Combustion and emission characteristics of a compression ignition engine fuelled with Diesel–dimethoxy methane blends

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Abstract

The combustion and emission characteristics of a compression ignition engine fuelled with Diesel–dimethoxy methane (DMM) blends were investigated. The results showed that the initial combustion duration experienced a slight variation with the increase of DMM fraction in the fuel blends, while the rapid combustion duration and the total combustion duration decreased with the increase of DMM fraction in the fuel blends, and the crank angle of the centre of heat release curve moved closer to the top dead center. This would be due to the increase in the amount of combustible mixture for the initial premixed combustion phase because of the fast evaporation of DMM and the promotion of the subsequent diffusive combustion phase resulting from the oxygen enrichment. The maximum cylinder pressure and the maximum mean cylinder temperature showed a slight increase with the increase of DMM addition. The enrichment of oxygen by injecting the oxygenate fuel is responsible for the combustion improvement. The maximum rate of pressure rise and the maximum rate of heat release will increase with the increase of the DMM fraction in the fuel blend, and this would also be due to the increase in the amount of combustible mixture for the premixed combustion phase fuelled with the Diesel–DMM blends. A remarkable reduction in the exhaust CO and smoke can be achieved when operating on the Diesel–DMM blend. Flat NOx–smoke curves are presented when operating on the Diesel–DMM fuel blends, and a simultaneous reduction in both NOx and smoke can be realised at large DMM additions. Thermal efficiency and NOx give the highest value at 2% oxygen mass fraction (or 5% DMM volume fraction) for the combustion of Diesel–DMM blends.

Keywords: Combustion; Emissions; Diesel–dimethoxy methane blends; Compression ignition engine

1. Introduction

Reduction of engine emissions is a major research aspect in engine development with the increasing concern on environmental protection and the stringent exhaust gas regulations. It is difficult to reduce NOx and smoke...
simultaneously in normal Diesel engines due to the trade off curve between NO\textsubscript{x} and smoke. One prospective method to solve this problem is to use oxygenated alternative fuels or to add the oxygenated fuels in the Diesel fuel to provide more oxygen during combustion. In the application of pure oxygenated fuels, Fleisch et al.\cite{1}, Kapus and Ofner\cite{2} and Sorenson and Mikkelsen\cite{3} have studied dimethyl ether (DME) in a modified Diesel engine, and their results showed that the engine could achieve ultra-low emission prospects without a fundamental change in the combustion systems. Huang et al.\cite{4} investigated the combustion and emission characteristics in a compression ignition engine with DME and found that the DME engine has high thermal efficiency, short premixed combustion and fast diffusion combustion, and their work was to realize low noise, smoke free combustion. Kajitani et al.\cite{5} studied the DME engine with delaying the injection time to reduce both smoke and NO\textsubscript{x}.

Practically, adding some oxygenated compounds to fuels to reduce engine emissions without engine modification seems a more attractive proposition. Huang et al. tested the gasoline-oxygenate blends in a spark ignition engine and got satisfactory results on emission reduction\cite{6} and also investigated the combustion and emission characteristics of Diesel/dimethyl carbonate (DMC) in a compression ignition engine\cite{7}. Murayama et al.\cite{8} studied the emissions and combustion with EGR (exhaust gas recirculation) and dimethyl carbonate.

