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Dependency of engine combustion on blending ratio variations of lipase-catalysed coconut oil biodiesel and petroleum diesel

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Dependency of engine combustion on blending ratio variations of lipase-catalysed coconut oil biodiesel and petroleum diesel

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Abstract

From our previous study about coconut oil-based biodiesel blended with petroleum diesel at a ratio of 1:9 (B10), it was found that the lipase catalysed ethyl ester B10 can achieve not only simultaneous reduction of smoke and NO_x emissions but also improved brake power compared to petroleum diesel. This paper presents engine performance of this biodiesel fuel at higher blending ratios with the expectation of further improved emissions and brake power. The experiments were performed in a single-cylinder light-duty diesel engine equipped with a common-rail injection system. Prior to the engine performance and emissions testing, the fuel injection rate measurement was conducted for various biodiesel blending ratios to find the injected fuel mass for the same total energy of 1080 J, considering 6% lower calorific value of the tested biodiesel than that of petroleum diesel. The engine experiments were performed at fixed engine speed of 2000 rpm and common-rail pressure of 130 MPa. In addition to the variations of biodiesel blending ratio, the injection timing was also swept from 13 to 3 crank angle degrees before top dead centre to evaluate combustion of biodiesel blends at various combustion phasing conditions. The in-cylinder pressure traces were measured using a piezo-electric pressure transducer, which was used to calculate key performance parameters such as the indicated mean effective pressure (IMEP), apparent heat release rate (aHRR), and burn duration. The brake MEP (BMEP) was also calculated using the measured brake torque from the eddy current (EC) dynamometer and subsequently the friction MEP (FMEP) was obtained. From the engine tests, it is found that a higher biodiesel blending ratio results in decreased IMEP because the lower calorific value of coconut oil-based biodiesel and overall leaner mixture condition cause the decreased diffusion flame temperature and extended burn duration. The improved lubricity of coconut oil biodiesel and hence reduced friction losses, however, leads to similar BMEP of petroleum diesel even for high biodiesel blends. Nevertheless, a significant increase in the brake specific fuel consumption is unavoidable at high biodiesel blending ratios. From the engine-out emission measurements, a significant reduction of smoke emissions were observed with an increase in the biodiesel blending ratio, which is explained by the oxygenated molecular structures and reduced aromatics contents of biodiesel. Also, the slower reaction and leaner mixture of high biodiesel blends, together with shorter carbon chain length of coconut oil-based biodiesel, cause the reduced flame temperature and thereby decreasing NO_x emissions. Therefore, the high biodiesel blends using coconut oil feed stock is very promising to overcome the smoke-NO_x trade-off of petroleum diesel. When both the brake specific fuel consumption and smoke/NO_x emissions are considered, however, the optimised biodiesel blending ratio of the tested conditions of this study is found at low B10.

Keywords: Biodiesel, Blending ratio, Coconut oil, Common-rail diesel engine

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1. Introduction

Biodiesel produced from rapeseed or sunflower seed in European Union [1] and soybean oil in the United States [2] penetrates into the market thanks to its renewable nature and capacity to lower greenhouse gas emissions compared to those of petroleum diesel. As of 2011, the largest biodiesel market is EU which accounts for 44% of the worldwide biodiesel production [3]. One study predicted that the global production of biodiesel will increase from 24 billion litres to 42 billion litres between 2011 and 2020 [4], which is largely driven by European government's mandate on biofuel. For example, EU requires that 10% of the total transport fuel supply should be from bio-feedstock by 2020 [5]. Due to the hardware compatibility issue, however, neat biodiesel (i.e. B100) is rare but blends of biodiesel and petroleum diesel are available in fuel stations (e.g. B2, B5 and B20 [6]). Various tests suggest that blends of 20% biodiesel (B20) and lower can be used in the existing diesel equipment with no hardware modification [7]; however, to meet the government's mandates, higher blending ratio **biodiesel fuels** might be required.

There are well known advantages of higher biodiesel blends in engine-out emissions. For example, many studies reported reduced smoke emissions with the increasing biodiesel blending ratio [8-16]. This is because biodiesel is an oxygenated fuel with a reduced amount of carbon compared to petroleum diesel. **Oxygenated fuels yield less soot formation** [17] and higher soot oxidation rates [8, 10, 18], leading to reduced engine-out smoke emissions. **Although the increased emissions of soluble organic fractions (SOF) raise a new issue [19], the significance decrease in particulate matter emissions is a major advantage of biodiesel fuels.** Similarly, the unburned hydrocarbon (uHC) and carbon monoxide (CO) emissions are reduced due to the enhanced oxidation [11, 12, 20, 21]. Some studies, however, reported that the increased viscosity of biodiesel makes a negative impact on fuel atomisation [22], which can lead to increased uHC emissions [9].

Many studies have shown that, **despite similar thermal efficiency**, the engine brake power decreases with an increasing biodiesel blending ratio due to a lower calorific value of biodiesel than that of petroleum diesel [23, 24]. This means higher brake specific fuel consumption (BSFC) for higher blending ratio **biodiesel fuels**, which is a **disadvantage of using biodiesel in diesel engines** [8-11, 20, 23-25]. Another issue is the increased engine-out emissions of the oxides of nitrogen (NO_x) for higher biodiesel blends [9, 11, 14, 20, 25-30]. **From their optical diagnostics performed in a heavy-duty diesel engine to clarify the origin of increased NO_x emissions of biodiesel, Mueller et al. [31] suggested that the charge-gas mixture conditions near the flame base play a major role such that the mixtures closer to the stoichiometric at ignition cause higher local and average in-cylinder temperatures and thereby increasing thermal NO formation. From in-cylinder pressure measurements, the increased thermal NO formation shows a good correspondence with the higher peak rate of heat release [30, 32], which was used to explain the increased NO_x emissions in many biodiesel studies [20, 25, 30]. In addition, previous studies focused on fuel molecular structures reported that the flame temperature of biodiesel fuels increases with increasing carbon chain length and unsaturation degree of fatty acids [30, 33]. Therefore, it could be thought that the engine tests reporting increased NO_x emissions were conducted at charge-gas mixtures leading to higher in-cylinder temperature and using the biodiesel fuels with long carbon chain length and a high proportion of unsaturated fatty acids. The latter is further supported by the fact that many studies reporting higher NO_x emissions [11, 14, 20, 27-31, 33, 34] used biodiesel fuels produced from soybean, sunflower, and rapeseed, in which unsaturated fatty acids (e.g. C18:1 and C18:2) are major components [9, 35]. However, some studies [12, 15, 21, 36] reported a directly opposite trend of the decreased engine-out NO_x**

