Combustion characteristics of a compression ignition engine fuelled with diesel–ethanol blends

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Abstract: Combustion characteristics were analysed in a compression ignition engine fuelled with diesel–ethanol blends with and without a cetane number improver. The results show that, for the same brake mean effective pressure and engine speed, the maximum cylinder pressure \( P_{\text{max}} \), the ignition delay, the premixed combustion duration, and the fraction of heat release in premixed combustion phase will increase, while the diffusive combustion duration, the fraction of diffusive combustion phase, and the total combustion duration decrease with increase in the ethanol fraction in the blends. Meanwhile, the centre of the heat release curve moves close to the top dead centre, and the maximum rate \( \left( \frac{dQ_b}{dQ_\theta} \right)_{\text{max}} \) of heat release and the maximum rate \( \left( \frac{dp}{dQ_\theta} \right)_{\text{max}} \) of pressure rise increase with increase in the ethanol fraction in the blends. The addition of a cetane number improver is beneficial to the decrease in the ignition delay, the cylinder peak pressure, the maximum rate of pressure rise, and the combustion noise when operating on diesel–ethanol blends.

Keywords: combustion, diesel, ethanol, blends, diesel engine

1 INTRODUCTION

Reduction in engine emissions is a major research aspect in engine development with increasing concern about environmental protection and stringent exhaust gas regulations. It is difficult to reduce nitrogen oxides (NO\(_x\)) and smoke simultaneously in a normal diesel engine because of the trade-off curve between NO\(_x\) and smoke. Promising methods to solve this problem are to use a clean fuel such as natural gas [1, 2], an oxygenated alternative fuel such as alcohol or dimethyl ether (DME) [3–5], or to add oxygenated fuels to diesel fuel to provide more oxygen during combustion. It is well known that diesel engine combustion produces particulate matter (PM), in which the fine particulates are believed to be the main factor accounting for problems of the human respiratory tract. The fine particles most likely to cause adverse health effects are PM10 and PM2.5 (particles with an aerodynamic diameter smaller than 10 \( \mu \text{m} \) and 2.5 \( \mu \text{m} \) respectively). Almost all fine particulates are generated as a result of combustion processes, diesel-fuelled engine combustion, and various industrial processes. PM can be reduced when sufficient oxygen is available in the combustion chamber; thus utilization of oxygen-containing fuels in diesel engines is expected to decrease PM10 and PM2.5.

In the utilization of pure oxygenated fuels, Fleisch et al. [3], Kapus and Ofner [4], and Sorenson and Mikkelsen [5] have studied DME in the modified diesel engine, and their results showed that the engine could achieve ultra-low emissions without a fundamental change in combustion system. Huang et al. [6] investigated the combustion and emission characteristics in a compression ignition engine with DME and found that the engine had a high thermal efficiency, a short premixed combustion phase, and a fast diffusive combustion phase; its advantage was low noise with smoke-free combustion. Kajitani et al. [7] studied the DME-fuelled engine with retarded injection timing to reduce both smoke and NO\(_x\).

Practically, adding some oxygenated compounds to diesel fuel to reduce engine emissions without

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engine modification seems to be more attractive. Huang et al. [8] tested gasoline–oxygenate blends in a spark ignition engine and achieved satisfactory results in emission reduction; they [9] also investigated the combustion and emission characteristics of diesel–dimethyl carbonate (DMC) in a compression ignition engine [9]. Wang et al. [10] conducted research on the combustion of oxygenated fuel with exhaust gas recirculation (EGR) while Murayama et al. [11] studied the emissions and combustion with EGR and DMC. In the study, they used the external EGR rate which was measured by determining the ratio of CO₂ concentration in the intake port to that in the exhaust port. Huang et al. [12] studied the combustion and emission characteristics of a compression ignition engine fuelled with diesel–dimethoxy methane blends. Bertoli et al. [13], Miyamoto et al. [14], and Akasaka and Sakurai [15] also conducted research on diesel combustion improvement and emission reduction by using various types of oxygenated fuel blend.

Ethanol is regarded as one of the promising alternative fuels or an oxygen additive in diesel engines with its advantages of low price and high oxygen fraction. However, the addition of ethanol to diesel will increase the ignition delay and combustion noise; therefore, a small addition of a cetane number (CN) improver is helpful to reduce the combustion noise. Some previous work was conducted on the utilization of diesel–ethanol blends in a compression ignition engine [16–20]. However, these studies mainly concentrated on engine performance and emissions when using ethanol–diesel blends; little work deals with the combustion parameter analysis based on the heat release process, such as combustion phase analysis and combustion duration analysis, with and without a CN improver. The clarification of these parameters can supply more information on the combustion mechanisms of an engine fuelled with diesel–ethanol blends and can provide practical measures for engine operation.