\begin{table}
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\begin{tabular}{|l|l|}
\hline
\textbf{Nomenclature} & \\
\hline
\textit{A} & wall area (m\textsuperscript{2}) \\
\textit{ATDC} & after top dead centre \\
\textit{bmeP} & brake mean effective pressure (MPa) \\
\textit{BTDC} & before top dead centre \\
\textit{C\textsubscript{p}} & constant pressure specific heat (kJ/kgK) \\
\textit{C\textsubscript{v}} & constant volume specific heat (kJ/kgK) \\
\textit{C} (wt.\%) & mass fraction of carbon in fuel blend \\
\textit{dp/d\phi} & pressure rise with crank angle (MPa/CA) \\
\textit{(dp/d\phi)}\textsubscript{max} & maximum rate of pressure rise with crank angle (MPa/CA) \\
\textit{dQ\textsubscript{p}/d\phi} & heat release rate with crank angle (kJ/CA) \\
\textit{(dQ\textsubscript{p}/d\phi)}\textsubscript{max} & maximum rate of heat release with crank angle (kJ/CA) \\
\textit{dQ\textsubscript{w}/d\phi} & heat transfer rate with crank angle (kJ/CA) \\
\textit{h\textsubscript{c}} & heat transfer coefficient (J/m\textsuperscript{2}sK) \\
\textit{H\textsubscript{u}} & lower heating value (MJ/kg) \\
\textit{H} (wt.\%) & mass fraction of hydrogen in fuel blend \\
\textit{m} & mass of cylinder gases (kg) \\
\textit{O} (wt.\%) & mass fraction of oxygen in fuel blend \\
\textit{p} & cylinder gas pressure (MPa) \\
\textit{p\textsubscript{max}} & maximum cylinder gas pressure (MPa) \\
\textit{R} & gas constant (J/kgK) \\
\textit{T} & mean gas temperature (K) \\
\textit{T\textsubscript{max}} & maximum mean gas temperature (K) \\
\textit{T\textsubscript{w}} & wall temperature (K) \\
\textit{TDC} & top dead centre \\
\textit{V} & cylinder volume (m\textsuperscript{3}) \\
\textit{\phi\textsubscript{c}} & crank angle of centre of heat release curve (CA degrees ATDC) \\
\textit{\phi\textsubscript{e}} & crank angle of heat release ending (CA degrees ADTC) \\
\textit{\phi\textsubscript{s}} & crank angle of heat release beginning (CA degrees BTDC) \\
\textit{\theta\textsubscript{ld}} & fuel delivery advance angle (CA degrees BTDC) \\
\textit{\Delta\phi\textsubscript{ic}} & initial combustion duration (CA) \\
\textit{\Delta\phi\textsubscript{rb}} & rapid combustion duration (CA) \\
\textit{\Delta\phi\textsubscript{tc}} & total combustion duration (CA) \\
\hline
\end{tabular}
\end{table}
Ajav et al. [9] studied Diesel/ethanol blends for emission reduction, and Huang et al. investigated the engine performance and emissions of a Diesel engine fuelled with Diesel/methanol blends [10]. Miyamoto et al. [11] and Akasaka and Sakurai [12] also conducted research on Diesel combustion improvement and emission reduction by the use of various types of oxygenated fuel blends.

Dimethoxy methane (DMM) has a high oxygen fraction and relatively high cetane number, and this makes it a better oxygenate additive for a Diesel/oxygenate blended fuel. Some preliminary studies revealed that the reduction of particulate emissions [13] and toxic gas pollutants could be achieved by fuelling with the Diesel/DMM blends [14]. However, those previous works just reported the emission results for a specific DMM addition and did not give the results on combustion characteristics and/or heat release analysis under various DMM additions. However, the combustion behaviors are directly linked to the engine performances and the mechanism of emission reduction operating on various fractions of Diesel–DMM blends. In order to acquire a comprehensive evaluation for Diesel–DMM blends, many aspects are still worth investigating, especially in a quantitative level. These quantitative parameters are expected to supply more information on engine combustion operating on oxygenated fuels and provide more practical measures for the improvement of combustion and reduction of emissions.

Based on the authors’ previous study, the objectives of this study will investigate the combustion characteristics based on the heat release analysis for different fractions of DMM addition in Diesel fuel and expect to increase the understanding of the combustion parameters versus the DMM fraction and/or the oxygen fraction in the fuel blends of a compression ignition engine operated on Diesel–DMM blends.