emissions. Interestingly, these studies used coconut oils as a biodiesel feedstock and it is believed that short carbon chain length and a high proportion of saturated fatty acids (e.g. C12:0 and C14:0) in coconut **biodiesel fuels** caused lower flame temperature than that of petroleum diesel [23, 30, 33]. Our previous work [37] showed a consistent result such that a biodiesel blend produced from a lipase-catalysed trans-esterification process in ethanol moiety produced lower NO_x emissions than petroleum diesel due to the short carbon chain length and low flame temperature.

In addition to reduced NO_x emissions, our previous study on coconut oil-based biodiesel also demonstrated the improved brake power over petroleum diesel [37]. This was likely due to the improved lubricity of biodiesel and thus decreased friction loss in the common-rail fuel pump. The lubricity benefit of **biodiesel fuels**, regardless of feedstock, is well known to be caused by inherent lubrication properties from fatty acid esters [38, 39]. **For example, it is reported that the use of biodiesel can lead to the decreased wear and friction within the high-pressure fuel pump in which fuel provides lubrication [40-42]. Some studies also showed the reduced friction on the cylinder liner [41-46]. However, the viscosity of biodiesel fuels is also higher leading to the increased friction loss, which generally outperforms the friction loss reduction caused by the improved lubricity. It is therefore interesting that the viscosity of coconut oil biodiesel fuels is much lower than that of other widely produced biodiesel fuels (e.g. soybean, rapeseed, or sunflower [47]) and thus the positive effect of improved lubricity on reduced friction loss and increased brake power becomes measurable [37, 45].**

Our previous study [37], however, was focused on the impact of fuel production process on engine performance while a limited biodiesel blending ratio of 10% was used. **The present work studies potential benefits of using higher blending ratio biodiesel fuels produced from the same coconut oils and production method.** We conducted engine performance and emissions testing of four different biodiesel blends including petroleum diesel (B0), B10, B25, and B40. Higher blending ratios than 40% were also considered but were not tested due to decreased brake power. This will be reported in detail later in the results and discussion section. The biodiesel blends were tested in a single-cylinder, light-duty diesel engine equipped with a common-rail fuel injection system. The in-cylinder pressure and brake torque were measured while engine-out emissions of smoke (opacity), NO_x, uHC, and CO were also recorded. In addition to biodiesel blending ratio variations, the fuel injection timing was also altered, considering significant influence of combustion phasing on brake power and engine-out emissions [48-53].

2. Experiments

2.1 Fuels

In-house produced coconut oil biodiesel was used in the present study. As reported previously [54, 55], transesterification of the triglycerides in coconut oil into esters was achieved through an enzyme catalytic conversion process using a commercial lipase (Novozyme 435). The main components of coconut oil are triglycerides, and up to 99.8% of these triglycerides have high carbon numbers ranging from 28 to 52, which is not suitable for the direct use in diesel engines. Coconut oil also contains a small 4.5% of low molecular weight free fatty acids. The transesterification was performed in a shaking incubator at fixed reaction temperature of 50°C for 50 hour incubation. For this reaction, a mixture of coconut oil and ethanol at a 5:1 ratio by volume was used. The mixture was agitated at 350 rpm for at least 30 min to ensure adequate homogenisation before adding

1 g lipase per 100 ml coconut oil to initiate the reaction. To overcome a lower production rate of the lipase catalyst than that of traditional alkali catalysts, an ultrasonic cleaner (DeltaNewInstrument D80H) was used with an operating frequency of 43 kHz and output power of 80 W. After the 50 hour incubation, the mixture was then left overnight at room temperature to achieve effective separation of the ester-containing phase (i.e. fatty acid ethyl ester - FAEE) from the aqueous glycerol phase. The high purity of this enzyme reaction had a great advantage over an alkali-based reaction as no additional treatment such as waste water treatment or adsorptive post-treatment was required [56].

Table 1 summaries the key properties of coconut oil based neat FAEE biodiesel produced from this process. It is notable that the density of neat biodiesel is only slightly higher than that of petroleum diesel but the kinematic viscosity of 3.8 mm²/s is twice higher than petroleum diesel. The cetane number was not directly measured for the biodiesel; however, literature suggests higher cetane number of 57~59 [57, 58] compared to petroleum diesel's 51. As expected [8-10, 13, 30], the calorific value of neat biodiesel (39 MJ/kg) is 6% lower than that of petroleum diesel (41.66 MJ/kg). The oxygen concentration in neat biodiesel is estimated at 14.04% by mass, which means 12.55% and 1.49% less carbon and hydrogen, respectively, compared to petroleum diesel. The neat biodiesel was mixed with petroleum diesel to produce three different biodiesel blends including B10, B25, and B40 for engine performance and emissions testing. The fuel properties estimated by volume averaging are also given in Table 1.