The objective of this paper is to investigate the oxygenated fuel combustion characteristics by adding ethanol to diesel fuel with and without a CN improver and to clarify the relationship between the combustion parameters and oxygen fraction in the blends.

### 2 TEST ENGINE AND FUEL PROPERTIES

The test engine is a single-cylinder direct-injection diesel engine made at Chinese Nan Yang Diesel Engine Ltd. The specifications of the test engine are listed in Table 1. In the study, diesel fuel is the baseline fuel while ethanol is used as the oxygenated additive. Four fractions of the diesel–ethanol blends were designated for the study, and the volume fractions of ethanol in the fuel blends are 5 per cent (E5), 10 per cent (E10), 15 per cent (E15), and 20 per cent (E20) respectively. Meanwhile, 0.2 per cent of CN improver (iso-amy l nitrite) was added to the oxygenated blends to increase their CN, and the corresponding blends are designated E5A, E10A, E15A, and E20A respectively. The fuel properties and the fractions of oxygenated blends are given in Table 2 and Table 3 respectively; the mass fraction of oxygen in the fuel blends ranges from 2.26 per cent to 7.62 per cent, as shown in Table 3. The fuel properties show that ethanol has a high oxygen content while the heat value and CN are low compared with those of pure diesel fuel. In the experiment, the above fuel blends with different ethanol fractions were tested on the engine. Needle-lift-measuring equipment was mounted in the injector to measure the needle movement, and a Horiba exhaust gas analyser was used to measure the combustion products.

### 3 RESULTS AND DISCUSSION

Figure 1 shows the cylinder pressure of diesel–ethanol blends with and without a CN improver. It
can be seen that the peak value of cylinder pressure increases with increase in the ethanol fraction in the blends. The addition of a CN improver decreases the peak value of the cylinder pressure compared with that without a CN improver for the same ethanol fraction. The addition of ethanol to diesel fuel decreases the CN of the blends, increasing the ignition delay and the amount of combustible mixture available within the ignition delay period and subsequently increasing the amount of fuel burned in the premixed burning phase, which increases the rate of pressure rise and combustion noise. Previous work also revealed that cycle-by-cycle variations, vibration, and wearing increase when an engine operated on diesel–ethanol blends [16]. The addition of a CN improver to diesel–ethanol blends can shorten the ignition delay, decreasing the maximum value of cylinder pressure and rate of pressure rise, which reduces combustion noise and cycle-by-cycle variations.

Figure 2 gives the heat release curves of diesel–ethanol blends with and without a CN improver at two engine speeds. It is shown that the initial stage of the heat release process is postponed while the maximum rate of heat release increases with increase in the ethanol fraction in the blends. Because of the low CN of ethanol, the CN of oxygenated blends will decrease with increase in the ethanol fraction, increasing the ignition delay period and the amount of combustible mixture during the ignition delay period, which results in an increase in the fuel burned in the premixed burning phase. Moreover, the high volatility of ethanol makes more ethanol evaporate, and this will also increase the amount of combustible mixture within the ignition delay period. The greater the ethanol fraction, the larger the amount of combustible mixture will be. However, the addition of a small amount of a CN improver to diesel–ethanol blends can decrease the ignition delay phenomenon and reduce the peak value of the heat release rate. This would be due to the improvement in the ignition of the blends by adding a CN improver, reducing the ignition delay and amount of combustible mixture in the premixed burning phase.

Figure 3 illustrates the ignition delay versus oxygen mass fractions for diesel–ethanol blends with and without a CN improver. The ignition delay is defined as the time interval from the start of injection to the start of combustion. In this paper, the crank angle where the needle begins to lift is

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Mass fractions (wt %) of the following in the fuels</th>
<th>Lower heating value $H_l$ (MJ/kg)</th>
<th>Heat of evaporation (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E0</td>
<td>Carbon 85.60, Hydrogen 14.00, Oxygen 0.4</td>
<td>42.50</td>
<td>260</td>
</tr>
<tr>
<td>E5</td>
<td>Carbon 83.80, Hydrogen 13.95, Oxygen 2.25</td>
<td>41.78</td>
<td>288</td>
</tr>
<tr>
<td>E5A</td>
<td>Carbon 83.80, Hydrogen 12.95, Oxygen 2.25</td>
<td>41.78</td>
<td>296</td>
</tr>
<tr>
<td>E10</td>
<td>Carbon 82.02, Hydrogen 13.91, Oxygen 4.07</td>
<td>41.05</td>
<td>317</td>
</tr>
<tr>
<td>E10A</td>
<td>Carbon 82.02, Hydrogen 13.91, Oxygen 4.07</td>
<td>41.05</td>
<td>317</td>
</tr>
<tr>
<td>E15</td>
<td>Carbon 80.29, Hydrogen 13.86, Oxygen 5.85</td>
<td>40.31</td>
<td>346</td>
</tr>
<tr>
<td>E15A</td>
<td>Carbon 80.29, Hydrogen 13.86, Oxygen 5.85</td>
<td>40.31</td>
<td>346</td>
</tr>
<tr>
<td>E20</td>
<td>Carbon 78.57, Hydrogen 13.81, Oxygen 7.62</td>
<td>39.57</td>
<td>375</td>
</tr>
<tr>
<td>E20A</td>
<td>Carbon 78.57, Hydrogen 13.81, Oxygen 7.62</td>
<td>39.57</td>
<td>375</td>
</tr>
</tbody>
</table>