2. Test engine and fuel properties

The specifications of the test engine are listed in Table 1. In the study, Diesel fuel is the base fuel, while DMM is used as the oxygenated additive. Four fractions of the Diesel–DMM blends were designated for the study, and the volume fraction of DMM in the fuel blends are 5%, 10%, 15% and 20%, respectively. Fuel properties and the fraction of the four blends are given in Table 2, Table 3, Fig. 1 and Fig. 2, and the mass fraction of oxygen in the fuel blends ranges from 2.16% to 8.62% as shown in Fig. 2 and Table 3. The fuel properties show that DMM has a high oxygen content while the heat value and cetane number are low compared to those of pure Diesel fuel. In the experiment, the above four fuel blends with different DMM fractions were tested on the engine. Meanwhile, the engine exhaust emissions were analyzed under the same brake mean effective pressure (bmep). The resolution for NOx measurement is 1 ppm, the resolution for CO is 0.01% and the resolution for smoke measurement is 0.1 Bosch unit. Furthermore, these parameters were made in comparison with those of pure Diesel combustion in order to clarify the effect of oxygenate additives on combustion. A Horiba exhaust gas analyzer was used to measure the combustion products.

Fig. 3 gives the fuel properties of the Diesel–DMM blends versus DMM volume in the blends. The figure shows that the cetane number decreases slightly with the increase of DMM addition due to the low value of cetane number of dimethoxy methane, and this will probably lead to an increase in ignition delay of the Diesel–DMM fuel blends. The lower heating value of the blends decreases, and the heat value of evaporation increases with the increase in DMM addition, the former factor requires more fuel to be supplied for obtaining the same power output (bmep) and the latter would lead to a temperature drop of the cylinder gas due to the evaporation of the injected fuel.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Engine specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore (mm)</td>
<td>100</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>115</td>
</tr>
<tr>
<td>Displacement (cm³)</td>
<td>903</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>18</td>
</tr>
<tr>
<td>Shape of combustion chamber</td>
<td>ω shape in the bottom of bowl-in-piston</td>
</tr>
<tr>
<td>Rated power (speed)</td>
<td>11 kW/2300 rpm</td>
</tr>
<tr>
<td>Nozzle hole diameter (mm)</td>
<td>0.3</td>
</tr>
<tr>
<td>Number of nozzle holes</td>
<td>4</td>
</tr>
</tbody>
</table>
3. Instrumentation and method of calculation

The cylinder pressure was recorded by a piezoelectric transducer with the resolution of 10 Pa, and the dynamic top dead centre (TDC) was determined by motoring. The crank angle signal was obtained from an angle generating device mounted on the main shaft. 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 was outputted as the pressure data used for calculation of the combustion parameters. Bmep was calculated from the engine power, speed and configuration specifications.

A thermodynamic model is used to calculate the thermodynamic parameters in this paper. The model neglects the leakage through the piston rings [7], thus the energy conservation in the cylinder can be written as follows:

\[
\frac{dQ_B}{d\phi} - \frac{dQ_W}{d\phi} = \frac{d(mu)}{d\phi} + p \frac{dV}{d\phi} = mC_v \frac{dT}{d\phi} + p \frac{dV}{d\phi} \tag{1}
\]
The gas state equation is

$$pV = mRT \quad \text{(2)}$$

The variation of the gas state equation with crank angle is given by

$$p \frac{dV}{d\phi} + V \frac{dp}{d\phi} = mR \frac{dT}{d\phi} \quad \text{(3)}$$

The heat release rate $$\frac{dQ_B}{d\phi}$$ can be derived from Eqs. (1) and (3) as follows:

$$\frac{dQ_B}{d\phi} = p \cdot \frac{C_p}{R} \frac{V}{d\phi} + \frac{C_V}{R} \frac{dp}{d\phi} + mR \frac{dT}{d\phi} + \frac{dQ_W}{d\phi} \quad \text{(4)}$$

where the heat transfer rate is given by

$$\frac{dQ_W}{d\phi} = h_c \cdot A \cdot (T - T_w) \quad \text{(5)}$$

The heat transfer coefficient $$h_c$$ uses the correlation formula given by Woschni [15]. $$C_p$$ and $$C_V$$ are temperature dependent parameters whose formulae are given in Ref. [15].

The primary data is the cylinder pressure-crank angle data. Using these primary data and the above formulae, the maximum cylinder gas pressure $$p_{\text{max}}$$, the mean gas temperature $$T_{\text{max}}$$, the rate of pressure rise ($$\frac{dp}{d\phi}$$), the rate of heat release ($$\frac{dQ_B}{d\phi}$$), the maximum rate of pressure rise ($$\frac{dp}{d\phi}_{\text{max}}$$) and the maximum rate of heat release ($$\frac{dQ_B}{d\phi}_{\text{max}}$$) can be calculated.