2.2 Single-cylinder diesel engine and operating conditions

Fig. 1 shows a single-cylinder direct-injection diesel engine and measurement tools used in the present study. The single-cylinder diesel engine shares the production engine head and common-rail fuel injection system of a four-cylinder small-bore engine, widely available in the Australian market. The engine specifications are summarised in Table 2. The displacement volume of engine is 497.8 cm³ with 83 mm bore and 92 mm stroke. The piston crown has a cylindrical bowl shape (55 mm in diameter), which leads to a geometric compression ratio of 17.7. The swirl ratio is 1.4 according to manufacturer's specification. Two 60-litre surge tanks were used to minimise pressure fluctuations associated with the single-cylinder engine operation both upstream of the intake ports and downstream of the exhaust ports.

A common-rail fuel injection system (Bosch CP3) was used for all the tested biodiesel blends of the present study. A solenoid-type injector has 7 nozzle holes with a nominal hole diameter of 134 µm and included angle of 150°. The nozzle was a mini-sac type and has a convergent, hydro-ground orifice with a discharge coefficient of 0.86 (KS1.5/86 specification by Bosch). The tested hydraulic flow rate (HFR) was 400 cm³ for 30 s at 10 MPa fuel pressure. The injection pressure, duration and timing were controlled independently using a universal control unit (Zenobalti ZB-9013P).

The engine was operated at fixed speed of 2000 revolutions per minute (rpm) at which the referred production engine generates the maximum torque. The constant-speed operation was achieved using an eddy current dynamometer (Froude Hoffmann, AG-30HS). The coolant temperature was maintained at 90°C by a water temperature controller/circulator (ThermalCare Aquatherm RA Series) to simulate fully warmed-up engine conditions. The intake air temperature was monitored using a K-type thermocouple upstream of intake air ports throughout the experiments, which indicated 27°C with minimal variations (< ±1°C). The fuel injection pressure was held constant at 130 MPa while the injection timing was varied from 13° crank angle before top

dead centre (CA bTDC) to 3°CA bTDC. The injection timing sweep was repeated for all the biodiesel blends tested in the present study.

Fig. 2 shows a Bosch-tube-type injection rate meter (top) used to measure an injection profile (bottom) of all the tested biodiesel blends of the present study. The injection rate meter measures the pressure deviation occurring within the fuel-filled tube, which is converted to the injection rate profile. The injection was repeated for 20 times and the averaged injection rate profiles are plotted in Fig. 2 (bottom). A measurable delay between the start of injection current and the actual start of injection is noticed. This delay, a time required to activate the solenoid valve and the needle lift, was consistently measured at 0.36 ms regardless of the biodiesel blending ratio. The injection rate shows an overall decreasing trend in the peak value with an increasing biodiesel blending ratio. The lower fuel discharge of higher biodiesel blends was explained by increased fuel viscosity, which should be taken into account in determining the fuel injection duration together with the decreased calorific value and increased density. As shown in Table 3, a baseline operating condition of the present study required 25.98 mg of petroleum diesel injection for 1.08 kJ of total fuel energy, which corresponds to 0.65 ms of fuel injection duration (electronic). To maintain the same fuel energy input, the injected fuel mass was increased by 0.63 mg for B40, which required the increased injection duration of 0.66 ms. **The increased fuel mass led to the decreased air/fuel ratio, which would increase the equivalence ratio of petroleum diesel. However, for the oxygenated fuels used in the present study, the oxygen in fuel needs to be considered for the air/fuel ratio (or equivalence ratio) calculation. Therefore,** the traditional definition of ϕ is not necessarily an accurate measure of mixture stoichiometry when oxygenated fuels are used because oxygen must be classified as an oxidiser. Therefore, the oxygen equivalence ratio ϕ_Ω is calculated, which is defined as [59]:

$$\phi_\Omega = \frac{\phi}{1 - \Omega_f(1 - \phi)} \quad (1)$$

In this equation, the traditional definition of ϕ is used together with the oxygen ratio of the fuel:

$$\Omega_f = \frac{n_o}{2n_C + 1/2n_H} \quad (2)$$

Where the Ω_f represents a ratio of the number of O-atoms per mole of fuel to the number of O-atoms required to convert all C- and H-atoms in a mole of fuel to CO₂ and H₂O. Since ϕ_Ω is the equivalence ratio based on the oxygen ratio, it can be used to accurately quantify mixture stoichiometry even for biodiesel blends containing O atoms. Table 3 shows that both ϕ and ϕ_Ω decrease with an increasing biodiesel blending ratio but the differences are less for ϕ_Ω as oxygen in fuel is counted as an oxidiser. Between petroleum diesel and B40, the ϕ_Ω reduction is estimated at 5.4% and **therefore the impact of leaner mixture condition on combustion and emissions must be considered.**

2.3 Measurements

A piezo-electric pressure transducer (Kistler 6056A) was used to measure the in-cylinder pressure traces. The pressure data were recorded for 100 cycles for each engine operating condition and then used to obtain an average pressure trace, indicated mean effect pressure (IMEP), and the coefficient of variation of IMEP (CoV of IMEP). The apparent heat release rate (aHRR) was also calculated from the in-cylinder pressure trace. From the aHRR trace, the ignition delay was estimated by reading the time between the start of injection (SOI) and the start of combustion (SOC). In this study, the measured injection rate (Fig. 2) was used for SOI and the crank angle location for 10% heat release (i.e. CA10) was defined as SOC. The aHRR trace was also used to

analyse burn durations for both the initial burn (CA10 to CA50) and late-cycle burn (CA50 to CA90). For each engine operating condition, engine-out emissions were measured. We used an opacimeter (Horiba MEXA-600S, accuracy: $\pm 0.15 \text{ m}^{-1}$ light absorption coefficient) to measure the smoke emissions, a chemiluminescence-type gas analyser (Ecotech 9841AS, 1% accuracy) for NO_x emissions, and a non-dispersive infrared (NDIR) analyser (Horiba MEXA-584L, 1.7% accuracy) for unburnt hydrocarbon (uHC) and carbon monoxide (CO) emissions.