Fig. 1 Cylinder pressure curves of the blends (CA, crank angle; BTDC, before top dead centre)
regarded as the timing of the start of injection and the crank angle corresponding to the shifting position in the heat release rate curve is considered as the start of combustion. Figure 3 shows that the ignition delay of diesel–ethanol blends is increased with increase in the ethanol or oxygen fraction. The high evaporation heat value and the very low CN of ethanol will lead to the slowdown in the chemical reaction rate at the early stage and the increase in the ignition delay. Moreover, it is suggested that diesel fuel would probably first burn and then ignite the surrounding ethanol fuel. However, the addition of a CN improver can shorten the ignition delay of the blends, and this is more obvious on the addition of a large fraction of ethanol. The addition of a CN improver will increase the CN of the blends, and the influence is more effective for a large ethanol fraction where the CN is low \cite{21}. Thus, the addition of a CN improver in the case of a greater ethanol fraction will play a large role in reducing the ignition delay.

Figure 4 shows the premixed combustion duration of diesel–ethanol blends with and without a CN improver. In this paper, the premixed combustion duration is defined as the crank angle from the beginning of heat release to the crank angle of the valley position between the two peaks in the heat release curve. The results show that the premixed combustion duration increases with increase in the oxygen or ethanol mass fraction, and a relatively long duration in premixed combustion phase is presented under the low-load condition. However, the premixed combustion duration shows a slight increase with increase in the oxygen mass fraction after the addition of a CN improver. As the CN of diesel–ethanol blends increases on the addition of a CN improver, the increase in ignition delay of the blends is lower compared with that without a CN improver. Nevertheless, the high volatility of ethanol will increase the amount of combustible mixture during the ignition delay period, and this will give an increasing trend of premixed combustion duration.

**Fig. 2** Heat release rate of the blends (CA, crank angle; BTDC, before top dead centre)

**Fig. 4** Premixed combustion duration of diesel–ethanol blends with and without a CN improver.
with increase in the oxygen fraction; the trend is flat on the addition of a CN improver.

Figure 5 gives the maximum heat release rate \( \left( \frac{dQ}{d\phi} \right)_{\text{max}} \) versus oxygen mass fraction in the blends. It is shown that the maximum heat release rate increases with increase in the oxygen mass fraction in the blends. The behaviour is more obvious for middle and high loads while it gives a slight increase at low loads. Ethanol is oxygenated fuel, and so oxygen in the fuel can promote combustion of the mixture. Meanwhile, it is well known that the combustion rate has an exponential function relationship with temperature [22]. At low loads, the gas temperature is lower than at middle and high loads, and so the combustion promotion through oxygenated fuel is suppressed. In the case of the addition of a CN improver, the maximum heat release rate versus oxygen mass fraction shows a flat trend at all loads; the decrease in ignition delay and amount of combustible mixture during the ignition delay period by adding a CN improver is responsible for this.

Figure 6 shows the premixed combustion release heat versus oxygen mass fraction in the blends. The amount of release heat in the premixed combustion phase is the accumulated heat release in the premixed combustion duration. It is shown that premixed combustion heat release increases with increase in the ethanol mass fraction in the blends, and this would be due to the increase in ignition delay and combustible mixture available during the ignition delay period. The addition of ethanol to diesel fuel will increase the ignition delay and the amount of combustible mixture therein; moreover, the high volatility of ethanol will provide more evaporated fuel within the ignition delay, increasing the amount of combustible mixture therein; all these factors cause the increase in the premixed combustion heat release with increase in the ethanol.
fraction in the blends. On addition of a CN improver to diesel–ethanol blends, the ignition delay is decreased compared with that without a CN improver, decreasing the amount of combustible mixture within the ignition delay period, and resulting in a decrease in the premixed combustion heat release compared with that without a CN improver. This reveals that the amount of premixed combustion heat release for diesel–ethanol blends can be controlled by adding a CN improver to the blends.

