Effect of dimethoxy-methane and exhaust gas recirculation on combustion and emission characteristics of a direct injection diesel engine

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1. Introduction

Diesel engine has been widely used in internal combustion engines due to its reliability, durability and high fuel efficiency. However, there exist two major challenges to keep diesel engine as one of the most popular power providers. One is related to fossil fuel sustainability: the crude oil resource on earth is limited; this fact is pushing the search for suitable alternative fuels. The other challenge is related with environmental concern. So far, compression-ignition engines have adapted many technical breakthroughs to meet the requirements of more and more stringent emission regulations.

In recent years, extensive researches have been focused on the effect of oxygenated fuels on diesel engine emissions. Yao et al. [1] carried out experimental research with a diesel/methanol compound combustion system (DMCC) and found that DMCC method could simultaneously reduce the soot and NOx emissions but increase the HC and CO emissions compared with the original diesel engine. Miyamoto et al. [2] and Akasaka and Sakurai [3] conducted research on diesel combustion improvement and emission reduction by using various types of oxygenated fuel blends. They pointed out that the combustion process and thermal efficiency have a certain improvement while the smoke emission can be reduced evidently by the use of oxygenate fuels.

DMM can be used as an oxygenated additive to blend with diesel fuel to improve combustion and reduce pollutant emission of diesel engines. The reason lies in the facts that DMM has not only high oxygen fraction, but also relatively high cetane number, it is non-toxic and highly miscible with diesel fuel. Song and Litzinger [4] studied the effects of DMM on combustion and emissions in an optical diesel engine, using the diesel-DMM blends with the oxygen concentration of 2% and 4% by fuel mass. Huang et al. [5] and Ren et al. [6] carried out experiments by fueling a diesel engine with diesel-DMM blends, in which the DMM volume fraction was from 0% to 20%. Also, an extended experiment sees the possibility of using diesel-DMM blends with DMM volume fraction up to 50% (the oxygen concentration was further increased up to 21% by mass) [7]. It is observed from these reports that CO, smoke emission and particulate mass could be reduced remarkably, while NOx and HC emissions present a fluctuation or increased trend. Liu et al. [8] investigated the PM emission of a compression-ignition engine using DMM blended with gas-to-liquids (GTL) and found that the smoke emissions as well as the particulate mass concentrations decrease with an increase of oxygen content in the fuel.

However, it is also testified that soot and NOx emissions can not be reduced simultaneously by simply using these oxygenated fuels. Further experiments implied that both soot and NOx emissions could be reduced by using the oxygenated fuels coupled with high exhaust gas recirculation (EGR) ratio. Ogawa et al. [9] pioneered in using DMM as a sole fuel for a compression-ignition engine. By
employing an exhaust gas recirculation system and a three-way catalyst, they reported that NOx, HC, CO and PM emissions can be reduced simultaneously. Tsolakis [10] investigated the combination of rapeseed methyl ester (RME) with EGR and found that apart from resulting in higher NOx reduction, the smoke emission and total particle mass were maintained at relatively low levels in comparison with diesel fuel. Murayama et al. [11] studied the particulate mass were maintained at relatively low levels in comparison with diesel fuel. Their results indicated that NOx emission can be reduced significantly without increasing smoke emission or decreasing the thermal efficiency of the engine.

Previous investigations on DMM focused mainly on the combustion characteristics and some gaseous emissions at some specific engine modes, and there is lack of overall study on the effect of various EGR ratios on the combustion process and exhaust gaseous emissions as well as the particulate total number and size distributions. Thus this study we aim at studying the influence of fuel constituents and EGR on the combustion and emission characteristics of a diesel engine fueled with diesel-DMM blends, and expect to establish the quantitative relationship among exhaust emissions and EGR ratio as well as the percentage of DMM in the fuel. The brake thermal efficiency, NOx, HC and CO emissions as well as the smoke emission and particulate number-size/mass distributions are reported in this study.