3. Results and discussion

3.1 In-cylinder pressure

Fig. 3 (top) shows the averaged in-cylinder pressure traces for various biodiesel blends and fuel injection timings. Well expected trends for diesel engine combustion are reproduced. For example, advanced fuel injection leads to earlier start of combustion and higher pressure rise due to the main combustion occurring closer to TDC where the ambient gas pressure and temperature is high. This results in higher peak in-cylinder pressure for advanced fuel injection timings. For the tested injection timing range, the fuel injection at 13°CA bTDC shows the start of pressure rise near TDC and hence displays the highest peak pressure. In comparison, the start of pressure rise occurs later in the expansion stroke for retarded injection timings of 8 and 3°CA bTDC. The test of earlier injection timings than 13°CA bTDC was limited due to the increased negative work (i.e. pressure rise occurring against upward moving piston) and audible noise caused by high in-cylinder pressure and pressure ringing [60]. Also, the fuel injection timings later than 3°CA bTDC was not achievable due to misfiring associated with a very retarded combustion phasing.

Compared to measurable differences in the in-cylinder pressure traces caused by the fuel injection timing variations, the effect of biodiesel blending ratio on the in-cylinder pressure appears to be less significant [27, 49]. This suggests the control of combustion phasing using fuel injection timing variations will be effective for biodiesel blends as it is for petroleum diesel. However, a careful inspection of pressure traces around the start of pressure rise suggests that B40 (dot lines) tends to show earlier start of pressure rise regardless of the injection timing. This is not ignorable either in Fig. 3 (bottom) displaying the in-cylinder pressure traces with two standard deviations (2σ for 95% confidence) in consideration of cyclic variations. The injection timing of 13°CA bTDC is selected for this example plot. In other words, the earlier start of pressure rise for higher biodiesel blends is observed not only in the averaged in-cylinder pressure but also in individual pressure traces, suggesting the observed difference is real. This simply confirms that the tested biodiesel from coconut oils has higher cetane number than that of petroleum diesel (see Table 1).

3.2 Apparent heat release rate

To further discuss global in-cylinder phenomena for various biodiesel blending ratios, the averaged in-cylinder pressure was used to obtain an aHRR trace as shown in Fig. 4 (top). The ignition delay was also estimated by reading a time between the start of injection and CA10 (i.e. the crank angle location for 10% heat release), which is plotted in Fig. 4 (bottom) for all tested conditions of the present study. Similar to Fig. 3, the SOC is further away from TDC with retarded injection timing which causes longer ignition delay due to the reduced ambient pressure and temperature. Indeed, a well expected trend is observed such that the peak aHRR is higher for later injection timings due to longer ignition delay and thus increased time for pre-combustion mixing

[61]. For instance, the 3°CA bTDC injection with over 2°CA longer ignition delay time than the other two injection timings shows more than 60 J/°CA higher peak aHRR at any fixed biodiesel blending ratio. Since the main heat release is largely driven by the combustion of pre-mixed charge, the increased premixing would lead to increased peak aHRR and likely higher flame temperature.

For various biodiesel blending ratios, higher biodiesel blends tend to show the earlier rise of aHRR, which is consistent with decreasing ignition delay time. Specifically, as the biodiesel blending ratio increases from 10% to 40%, the ignition delay decreases gradually within a 1°CA window. While variations in the ignition delay time is not so sensitive to the biodiesel blending ratio as they are to the injection timing, the decreased ignition delay time for higher biodiesel blends is measurable and as mentioned previously, this trend is expected considering higher cetane number of coconut oil biodiesel than that of petroleum diesel. However, a direct opposite trend is also observed. Between petroleum diesel and B10, the ignition delay time of B10 is up to half crank angle longer than that of diesel. A cause for the observed trend is not entirely clear but some studies reported that poor atomisation of biodiesel associated with higher viscosity could increase the ignition delay time [9, 62]. How the increased cetane number and worse atomisation impact the ignition delay time differently depending on the biodiesel blending ratio requires a further investigation.

3.3 Indicated mean effective pressure and burn duration

The in-cylinder pressure traces in Fig. 3 together with known cylinder volume at a given crank angle degree were used to estimate the indicated mean effective pressure (IMEP). The results are plotted in Fig 5. Two noticeable trends are observed in the figure such that the IMEP of advanced injection timing shows higher value than that of retarded injection timing at any fixed biodiesel blending ratio and the IMEP decreases with an increasing biodiesel blending ratio for a given injection timing. The former was due to the combustion phasing that was positioned just after TDC for the 13°CA bTDC injection, which caused the highest in-cylinder pressure and thus highest IMEP. By contrast, later injection timings with more retarded combustion phasing led to lower in-cylinder pressure during the combustion event and thus lower IMEP. The latter trend associated with the variations in biodiesel blending ratio, however, cannot be explained by in-cylinder pressure traces because their differences were very minor.