The initial combustion duration $$\Delta \phi_{\text{ic}}$$ is defined as the crank angle interval from the ignition start to that where 10% of the heat release is reached; the rapid combustion duration $$\Delta \phi_{\text{rb}}$$ is defined as the crank angle interval from the crank angle where 10% of the heat release has occurred to the crank angle where 90% of the heat release is reached; and the total combustion duration $$\Delta \phi_{\text{tc}}$$ is the crank angle interval from the beginning of heat release (crank angle of heat release curve starts rising after fuel injection) to the ending of heat release (crank angle of heat release curve falls into zero). The premixed combustion phase refers to the combustion of the combustible mixture prepared within the ignition delay period, while the diffusive combustion phase refers to the subsequent mixing controlled combustion period. The crank angle of the centre of the heat release curve is determined by the following formula:

$$\varphi_c = \frac{\int_{\varphi_s}^{\varphi_e} \frac{dQ_B}{d\phi} \cdot \varphi \cdot d\phi}{\int_{\varphi_s}^{\varphi_e} \frac{dQ_B}{d\phi} \cdot d\phi} \quad \text{(6)}$$

in which $$\varphi_s$$ is the crank angle at the beginning of heat release, and $$\varphi_e$$ is the crank angle at the end of heat release.

4. Results and discussion

Fig. 4 gives the combustion duration of the Diesel–DMM blends versus the DMM volume fraction in the fuel blends under various engine loads (bmeP). In the case of the initial combustion duration $$\Delta \phi_{\text{ic}}$$, a slight variation of $$\Delta \phi_{\text{ic}}$$ versus the fraction of DMM addition is observed for the specific bmeP at both high engine speed (1600 rpm) and low engine speed (1200 rpm), see Fig. 4a and b. However, a different behavior of $$\Delta \phi_{\text{ic}}$$ versus the engine load is shown at high and low engine speeds, that is, a slight difference is presented at low engine speed (1200 rpm), while the initial combustion duration shows an increase with the increase of engine load at high engine speed (1600 rpm). The following interpretation can explain this behavior. With the increase of engine speed, the amount of fuel for 10% of the heat release will increase, and this will subsequently increase the duration to burn this fraction of fuel. On the other hand, a high load will increase the cylinder gas temperature, and this will supply the environment for increasing the combustion speed. The combined effects of these two factors would make the initial combustion duration increase with the increase of engine load. For a given duration of time (ms), the duration in crank angle will increase with the increase of engine speed, and a
relative large difference in the initial combustion duration under different engine loads is presented at the high engine speed. The results indicated that under the same bmep condition, the addition of DMM in the Diesel fuel had little influence on the initial combustion duration.

In the case of the rapid combustion duration $\Delta\varphi_{rc}$, a slight decrease of $\Delta\varphi_{rc}$ is observed with the increase of DMM in the fuel blends at both high (1800 rpm) and low (1200 rpm) engine speeds, see Fig. 4c and d, while $\Delta\varphi_{rc}$ shows an increase with the increase of engine load. Increasing the DMM fraction in the fuel blend will increase the oxygen fraction in the fuel blend, and this will contribute to improvement of the combustion, especially for improvement of the diffusive combustion phase where the enrichment of oxygen will promote the combustion process, indicating a better advantage by oxygenate fuels in Diesel fuel. The long duration of $\Delta\varphi_{rc}$ at high engine load would be due to the increase in fuel amount and the increase of the duration for reaching 90% of the heat release.
The total combustion duration $\Delta \varphi_{tc}$ versus DMM fraction in the fuel blend, Fig. 4e and f, exhibits a similar behavior to that of the rapid combustion duration, the shortening of the rapid combustion duration is considered to contribute to the decrease of the total combustion duration. Thus, the results indicate that adding the oxygenate fuel in the Diesel fuel can shorten the combustion duration due to the oxygen enrichment, and it is beneficial to the increase of engine thermal efficiency.