2. Experimental setup

2.1. Test engine and fuel properties

Experiments were conducted on a naturally-aspirated, water-cooled, single-cylinder, direct injection diesel engine. Specifications of the engine are shown in Table 1. The engine was coupled with an eddy-current dynamometer and engine operation was controlled by the diesel engine test system. The fuels adopted in the test include diesel fuel and three kinds of high oxygen content diesel-DMM blends in order to achieve more effective emission reductions. The blended fuels contain 20%, 30% and 50% by volume of DMM which are named as DMM 20, DMM 30 and DMM 50, respectively, corresponding to 8.6%, 12.7% and 21.1% by mass of DMM which are named as DMM 20, DMM 30 and DMM 50, respectively, corresponding to 8.6%, 12.7% and 21.1% by mass of oxygen in the blended fuels. Properties of diesel fuel, DMM and the three blended fuels are given in Table 2. As shown in Table 2, with the increase of the DMM volume fraction from 20% to 50%, the lower heating value becomes lower, which suggests that the amount of fuel injected per cycle should be increased to maintain the power output. In addition, DMM has lower viscosity and lower boiling point compared to the diesel fuel which could improve fuel atomization, leading to better air–fuel mixing. Moreover, with the increase of the DMM fraction in the fuel, the cetane number decreases and the latent heat of evaporation increases, indicating the possibility of lengthened ignition delay.

2.2. Experimental setup and measurements

Fig. 1 shows the schematic of the experimental system while the measuring instruments used are given in Table 3. Cylinder pressure was measured by a water-cooled Kistler type 6056A piezoelectric transducer coupled with a Kistler type 5011B charge amplifier. A data acquisition and analysis system (AVL DL 750) was used to digitize and record the analog signal from the amplifier for combustion analysis. Crankshaft position was measured by a Kistler crank angle encoder with a resolution of 0.2 crankshaft angle (deg) interval. The gaseous species in the engine exhaust were measured using online exhaust gas analyzers. The gaseous species in the engine exhaust were measured using online exhaust gas analyzers. A Horiba exhaust gas analyzer (MEXA-7100) was used to measure the exhaust gas; an opacity smoke meter, AVL DiSmoke 4000, was used to measure the smoke level.

Particle mass distribution can be estimated based on the particle size distribution of ELPI, if the particle density is known or can be assumed [13,15]. Nucleation particles are usually assumed to be spherical and their diameters determine the particle density. Also, researches showed that the use of a unit density can give qualitatively good agreement between the measured and calculated size distribution for the majority of the sizes, mainly for middle and low size (diameter) particles [16]. In this paper, diesel particle density is assumed to be 1 g/cm³.

In the experiment, the exhaust gas is introduced into the intake pipe to mix with fresh air after passing an EGR valve and an EGR cooler, in which the cooling water is fixed at a constant temperature. By adjusting the EGR value, the EGR ratio was set at 4%, 8%, 12%, 16%, 20% and 24%, which ranges from low to high EGR ratios. An ECM EGR 5230 analyzer was used to measure the EGR ratio which is kept on constant for each test mode, based on the CO2 concentration in the exhaust pipe and in the intake manifold as well as in the background gas. The EGR ratio can be calculated as follows:

\[
\text{EGR ratio} = \frac{[\text{CO}_2]_{\text{man}} - [\text{CO}_2]_{\text{bkg}}}{[\text{CO}_2]_{\text{exh}} - [\text{CO}_2]_{\text{bkg}}}
\]

Experiments were performed at the rated torque speed of 1600 rad/min and at engine loads of 15, 30, 45 and 55 Nm, corresponding to the brake mean effective pressures (BMEP) of 0.189, 0.379, 0.568, and 0.694 MPa. The selected engine loads varied from 25% to 95% of the maximum attainable full load. Due to the high engine load has great influence on the EGR system, thus BMEP = 0.694 MPa was selected and the results was presented in this paper. Before each measurement, the engine was warmed up until the cooling water temperature varied from 80 to 85 °C while the lubricating oil temperature varied from 90 to 100 °C and the engine running steadily. The experimental uncertainty and standard errors in the measurements are shown in Table 4, which have been determined based on the method of Kline and McClelland [17]. Each test was conducted three times to ensure that the results are repeatable within the experimental uncertainties.