A possible explanation for the observed trend could be leaner mixture for high biodiesel blends. That is, the oxygen equivalence ratio of B40 was 5.4% lower than that of petroleum diesel and therefore reduced flame temperature could lead to lower IMEP. To further discuss this trend, burn durations were calculated using the aHRR data presented in Fig. 4 (top) and the results are shown in Fig. 6. The burn duration was estimated by reading the crank angle locations of 10%, 50% and 90% of the total heat release for the early “CA10-CA50” phase and late “CA50-CA90” phase of diesel combustion. It appears that the burn duration of the early combustion does not vary much with the biodiesel blending ratio for fixed injection timing. It is also interesting to note that both the decreasing ignition delay from B10 to B40 and the longer ignition delay of B10 than that of petroleum diesel observed in Fig. 4 (bottom) do not make a significant impact on the burn duration of the following combustion phase. By contrast, the burn duration for the late combustion phase (i.e. CA50-CA90) shows measurable variations such that it increases with an increasing biodiesel blending ratio at any fixed injection timing. The extended burn duration during the CA50-CA90 burn could be explained by the above-mentioned leaner mixture and thus decreased temperature of diffusion flames [63]. Moreover, the lower calorific value of coconut oil biodiesel would also decelerate the reaction [30]. This slower reaction then

explains the decreased IMEP for higher biodiesel blends since the slower burning deteriorates the combustion by releasing heat energy in relatively longer time and thereby weakening the subsequent reactions [32].

Regarding the fuel injection timing, the burn duration appears to be longer for more advanced injection timing. However, this timing showed higher IMEP in Fig. 5 and thus conflicts with the earlier explanation for lower IMEP of higher biodiesel blends due to the increased burn duration. This was because IMEP is not only impacted by the burn duration but also by the combustion phasing [32]. In other words, the combustion phasing of more advanced injection timing was placed closer to TDC where the ambient gas pressure and temperature is high and thereby increasing the resulting in-cylinder pressure during diesel combustion. This often plays a more dominant role than the burn duration [37], leading to higher IMEP. Since the combustion phasing does not vary much for various biodiesel blends at any fixed injection timing (Fig. 3), the IMEP variations in this case were predominantly governed by the burn duration.

3.4 Brake/friction mean effective pressure and brake specific fuel consumption

To evaluate how the variations in IMEP with the biodiesel blending ratio and fuel injection timing impact the shaft power, the brake mean effective pressure (BMEP) was calculated by reading the brake torque of EC dynamometer. The results are plotted in Fig. 7 for all tested conditions of the present study. The figure shows that the advanced fuel injection achieving higher IMEP (Fig. 5) results in higher BMEP, which is well expected as the increased work produced inside the engine cylinder leads to the increased shaft power. However, the BMEP trend with an increasing biodiesel blending ratio does not show a simple reflection of the decreasing IMEP trend. For the advanced 13°C**A** bTDC injection, the BMEP of B10 is higher than that of petroleum diesel while B25 shows the same BMEP of 425 kPa. Even at 40% biodiesel blending ratio, the BMEP is only 2 kPa (or 0.2%) lower than that of petroleum diesel, which is an order of magnitude lower than 20 kPa (or 2%) decrease in IMEP (see Fig. 5). A similar trend is also observed for the 8°C**A** bTDC injection such that 25 kPa (or 2.7%) reduction in IMEP at 40% biodiesel blending ratio is not fully reflected in 5 kPa (or 1.2%) decrease in BMEP.

The friction mean effective pressure (FMEP) is a difference between IMEP and BMEP, meaning that the less significant decrease of BMEP than that of IMEP at high biodiesel blending ratios can be interpreted as reduced friction losses. Fig. 7 shows FMEP values calculated for all tested conditions of the present study, which confirms that indeed the friction losses decrease with an increasing biodiesel blending ratio. This reduction in FMEP could be explained by higher lubricity of coconut oil biodiesel. That is, the enhanced fuel lubricity due to the use of biodiesel [64] could reduce the friction in the common-rail pump [40-42] and inside the cylinder [37, 38, 41-46]. Moreover, slightly higher viscosity of biodiesel than that of petroleum diesel could reduce the blow-by through the ring clearance, resulting in less fuel-in-oil issue and thus enhanced lubrication in the crank case [23]. Therefore, results in Fig. 7 suggest that the reduced FMEP helped increase BMEP for B10 when the IMEP reduction was not significant. The BMEP eventually decreased below the petroleum diesel level due to a further reduction in IMEP, which however was also compensated by the decreased FMEP [45]. An extreme case that the improved lubricity of biodiesel blends made a significant impact on BMEP is found for the 3°C**A** bTDC injection – even the BMEP of B40 is higher than that of petroleum diesel. It could be thought that the lower in-cylinder pressure and the combustion occurring late in the expansion stroke (Fig. 3) further reduced the friction losses compared to the higher in-cylinder pressure and near TDC combustion of 13 and 8°C**A** bTDC injection cases. Indeed, the FMEP for the 3°C**A** bTDC injection shows lower value than the other two injection timings, particularly at the higher biodiesel blending ratio range.

Considering the use of a single-cylinder diesel engine in the present study, care should be taken to interpret the results of BMEP and FMEP. It is known that single-cylinder engines have higher FMEP than multi-cylinder engines due to less inertia [31]. Also, we used a heavy flywheel to compensate the loss of inertia due to the lack of other pistons. Therefore, the impact might have been exaggerated as the absolute value of FMEP was very high and thus had a larger room for the FMEP improvement.