Fig. 5 illustrates the crank angle of the centre of heat release curve $\varphi_c$ for the Diesel–DMM fuel blends. The figure also shows the decrease of $\varphi_c$ with the increase of DMM addition. As explained above, the decrease in combustion duration and the improvement of the diffusive combustion phase will bring the heat release process closer to top dead centre.

The maximum cylinder pressure ($p_{\text{max}}$) and the maximum mean cylinder temperature ($T_{\text{max}}$) versus DMM fraction in the fuel blend is plotted in Fig. 6. For a given engine load (bmeP), the maximum cylinder pressure and the maximum mean gas temperature show a slight increase with an increase of the DMM addition. The amount of blend will increase with the increase of the DMM fraction in the fuel blend in order to get the same-bmeP, thus more combustible mixture will be available for the premixed combustion phase due to the low boiling point of the DMM. Moreover, the enrichment of oxygen by injecting the oxygenate fuel will also promote the combustion. The results suggest that Diesel–DMM blends would be helpful for the reduction of engine exhaust CO and smoke, since it could increase the in cylinder gas temperature and oxygen enrichment.

Fig. 7 gives the maximum rate of pressure rise ($dp/d\varphi_{\text{max}}$) and the maximum rate of heat release ($dQ_B/d\varphi_{\text{max}}$) versus the DMM fraction in the fuel blend. In the case of an engine speed of 1200 rpm, ($dp/d\varphi_{\text{max}}$) and ($dQ_B/d\varphi_{\text{max}}$) show an increase with the increase of DMM fraction in the fuel blend at all engine loads except for the lowest engine load (bmeP = 0.14 MPa). The low temperature and cylinder swirl under the small engine load is considered to be responsible for this behavior. In the case of an engine speed of 1600 rpm, ($dp/d\varphi_{\text{max}}$) and ($dQ_B/d\varphi_{\text{max}}$) show an increase with the increase of DMM fraction in the fuel blend, and little difference is observed for the engine loads between bmeP = 0.70 MPa and bmeP = 0.28 MPa while a large difference is
observed between bmep = 0.14 MPa and the other loads. High cylinder swirl at high engine speed will increase the amount of combustible mixture for the premixed combustion phase and result in the increase of \((dp/d\varphi)_{\text{max}}\) and \((dQ_b/d\varphi)_{\text{max}}\) with the increase of the DMM addition.

The NO\(_x\) concentration of Diesel/DMM blends under various brake mean effective pressures (bmePs) is shown in Fig. 8. The behavior of NO\(_x\) versus the oxygen mass fraction gives a similar trend under all engine loads and engine speeds, while NO\(_x\) reaches its highest value at 2% oxygen mass fraction (or 5% DMM volume fraction), indicating the improvement of combustion at this fraction of DMM addition. NO\(_x\) slightly decreases with the increase of DMM in the Diesel fuel, and the decrease in heating value and the increase in fuel injection duration of the blends is contributed to the lowering of the NO\(_x\) concentration. Generally speaking, for a given bmep, NO\(_x\) shows little variation with the addition of DMM in the Diesel fuel, and this is consistent with the results obtained by Yeh et al. [16] whose work showed that little or no change in NO\(_x\) emissions was observed with the oxygenated fuels.

Fig. 9 shows the exhaust CO concentration of the Diesel/DMM blends and its reduction rate versus the oxygen mass fraction at various brake mean effective pressures (bmePs). At all engine loads (bmePs) and engine speeds, the CO concentration shows a decrease with the increase of the oxygen mass fraction (DMM mass fraction), indicating the improvement of combustion at this fraction of DMM addition. NO\(_x\) shows little variation with the addition of DMM in the Diesel fuel, and this is consistent with the results obtained by Yeh et al. [16] whose work showed that little or no change in NO\(_x\) emissions was observed with the oxygenated fuels.

Fig. 6. \(p_{\text{max}}\) and \(T_{\text{max}}\) of the Diesel/DMM fuel blends.
Fig. 7. \((\frac{dp}{d\phi})_{\text{max}}\) and \((\frac{dQ_B}{d\phi})_{\text{max}}\) of the Diesel/DMM fuel blends.