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3. Results and discussion

3.1. Effect of DMM and EGR on combustion characteristic

The combustion characteristics were analyzed based on the measured in-cylinder pressure. The pressure data, averaged over 400 cycles to reduce the influence of cycle-by-cycle variation, were used to obtain the heat release rate [18,19].

The cylinder pressure and heat release rate from the combustion of the diesel fuel and blended fuels at 0.694 MPa and engine speed of 1600 rad/min are presented in Fig. 2. It is observed that the curves are all bimodal in shape, indicating a rapid premixed combustion phase followed by a slower diffusion combustion phase. Compared to the diesel fuel, the peak in-cylinder pressure increases and the heat release rate in the premixed combustion phase become larger, while the curves move further away from TDC with an increase of the DMM fraction in the blends, indicating more heat is released in the premixed phase and there is a delay in the start of combustion. In general, the combustion of DMM in unmodified diesel engines results in the retarded ignition due to the differences in chemical and physical properties of the fuels. The longer ignition delay can be attributed to two reasons. Firstly, DMM has higher latent heat of evaporation than the diesel fuel. As a result, the more the DMM fraction is, the more heat is absorbed for evaporation of the blended fuel, and the in-cylinder temperature will become lower during the ignition delay period when the fuel is evaporating and mixing with air. Therefore, the ignition delay tends to be lengthened. Secondly, as shown in Table 2, the
cetane number is decreased with an increase of DMM in the blends, which also suggests longer ignition delay. Thus more fuel–air mixture is formed during the ignition delay period, leading to higher heat release rate.

Fig. 3 shows the effects of EGR (12% and 24%) on the cylinder pressure and heat release rate for DMM 20 at engine speed of 1600 rad/min and BMEP = 0.694 MPa. The figure indicates that the increase of the EGR ratio resulted in longer ignition delay and the overall combustion shifted to a later stage while the combustion duration was reduced. Both the cylinder pressure and the amount of fuel burnt in the premixed combustion phase were reduced. The shorter combustion duration is possibly related to incomplete combustion or bad fuel–air mixing. It is regarded that the introduction of EGR influences diesel engine combustion in three ways. The first is dilution mechanism: due to the dilution effect, the mixing time of air–fuel will potentially be increased, reducing flame temperature. The second is thermal mechanism: the increased heat capacity of an EGR mixture will also result in decreased flame temperature. The third is chemical mechanism: increased dissociation from the more complex EGR molecules (such as CO₂ and H₂O) results in lower flame temperatures [20].

3.2. Effect of DMM and EGR on engine performance

In the experiment, the fuel consumption for each fuel was measured. Based on the engine torque, the engine speed and the mass consumption rate of the fuel, the brake thermal efficiency (BTE) can be calculated. Fig. 4 shows the variation of BTE with EGR ratio for each fuel. In general, the BTE increases with an increase of DMM fraction in the fuel, indicating an improved combustion process. At engine speed of 1600 rad/min and BMEP = 0.694 MPa, the BTE changes from 36.4% (diesel) to 37% (20% DMM) and 37.6% (50% DMM). Regarding the effect of EGR on the engine performance, it is observed that the BTE fluctuates slightly at small EGR ratio, while it decreases evidently with further increase of EGR ratio for each fuel. Ladommatos et al. [20] concluded that when the exhaust gas is introduced into intake manifold, the increased ignition delay period was expected to result in more fuel burned in the initial uncontrolled premixed combustion phase on the one hand. Besides, the radicals in exhaust gas and higher temperature will also prompt the ignition process. In the case of small EGR ratio, the exhaust gas introduced into the combustion chamber is relatively small coupled with a higher air/fuel ratio, the BTE does not deteriorate remarkably or even becomes slightly higher. With further increase of EGR ratio, a notable deterioration in the BTE is presented which is mainly due to the reduced oxygen content available inside the cylinder.