The diffusive combustion duration versus oxygen mass fraction is plotted in Fig. 7. In this paper, the diffusive combustion duration is defined as the crank angle from the end of the premixed combustion phase to the end of combustion. The results show that the diffusive combustion duration of diesel–ethanol blends increases with increase in the engine load regardless of the addition of a CN improver. For a specific engine load (brake mean effective pressure (b.m.e.p.)), the diffusive combustion duration of the blends decreases with increase in the oxygen mass fraction. As explained above, the addition of ethanol increases the fraction of fuel burned in the premixed combustion phase, and this will decrease the fraction of fuel burned in the subsequent diffusive combustion phase, thereby reducing the diffusive combustion duration. In addition to this, oxygenated fuel can reduce the over-rich mixture region, improving the mixture quality and increasing burning rate, and this is more effective for the diffusive combustion phase where oxygen is insufficient. Thus, oxygenated fuel can promote diffusive combustion and reduce soot formation.

Figure 8 gives the diffusive combustion release heat versus oxygen mass fraction in the blends. Diffusive combustion heat release decreases with increase in the ethanol or oxygen mass fraction. A relatively slight decrease in the diffusive combustion heat release (dp/dφ)max of the blends (CA, crank angle; BTDC, before top dead centre)
heat release is observed in the case of the addition of a CN improver compared with that without a CN improver. The reduction in diffusive combustion duration and amount of fuel burned in the diffusive combustion duration is responsible for this. While the addition of a CN improver in the blends can relatively decrease the premixed combustion duration and the amount of fuel burned in the premixed combustion duration, this will relatively increase the amount of fuel burned in the subsequent diffusive combustion duration.

Figure 7 shows the total combustion duration versus oxygen mass fraction for diesel–ethanol blends. For specific engine load (b.m.e.p.), the total combustion duration decreases with increase in the ethanol or oxygen mass fraction with and without a CN improver. In one respect, the addition of ethanol to diesel decreases the heating value of the blends; much more fuel should be injected for oxygenated fuel to achieve the same engine load (b.m.e.p.). In other respects, oxygenated fuel can promote the combustion rate and especially the diffusive combustion rate. The comprehensive influence of the above two factors results in a decrease in the total combustion duration with increase in the ethanol or oxygen mass fraction.

Figure 10 gives the crank angle for the centre of the heat release curve $Q_c$. $Q_c$ will move close to the top dead centre (TDC) with increase of ethanol or oxygen mass fraction in the blends. This is reasonable since the amount of premixed combustion increases and the amount of diffusive combustion decreases. Furthermore, decreasing the diffusive combustion duration and increasing the maximum heat release rate of the premixed combustion phase will contribute to the variation in $Q_c$. The study also shows that a CN improver has little influence on $Q_c$.

Figure 11 shows the maximum rate $(dp/d\varphi)_{\text{max}}$ of pressure rise versus ethanol or oxygen mass fraction in the blends. $(dp/d\varphi)_{\text{max}}$ increases with increase in
the oxygen mass fraction and this means that the combustion noise will increase for diesel–ethanol blends. However, the addition of a CN improver can suppress the increasing rate of \( \frac{dp}{d\phi} \) and this is beneficial for controlling the combustion noise when operating on diesel–ethanol blends.

**4 CONCLUSIONS**

Combustion characteristics were analysed in a compression ignition engine fuelled with diesel–ethanol blends with and without a CN improver. The main results are summarized as follows.

1. For the same b.m.e.p. and engine speed, the maximum cylinder pressure \( P_{\text{max}} \), the ignition delay, the premixed combustion duration, and the fraction of heat release in the premixed combustion phase increase, while the diffusive combustion duration, the fraction of diffusive combustion phase, and the total combustion duration decrease with increase in the ethanol fraction in the blends.

2. The centre of the heat release curve moves close to the TDC; the maximum rate of heat release and the maximum rate of pressure rise increase with increase in the ethanol fraction in the blends.

3. The addition of a CN improver is beneficial to the decrease in the ignition delay, cylinder peak pressure, maximum rate of pressure rise, and combustion noise when operating on diesel–ethanol blends.

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REFERENCES


**APPENDIX**

**Notation**

- ATDC: after top dead centre
- b.m.e.p.: brake mean effective pressure (MPa)
- BTDC: before top dead centre
- CN: cetane number
- $\frac{dp}{d\phi}_{\text{max}}$: maximum rate of pressure rise
- $\frac{dQ_b}{d\phi}$: heat release rate (kJ/crank angle)
- $\frac{dQ_b}{d\phi}_{\text{max}}$: maximum rate of heat release
- $H_l$: lower heating value (MJ/kg)
- TDC: top dead centre
- $\theta_{\text{fd}}$: fuel delivery advance angle
- $\phi_c$: crank angle of the centre of the heat release curve (deg crank angle after top dead centre)