It should be reminded that the brake power measurements were conducted at fixed fuel energy conditions, i.e. the total fuel energy was held constant at 1.08 kJ per cycle (Table 3). Therefore, one might think that a very minor decrease in BMEP ($<1.2\%$) is acceptable in terms of the energy cost. This will raise a question on why the maximum biodiesel blending ratio of the present study is limited at B40. However, the lower calorific value of coconut oil biodiesel required increased injection mass for biodiesel blends. Therefore, the brake specific fuel consumption (BSFC) and fuel conversion efficiency should be considered together with BMEP. The results are shown in Fig. 8. It is seen that petroleum diesel achieves the lowest BSFC of 442 g/kWh and highest fuel conversion efficiency of 19.5% when the fuel injection timing is advanced at 13°C bTDC. A similar or lower BSFC is measured for B10 with 8 or 13°C bTDC injection timings; however, higher biodiesel blends show higher BSFC than 442 g/kWh regardless of the fuel injection timing. For the fixed injection timing of 13°C bTDC, the BSFC of B40 is 16 g/kWh (or 3.6%) higher than the minimum BSFC of petroleum diesel. Therefore, the increasing BSFC trend with increasing biodiesel blending ratio limited the testing of higher biodiesel blends than B40 in the present study. Similarly, the decreasing fuel conversion efficiency limited the further increase in the biodiesel blending ratio. Despite the fact that the lower price of biodiesel blends will reflect the lower fuel economy and a primary motivation for the use of biodiesel is the fuel production from renewable feed stock, it should be considered that the conventional fuel economy is measured by fuel quantity, not by the fuel energy input or environmental impact. Therefore, the results in Fig. 8 suggest that higher coconut oil biodiesel blends than B10 cannot provide advantages over petroleum diesel.

3.5 Carbon monoxide and unburnt hydrocarbon emissions

From the previous section, the reduced BMEP and increased BSFC for high biodiesel blends suggest the disadvantages of using biodiesel in diesel engines. However, considering oxygenated nature of biodiesel (14% by mass in the tested coconut oil biodiesel), there could be benefits associated with enhanced oxidation reactions [11, 12, 20, 21]. Indeed, Fig. 9 (top) shows decreasing engine-out emissions of CO with an increasing biodiesel blending ratio at any fixed fuel injection timing. The reduction rate of CO emissions, however, is different for different fuel injection timings. For example, the 13°C bTDC injection shows high CO emissions of 35 g/kWh for petroleum diesel, which is decreased by 15 g/kWh (or 43%) for B40. In comparison, the 8°C bTDC injection exhibits lower CO emissions of 22.5 g/kWh for petroleum diesel, which is reduced by 4.5 g/kWh (or 20%) for B40. The high CO emissions for petroleum diesel at advanced fuel injection timing were expected because longer burn duration (i.e. slower reaction) was measured (see Fig. 6). The slower reaction will allow for increased chance of reaction quenching, which hinders the conversion of CO to CO₂ [65, 66]. When high biodiesel blends were used, this oxidation reaction was enhanced, leading to significantly reduced CO emissions. It appears that this effect of enhanced oxidation reaction on CO reduction has a certain limit as evidenced by nearly identical CO emission level for all tested injection timings at B40. Therefore, the reduction in CO emissions due to biodiesel blends was relatively larger for the advanced injection timing with much higher CO emissions for petroleum diesel. An interesting trend observed in Fig. 9 is that as the injection timing is further

retarded to 3°CA bTDC, the CO emissions increase again. For example, the CO emissions for petroleum diesel is measured at 28 g/kWh, which is higher than that of the 8°CA bTDC injection but lower than the 13°CA bTDC injection. This was likely due to extended ignition delay (Fig. 4) during which over-leaning of fuel-air mixture can lead to increased CO emissions [67-69]. Therefore, both the ignition delay time for pre-combustion mixing and the reaction rate during the main combustion event should be considered in determining CO emissions. Despite the varying significance, the trend of decreasing CO emissions with an increasing biodiesel blending ratio is always observed in Fig. 9 regardless of the fuel injection timing.

Fig. 9 (middle) also shows engine-out emissions of unburnt HC for various biodiesel blending ratios and fuel injection timings. A noticeable trend is that uHC emissions increase with an increasing biodiesel blending ratio, which is directly opposite to the trend observed for CO emissions. A likely cause for the observed trend is high viscosity and low volatility of biodiesel blends. As mentioned previously, the high viscosity could worsen atomisation [9], which together with low fuel volatility, could cause over-penetration of liquid-phase fuel and wall wetting issues [13, 70-72]. This means increased unburnt or partially burned hydrocarbons that cannot even be converted to CO. Therefore, as shown in Fig. 9, it is possible that uHC emissions increase while CO emissions decrease. The increased uHC emissions then further explain the non-linear trend observed for CO emissions in terms of the fuel injection timing (i.e. CO emissions at 3°CA bTDC injection being lower than that of the 13°CA bTDC injection but higher than that of 8°CA bTDC). Specifically, the significantly retarded combustion phasing of the 3°CA bTDC injection with the main combustion occurring late in the expansion stroke [37] led to a very high level of unburnt or partially burnt hydrocarbons [71], which could not even contribute to CO emissions. As such, both CO and uHC emissions should be considered together when overall combustion efficiency is evaluated. From the measured CO and uHC emission results, our previous studies [68] estimated combustion efficiency by considering these incomplete combustion products as a contributor to the combustion efficiency reduction. The same calculation was repeated for the CO and uHC emissions in Fig. 9 and the results are shown at the bottom. It is very clear that, despite the variations in CO and uHC emissions, the overall combustion efficiency is estimated at over 98% regardless of injection timing or biodiesel blending ratio. Therefore, the significance of CO and uHC emission trends shown in Fig. 9 might not be high unless these emissions are strictly regulated. The high combustion efficiency was likely due to our selection of the engine operating conditions involving overall lean mixture, no exhaust gas recirculation, fixed engine speed, fixed coolant temperature of 90°C, and high common-rail pressure of 130 MPa.