3.3. Effect of DMM and EGR on exhaust gas emissions

Fig. 5 gives the variation of NOₓ emissions with EGR ratio for each fuel at BMEP = 0.694 MPa. It is observed that NOₓ emission increase slightly with increasing DMM in the fuel when the EGR ratio varies from 0% to 8%. With further increase of EGR ratio, the variation of NOₓ emission is not evident among the blended fuels. Several mechanisms are involved when DMM is blended with diesel fuel. Firstly, the oxygen in the fuel might enhance NOₓ formation. Moreover, the high latent heat value of DMM might lower the combustion temperature and reduce NOₓ formation. Furthermore, DMM can lead to an increase of ignition delay and heat release resulting from an increase of fuel burned in the premixed mode and hence an increase in the combustion temperature. These positive and negative effects compete with each other and could lead to the variation of NOₓ emission.

As shown in Fig. 5, NOₓ emissions decrease almost linearly with the increase of EGR ratio at the high engine load of BMEP = 0.694 Mpa. When the EGR ratio is 24%, NOₓ emission is decreased by 75%. In fact, NOₓ formation is strongly affected by the temperature inside the combustion chamber. As mentioned in Section 3.1, the combustion temperature decreases when exhaust gas is introduced into cylinder, which leads to remarkable decrease in NOₓ emissions.
The variation of CO emissions with EGR ratio for each fuel is plotted in Fig. 6. The results indicate that DMM addition could reduce CO emission to a large extent at high engine load of BMEP = 0.694 MPa in comparison with diesel fueled engine. This suggests that DMM, as an oxygenated fuel, could increase the air–fuel ratio of the local rich regions where CO is mainly formed, especially at high engine load. Moreover, the oxygen enrichment can also improve the post-flame oxidation of CO in the late diffusion combustion stage. It is observed from Fig. 6 that there is an increase of CO emissions when the EGR ratio increases at the engine speed of 1600 rad/min and BMEP = 0.694 MPa. For diesel fueled engine, CO emission increases sharply from 400 ppm to 5800 ppm corresponding to the EGR ratio increases from 0% to 24%. It is believed that the decrease of oxidation at later diffusive combustion phase is responsible for this behavior.

Fig. 7 gives HC emissions versus EGR ratio for each fuel at high engine load of BMEP = 0.694 MPa, HC emission presents an increasing trend with the increase of EGR ratio, especially in the case of high EGR ratio. The reasons are similar to those of CO emissions. Our previous experimental results [7] showed that remarkable reduction of CO emission could be achieved by increasing DMM fractions in the blended fuels at high engine loads. However, HC emission increased with the increase of oxygen mass fraction in the blended fuel, and we believe that this is because the boiling point of DMM is about 40 °C, far below that of diesel fuel. After the diesel-DMM blends are injected into the cylinder, DMM evaporates rapidly and disperses into the area before flame front reaches there. High CO and HC emissions may be an obstacle for the use of larger EGR ratio.

3.4. Effect of DMM and EGR on smoke emission and particulate number-size distribution

Fig. 8 illustrates the smoke emission versus the EGR ratio for each fuel. Compared to diesel fueled engine, the smoke emission decreases evidently with an increase of DMM in the blends which is mainly due to the oxygen content in the fuel. Moreover, the smoke emission increases with the increase of EGR ratio, especially in the case of high engine load conditions. In fact, most soot or smoke emissions results from incomplete combustion. Soot precursor formation is a competition between the rate of fuel pyrolysis and the rate of precursor oxidation, both pyrolysis and oxidation rates increase with the increase of temperature but the oxidation rate increases faster. So the oxidation rate of combustion is an important factor, which is affected significantly by the local air/fuel ratio distribution and local temperature. After introducing EGR, both air–fuel ratio and cylinder temperature are reduced, and as a result, the formation of soot or smoke is promoted.

