duty diesel engine with specifications shown in Table 1. The engine was modified to be compatible with diesel/methanol dual fuel. Methanol injectors were mounted near the intake port of each cylinder to ensure methanol uniformity. An electronic methanol pump was used to keep the methanol injection pressure at 0.35 MPa. Methanol was injected into the intake of each cylinder by methanol injectors to form premixed homogeneous methanol/air mixture. The methanol injection quantity was controlled by an electronic control unit. The diesel fuel injection system was an electronic unit pump system and remained unchanged. The schematic diagram of engine setup is shown in Fig. 1. The engine was coupled with a hydraulic dynamometer while its speed and torque were controlled by the Yike diesel engine test system. A special DOC for after-treatment of the exhaust gas was installed at the downstream end, 700 mm from the out let of turbocharger.

Table 1
Engine main technical specifications.

Type	Inline 6 cylinder, turbocharged, inter-cooler
Bore/stroke	108 mm/130 mm
Compression ratio	18.1
Power	192 kW/2200 rpm
Displacement	7.14 L
Fuel injection pump	Electronic unit pump
Fuel injection nozzle	6 Holes
Nozzle diameter	0.235 mm
Combustion chamber	ω bowl in piston

The diesel fuel used in this study contained sulfur less than 350 ppm by weight and the methanol was industrial grade. The consumption rates of the two fuels were measured by two FCM-05 instantaneous automatic fuel consumption meters. The general properties of diesel and methanol are shown in Table 2.

2.2. Sampling and analysis

The in-cylinder pressures were traced using a pressure transducer (Kistler 6052C) at 0.5 °CA intervals across 100 cycles and these pressure signal outputs were amplified and averaged by the AVL combustion analyzer IndiSmart 612. The shaft encoder used was AVL 365C. Based on above data, heat release rate was computed from the averaged in-cylinder pressure using Eq. (1) [16,17]:

$$\frac{dQ_g}{d\varphi} = \frac{dQ_n}{d\varphi} + \frac{dQ_w}{d\varphi} \tag{1}$$

where the net heat release rate, $\frac{dQ_n}{d\varphi}$ was determined by the traditional first law Eq. (2) [18,19]:

$$\frac{dQ_n}{d\varphi} = \frac{\gamma}{\gamma - 1} \cdot p \cdot \frac{dV}{d\varphi} + \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\varphi} \tag{2}$$

And the heat loss rate, $\frac{dQ_w}{d\varphi}$ was obtained from Eq. (3) in which h_c was estimated by means of Woschni model.

$$\frac{dQ_w}{d\varphi} = A_{ht} h_c (T - T_w) \tag{3}$$

The concentrations of the gaseous emissions were measured by employing an AVL SESAM Fourier Transform Infrared Spectroscopy (FTIR) multi-component exhaust analyzer, which is capable to determine up to 25 gaseous exhaust components simultaneously. The test range, accuracy and margin of error for each emission were shown in the study of Yang et al. [20]. In the measurement of FTIR, the HC represented the sum of the main hydrocarbon components and NO_X represented the sum of NO and NO_2. Smoke was measured by an AVL 415S filter paper smoke meter with average over five samples. Then specific dry soot emission (g/kW h) can be calculated from filter smoke number (FSN) through the empirical formula provided by the instrument manual [20,21].

2.3. Operating conditions

Experiments were performed both at the rated speed of 2200 rpm and a lower speed of 1000 rpm. The two different engine speeds with the same engine load of 0.62 MPa, which representing the medium engine load, was selected and compared to evaluate the effect of engine speed. The other two operation conditions, namely, 0.35 MPa at 1000 rpm and 0.88 MPa at 2200 rpm were selected to explore the effects of high PR_m at low engine speed and load and high engine speed and load respectively. The details of operation conditions and the mass flow of diesel and methanol are shown in Table 3. In this paper, x% PR_m refers to the case that premixed methanol takes up x% of the engine load and A-0.35, A-0.62, B-0.62 and B-0.88 refer to the four operation conditions respectively. The maximum PR_m exceeded 70% at A-0.35, A-0.62 and B-0.62, but was limited to 40% at B-0.88.

Experiments were initially carried out with diesel fuel alone. Then the engine load was reduced to $1 - x$% of the desired engine load and the rest x% was recovered by premixed methanol. During

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