3.6 Smoke and oxides of nitrogen emissions

The same or slightly decreased BMEP of biodiesel blends (Fig. 7) with high combustion efficiency (Fig. 9) could offer a great advantage over petroleum diesel if reduced smoke emissions of biodiesel are considered. This is because engine developers often compromise the brake power to reduce the smoke emissions below the regulation limit. If low-sooting biodiesel blends are used, engines could be setup either for higher brake power while matching the same smoke level of petroleum diesel or for lower smoke emissions while matching the brake power. The latter is the approach of this study. Fig. 10 (top) shows a dramatic reduction of opacity (smoke emissions) with an increasing biodiesel blending ratio. One might think that the decreasing oxygen equivalence ratio (i.e. leaner mixture) with increasing biodiesel blending ratio (Table 3) would lead to reduced smoke emissions. However, the four-fold smoke reduction seen in Fig. 10 (top) cannot be explained by the mixture stoichiometry alone. **The reduced smoke emissions were due to a well-known effect of oxygenated fuel on**

suppressed soot formation [17] and enhanced soot oxidation with high reactivity [8, 10, 18, 19, 73, 74]. Also, high-sooting aromatic contents are much less in biodiesel fuels than petroleum diesel, which further suppressed the soot formation [75, 76]. Even compared to other biodiesel fuels produced from soybean, sunflower, and rapeseed, the coconut oil-based biodiesel fuels have higher oxygen content [33] and thus the reduced soot formation and increased soot oxidation could be more significant. For instance, the soot oxidation rate is higher when biodiesel fuels are comprised of shorter carbon-chain components with increased oxygen contents [33]. Also, a lower degree of unsaturated fatty acids in coconut oils (only 9%) can contribute to the reduced smoke emissions because the unsaturated fatty acids with double bonds have higher sooting propensity than the saturated fatty acids [77, 78]. In the present study, over 70% reduction of smoke emissions is achieved for B40 over petroleum diesel at any fixed fuel injection timing.

Regarding injection timing variations, more advanced injection with less premixing time and thus lower aHRR [49] (Fig. 4) shows higher smoke emissions at a fixed biodiesel blending ratio. However, the trend is reversed for NO_x emissions. Fig. 10 (bottom) shows higher NO_x emissions for 3°CA bTDC injection than earlier injection timings. This is also explained by the characteristics of heat release rate such that the higher peak aHRR and thus higher in-cylinder temperature [48] caused increased NO formation via thermal Zel'dovich mechanism [79]. For the biodiesel blending ratio variations, Fig. 10 (bottom) shows decreased NO_x emissions with an increasing biodiesel blending ratio. For example, the NO_x emissions were reduced by 10% for B40 at fixed fuel injection timing. This reduction is rather surprising if the well-known smoke-NO_x trade-off is considered. As mentioned previously in the introduction section, there are mixed conclusion about biodiesel NO_x emissions in the literature. For example, some studies report increased NO_x emissions of biodiesel compared to petroleum diesel [9, 11, 14, 20, 25-29] while other studies suggest decreased NO_x emissions [12, 15, 21, 36]. The detailed study conducted in a heavy-duty optical diesel engine [31] suggested that the charge-gas mixture conditions at the flame base play a key role in determining the engine-out NO_x emissions such that the biodiesel mixture being closer to stoichiometric can lead to increased local and average in-cylinder temperatures and thereby increasing thermal NO formation. As shown in Table 3, the oxygen equivalence ratio at the tested conditions of the present study was reduced at higher biodiesel blends. This might contribute to the reduced in-cylinder temperature and NO formation, which however is directly opposite to Mueller et al. [31]. Moreover, given that the small-bore diesel engine used in the present study does not show a (quasi-)steady flame base [60, 80], there can be no reference point to evaluate the correspondence between the mixture stoichiometry at the flame base and NO_x emissions. Another parameter of consideration is fuel molecular structure of coconut oil-based biodiesel. For example, coconut oil is comprised of hydrocarbons with shorter carbon chain length (i.e. C6~C14) and a lower degree of fatty acids unsaturation [30, 33], both of which would be expected to decrease the flame temperature and thus lower NO formation, compared to other biodiesel fuels (e.g. soybean, sunflower, rapeseed) [9, 35]. Since the simultaneous reduction of smoke and NO_x emissions can be achieved with high biodiesel blends, coconut oil-based biodiesel offers a significant benefit over other biodiesel fuels with higher NO_x emissions than petroleum diesel.

Finally, Fig. 11 shows smoke and NO_x emissions replotted over BSFC using results from Figs. 8 and 10. The figure clearly shows that the simultaneous reduction of smoke and NO_x emissions at high biodiesel blending ratios can be achieved only at the expense of BSFC. For instance, if engine developers concern primarily about smoke and NO_x emissions, B40 at 8°CA bTDC can be selected (illustrated by green rectangle).

However, its BSFC is 20 g/kWh (or 4.5%) higher than that of petroleum diesel. The optimised operating condition of the present study, therefore, is B10 at 13°CA bTDC as it shows lower smoke and NO_x emissions as well as lower BSFC than those of petroleum diesel. For the tested conditions of this study, the results suggest that high biodiesel blends are not preferred but B10 for both fuel economy and emission benefits.

4. Conclusions

Various blends of petroleum diesel and lipase-catalysed coconut oil-based ethyl ester biodiesel were tested for global in-cylinder phenomena and engine-out emissions in a single-cylinder, automotive-size diesel engine. The measured in-cylinder pressure was used to calculate key combustion parameters such as indicated mean effective pressure, apparent heat release rate, and burn duration. The brake torque measured in an eddy current dynamometer was used to estimate brake mean effective pressure and subsequently friction mean effective pressure. The brake specific fuel consumption was also calculated using the measured injection mass of all tested biodiesel blends. The major findings of this study focused on fuels effect on engine combustion are summarised as follows:

- For higher biodiesel blends, the lower calorific value of coconut oil-based biodiesel and overall leaner mixture cause the decreased diffusion flame temperature and the extended burn duration during the mixing-controlled phase of combustion. For the tested conditions of this study, the slower reaction leads to decreased indicated mean effective pressure.
- Despite the decreased indicated mean effective pressure of high biodiesel blends, the improved lubricity of coconut oil biodiesel, together with its relatively lower viscosity than other **biodiesel fuels**, reduces friction losses and thereby maintaining similar brake mean effective pressure of petroleum diesel. However, for fixed total fuel energy conditions of this study, a significant increase in the brake specific fuel consumption is problematic for high biodiesel blends.
- Being oxygenated fuel, the coconut oil-based biodiesel has a high oxidation rate of CO and thereby decreasing engine-out CO emissions. However, the deteriorated atomisation due to high viscosity and the lower fuel volatility are thought to cause the over-penetration of liquid-phase fuel and wall wetting issues, leading to increased unburnt HC emissions. While these interesting trends are observed, the overall combustion efficiency is estimated at over 98% regardless of a biodiesel blending ratio in this study. Therefore, the significance of CO and uHC emission trends is not high.
- Due to a well-known effect of oxygenated fuel on suppressed soot formation and enhanced soot oxidation as well as reduced aromatic contents, the engine-out smoke emissions are significantly lower for high biodiesel blends. For the tested conditions of this study, over 70% smoke reduction is achieved for B40 compared to petroleum diesel.
- A great advantage of the tested biodiesel blends is found in decreased NO_x emissions, which can be explained by decreased flame temperature associated with short carbon chain length of coconut oil-based biodiesel. For B40, over 10% decrease in NO_x emissions is measured compared to petroleum diesel.
- When the brake specific fuel consumption and smoke/NO_x emissions are considered altogether, however, the optimised conditions of the present study is found for B10, suggesting that the full use of

reduced emission benefits of high biodiesel blends is not realistic but it has to be compromised for the acceptable brake specific fuel consumption.

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Table 1 Fuel properties

Fuel		Diesel	B10	B25	B40	B100	Test Methods		
Density @ 15°C (kg/m ³)		848	849	849.75	850.80	860	ASTM D1298		
Kinematic viscosity @40°C (mm ² /s)		1.90	2.94	3.14	3.29	3.80	ASTM D445		
Flash point (°C)		>61.5	66.2	73.3	80.3	108.5	ASTM D93		
Cetane number		51	-	-	-	-	ASTM D613		
Calorific value (MJ/kg)		41.66	41.39	41.00	40.60	39	ASTM D240		
CHO wt. %	C	86.23	84.58	82.30	80.25	73.68	N/A		
	H	13.77	13.58	13.31	13.06	12.28	N/A		
	O	0	1.85	4.39	6.69	14.04	N/A		
Fatty acid composition of coconut oil									
	C6:0	C8:0	C10:0	C12:0	C14:0	C16:0	C18:0	C18:1	C18:2
Content wt. %	1	8	6	47	17	9	3	7	2

Table 2 Engine specification and operating conditions

Engine Specifications	
Displacement (single cylinder)	497.8 cm ³
Bore	83 mm
Stroke	92 mm
Piston	Cylindrical bowl (55 mm in diameter)
Compression ratio	17.7
Swirl ratio	1.4
Number of valves	2 intake and 2 exhaust
Injection system	7-hole Bosch common-rail Nominal hole diameter: 134 μ m K-factor: 1.5 Discharge coefficient: 0.86 (diesel fuel) HFR: 400 cm ³ for 30 s at 10 MPa fuel pressure Included angle: 150°
Operating Conditions	
Engine speed	2000 rpm
Coolant temperature	90°C
Intake air temperature	27°C
Injection pressure	130 MPa
Injection timing	3, 8 and 13°CA bTDC

Table 3 Injected mass, fuel energy, and stoichiometry for various biodiesel blending ratios

	Injection duration (elec.) [ms]	Injected mass [mg]	CV [MJ/kg]	Fuel energy [kJ]	ϕ	Ω_f	ϕ_Ω
Diesel	0.65	25.98	41.66	1.08	0.665	0	0.665
B10	0.65	26.12	41.39	1.08	0.653	0.006	0.654
B25	0.66	26.38	41	1.08	0.638	0.013	0.642
B40	0.66	26.61	40.6	1.08	0.624	0.021	0.629

Fig. 1. Diesel engine setup and measurement techniques

Fig. 2. Bosch-tube-type injection rate meter setup (top) and measured injection rates for petroleum diesel, B10, B25 and B40 fuels tested in the present study (bottom).

Fig. 3. Effect of biodiesel blending ratio and fuel injection timing on in-cylinder pressure (top). Shown at the bottom is the in-cylinder pressure traces of two standard deviations at a selected injection timing of 13°CA bTDC

Fig. 4. Effect of biodiesel blending ratio and fuel injection timing on the apparent heat release rate (aHRR) and ignition delay time

Fig. 5. Effect of biodiesel blending ratio and fuel injection timing on the indicated mean effective pressure (IMEP)

Fig. 6. Effect of biodiesel blending ratio and fuel injection timing on the burn durations

Fig. 7. Effect of biodiesel blending ratio and fuel injection timing on the brake mean effective pressure (BMEP) and friction MEP (FMEP)

Fig. 8. Effect of biodiesel blending ratio and fuel injection timing on the brake specific fuel consumption and fuel conversion efficiency

Fig. 9. Effect of biodiesel blending ratio and fuel injection timing on the emissions of carbon monoxide (CO) and unburnt hydrocarbon (HC). Shown at the bottom is the combustion efficiency estimated using the CO and HC emissions.

Fig. 10. Effect of biodiesel blending ratio and fuel injection timing on smoke (exhaust opacity) and NO_x emissions

Fig. 11. Smoke (exhaust opacity) and NO_x emissions over BSFC for various biodiesel blending ratios and fuel injection timings