Fig. 8. Exhaust NO\(_x\) concentration of the fuel blends.
The smoke and its reduction rate for the Diesel–DMM blends at various brake mean effective pressures (bmep) are illustrated in Fig. 10. The purpose of using the oxygen containing fuel blend is that it is expected to decrease the engine smoke by supplying more oxygen to make it burn more completely. The results clearly showed that the engine smoke could be decreased remarkably with the addition of dimethoxy meth-ane in the Diesel fuel at all engine speeds and engine loads, and this is reasonable since the oxygen containing fuel blends can reduce the rich spray regions and increase the post-flame oxidation of the formed soot. The results also showed 50–60% reduction in smoke could be achieved when the oxygen mass fraction in the blends reached 10% by adding DMM in the Diesel fuel, and this is consistent with the results obtained by Ullman et al.\[17\] whose study showed that smoke could be reduced by about 5% for each 1% increase in oxygenate addition. The reduction rate of smoke is more obvious at low engine speed. The combustion under low swirl intensity depends largely on the air–fuel ratio in the cylinder during combustion, and the oxygen enrichment will be beneficial to decrease the rich mixture region and decrease smoke formation in the cylinder. Furthermore, the long residence time during the engine expansion and exhaust processes at low engine speed will promote the post-flame oxidation of the combustion formed smoke and decrease the exhaust smoke level.

Fig. 11 gives the exhaust smoke versus engine speed under high and middle loads. The results show that the behavior of smoke versus engine speed for the Diesel/DMM blends showed a similar trend to that of Diesel combustion, that is, exhaust smoke increases with the increase in engine speed. The short time for fuel evaporation and mixing at high engine speed causes more rich mixture zones in the combustion chamber and increases the smoke formation during combustion, meanwhile, the short time of the expansion and exhaust processes at high engine speed restricts the oxidation of smoke in the post-flame process and gives a high exhaust smoke level.

The relationship between NO\textsubscript{x} and smoke of the Diesel/DMM fuel blends at various brake mean effective pressures (bmep) is plotted in Fig. 12. Unlike the engine operating on pure Diesel fuel, which has a trade off
Fig. 10. Engine exhaust smoke and its reduction rate of the fuel blends.

Fig. 11. Exhaust smoke versus engine speed for the fuel blends.
behavior between NO\textsubscript{x} and smoke, a flat NO\textsubscript{x}/smoke trade off curve is presented when operating on the Diesel/DMM fuel blends. Even a reduction in NO\textsubscript{x} and smoke together is observed with a large addition of DMM in the Diesel fuel, and this means a simultaneous reduction in both smoke and NO\textsubscript{x} can be realised when operating on the Diesel/DMM blends. Since the blends are oxygen containing fuels and have a high tolerance for exhaust gas recirculation, the combination of Diesel/DMM blends with exhaust gas recirculation can make a further decrease of NO\textsubscript{x} without increasing the smoke emission.

5. Conclusions

The combustion and emission characteristics of a compression ignition engine fuelled with Diesel–dimethoxy methane (DMM) blends were investigated, and the main results are summarized as follows:

1. The initial combustion duration shows a slight variation with the increase of the DMM fraction in the fuel blend, while the rapid combustion duration, the total combustion duration and the crank angle of the centre of heat release curve decrease with the increase of the DMM fraction in the fuel blend due to the improvement of the diffusive combustion phase.

2. The maximum cylinder pressure and the maximum mean cylinder temperature give a slight increase with the increase of the DMM addition. The enrichment of oxygen by injecting the oxygenate fuel is responsible for the combustion improvement.

3. The maximum rate of pressure rise and the maximum rate of heat release will increase with the increase of the DMM fraction in the fuel blends. This would be due to the increase in the amount of combustible mixture for the premixed combustion phase.
4. A remarkable reduction in the exhaust CO and smoke is realised when operating on the Diesel/DMM blends. Flat NOx/smoke and thermal efficiency/smoke curves are presented when operating on the Diesel/DMM fuel blends and even a simultaneous reduction in both NOx and smoke can be realised at large DMM additions.

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