Fig. 9 shows the relationship between NO$_x$ and smoke emission for different EGR ratio at the engine speed of 1600 rad/min and BMEP = 0.694 MPa. It is observed that smoke emission increases while NO$_x$ emission decreases with an increase of EGR ratio for all test fuels. For a specific EGR ratio, DMM 20 reduces smoke emission to some extent, while DMM 30 and DMM 50 could reduce the smoke emission remarkably and realize the double purpose of control in NO$_x$ and smoke emissions. For a specific fuel, with the EGR ratio varies from 16% to 24%, NO$_x$ emissions could be reduced significantly, whereas the smoke emission presents a sharply increase when the EGR ratio increases from 20% to 24%.
Fig. 10 illustrates the effect of DMM and EGR on the particulate number-size/mass distribution for diesel fuel and DMM 50 at the engine speed of 1600 rad/min and BMEP = 0.694 MPa. Compared to the diesel fuel, DMM 50 has lower particulate number for the geometric mean diameters \( D_i \) between 0.039 and 1.239 \( \mu \text{m} \), but has slightly higher particulate number when \( D_i < 0.039 \) \( \mu \text{m} \). It is observed also that the particulate mass distribution curves of DMM 50 are much flatter than that of pure diesel, indicating that DMM addition could significantly decrease particulate mass in the exhaust gas emissions. Compared to the diesel fuel, the maximum peak of the particulate number and mass of blended fuels shift towards the smaller diameter. In fact, because of the longer ignition delay period, more blended fuel is combusted in the premixed mode and less in the diffusion mode, leading to a reduction of particles being formed, in comparison with the diesel fuel. Besides, the carbon content in the fuel decreases with the addition of DMM in the blends, which might contribute to the reduction of soot nuclei and total particle number. Moreover, DMM has lower viscosity, higher vapor pressure and lower boiling point compared with the diesel fuel which could improve fuel atomization, leading to better air–fuel mixing. Mathis et al. [21] concluded that improved air–fuel mixture formation led to a reduction in both primary soot particle diameter and total number concentration of particles. Furthermore, the higher oxygen content of DMM can reduce soot precursor production in the fuel-rich zone, due to an increased concentration of \( O \) and \( OH \) which promote the oxidation of soot precursors to \( CO \) and \( CO_2 \), thus, the coagulation and agglomeration of the small particulates to form large particulates will be slowed down.

As shown in Fig. 10, the EGR addition increases the number and mass/size distribution of particles over the whole range of the aerodynamic diameters measured. In particular, there was a noticeable increase of the particle number mainly in the diameter range of 0.039–1.239 \( \mu \text{m} \) for both fuels. Moreover, the particulate number with the lowest aerodynamic diameter of 0.021 \( \mu \text{m} \) was almost not affected by the use of EGR for diesel fueled engine, whereas for the engine fueled with DMM 50, it shows a slight fluctuation. The retarded combustion with the use of EGR for each fuel (as shown in Fig. 3) can increase the unburned hydrocarbons in the engine combustion chamber and exhaust pipe. Under such conditions, the particles can grow in size due to condensation of the increased volatile material in the combustion product gas at the expansion stroke after the end of the combustion as well as in the engine exhaust gas pipe. Furthermore, the local air–fuel ratio is reduced and the cylinder temperature decreases after introducing EGR, which is prone to the formation of particulate.

4. Conclusion

The present study focuses on investigating fuel constituents and the exhaust gas recirculation (EGR) on combustion and exhaust gas emissions with special attention paid to the particulate mass concentration and number-size distribution. The results of the present study reveal that DMM results in longer ignition delay and longer premixed phase of combustion. Hence the peak cylinder pressure and the premixed heat release rate increase. As for the exhaust gas emissions, DMM could effectively reduce \( CO \) and smoke emissions as well as the particulate mass and total number but increase \( HC \) emissions and the nano-particulates, and it seems to be a common problem in using oxygenated fuels. Moreover, the maximum peak of the particulate number and mass shift to smaller diameter for DMM 50 compared to that of diesel fueled engine.

The increase of the EGR percentage resulted in longer ignition delay and the overall combustion shifted to a later stage while the combustion duration was reduced. Both the cylinder pressure and the amount of fuel burnt in the premixed combustion phase were reduced, thus leading to the degressive of BTE. Regarding the effect of EGR on the exhaust gaseous emissions, in general, EGR reduces \( NO_x \) emissions but increase other gaseous emissions. By the use of EGR, the total number and mass of particulate increase evidently for each fuel.

It seems that with correct oxygenate addition and EGR rate acceptable fuel efficiency and emission level can be achieved.
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