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Experimental investigation of ethanol/diesel dual-fuel combustion in a heavy-duty diesel engine

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ABSTRACT

Ethanol is a promising alternative fuel applicable in the internal combustion engine by virtue of its sustainability and soot-reducing potential. In this study, ethanol is injected into the intake port while diesel is directly injected into the cylinder of a heavy-duty diesel engine enabling dual-fuel operation. The main goals of the study are to probe the ethanol substitution ratio and load range and assess the resulting engine performance and emissions. Tests were performed with the original calibration at several loads using the European Stationary Cycle. The results show that ethanol mass ratios of up to 80% may be reached at low to medium loads without misfire. Addition of ethanol can reduce soot emissions, with no consistent effects on NOx emissions. As the ethanol mass ratio increases, dual-fuel operation suffers from incomplete combustion progressively. Increased HC and CO emissions are, however, believed to be manageable by a diesel oxidation catalyst at high loads. Both combustion and thermal efficiency decrease at low load when ethanol is introduced. However, thermal efficiency at medium load increases from 49.1% to 50%. For medium to high loads, thermal efficiency first increases to 50.7% and 49.7%, respectively, then decreases due to sub-optimal combustion phasing at high ethanol mass ratios. It is noteworthy that the pressure rise rate, ringing intensity, and peak pressure may appear to limit the ethanol ratio to below 40% for medium to high loads. However, this can be mitigated by delaying diesel injection timing, phasing the combustion later, without a large efficiency compromise.

1. Introduction

In the foreseeable future, the internal combustion engine (ICE) will continue to play an important role in powering transportation and industry due to its reliability and high efficiency [1]. The quest for higher efficiency and lower emissions has been driving the development of engine technology. In addition, the increasing global energy demand, and the concern for global warming have sparked the need for alternative fuels. Ethanol, in particular, is a promising alternative fuel due to its wide availability and established routes to production. Furthermore, bio-ethanol production from various wastes and agricultural residuals like woody feedstocks, corn stover, corn stock, rapeseed waste, sugarcane bagasse, citrus peel, leaves, and straws is possible [2,3]. In certain markets such as Brazil and Sweden, either neat ethanol or E85, respectively, have been commercially available for some time. In European Union countries and the United States, ethanol has been...
blended directly into conventional fuel up to 10% [3]. Thus, the interest in applying ethanol in different engine combustion concepts has emerged in the past decades. Investigations of utilizing ethanol, either in a fuel-blend or as a pure alternative fuel, have shown the benefits of reducing greenhouse gases, particulate matter, and carcinogens [4–6]. The level of improvement varies from engine to engine and depends on the operating range and, most importantly, the methodology used to inject ethanol into the combustion system.

The strategies of ethanol utilization in ICEs are various and require dedicated solutions. They can be mainly described in the following categories. A first category is ethanol fumigation into the intake air charge. It has been widely reported that ethanol fumigation can decrease soot and NOx emissions while increasing HC and CO emissions [7,8]. Notably, the reduction of NOx and soot emissions depend on engine load and the substitution rate of diesel respectively [9]. However, there is no consensus in the literature regarding its impacts on efficiency. In [10], an 88% reduction of NOx emissions at the cost of 18% loss in thermal efficiency was found, while in [8] a 3.6% increase of brake thermal efficiency was achieved. Most literature reports that the ethanol fumigation ratio is low (20% maximum) if the goal is to maximize the performance and minimize the emissions [9,10].

A second category is the total replacement of commercial gasoline/diesel via a port injection or direct injection. The high octane rating enables ethanol resistant to the auto-ignition, which makes it perfect for the spark-ignition (SI) engine and low temperature combustion (LTC) concepts in compression ignition (CI) engines to achieve high efficiencies. In [11,12], direct injection of ethanol (99.5 vol%) was applied in a heavy-duty (HD) engine running in partially premixed combustion (PPC). 56% peak gross indicated efficiency (GIE) was achieved with a combination of 32% exhaust gas recirculation (EGR) and 2.5 bar intake pressure. Negligible soot emissions and Euro VI level NOx were reported without any after-treatment. Besides, ethanol PPC operation was reported to be able to run in a near stoichiometric condition with low soot levels, which was unachievable with diesel fuel. Compared with diesel and gasoline PPC operation, ethanol experienced minimal efficiency reduction in moving from lean PPC mode to stoichiometric PPC. However, due to the long ignition delay and high latent heat, inlet heating (90–110 °C) was required to prevent misfire. Also, high pressure rise rates (PRR) were noticed due to the fast heat release from the premixed dominated combustion. Although these studies have shown superior soot-NOx characteristics without sacrificing efficiencies, the operating load range is limited. Furthermore, this strategy requires major changes in the hardware of engines to overcome the auto-ignition resistance of ethanol and a modified injection system due to its low lubricity [13].

A third category is blending ethanol with existing commercially available fuels. The advantage of ethanol fuel blending is that it can be used in engines without too many modifications to the engine itself. Compared with pure ethanol direct injection, the viscosity and lubricity of ethanol fuel blends do not change drastically. Most studies have illustrated the advantages of decreasing soot emissions when fueled with ethanol fuel blends [14]. Tse et al. investigated the combustion and emission characteristics of diesel-biodiesel-ethanol (DBE) fuel blends on a 4-cylinder naturally-aspirated diesel engine at 1800 RPM under five different loads. With increased ethanol ratio, the peak cylinder pressure and rate of heat release (ROHR) were higher due to a larger premixed combustion phase. The results showed that compared with ultra-low-sulfur diesel (ULSD), DBE can effectively reduce the soot emissions and particle numbers without increasing the NOx emissions [15]. In [16], 5 vol% (E5), 10 vol% (E10), and 15 vol% (E15) of anhydrous (99.8%) ethanol-diesel blends were studied at low, medium and high loads. The study revealed that the ignition delay increased at higher fuel oxygen mass content which consequently leads to later combustion phasing and decreased peak cylinder pressure. Compared with normal diesel operation, a higher brake efficiency was achieved at a lean combustible mixture (lambda = 5.5) but with higher PRR. For richer conditions, the addition of ethanol to diesel fuel reduced the NOx and the HC emissions. However, the brake specific fuel consumption was reported to be higher due to lower energy density. The influence of a higher ethanol mass content on CO emissions and smoke opacity depended on the air/fuel ratio and the engine speed. A downside of the blending approach is the polar nature of ethanol makes it easily miscible with water and thus lowers the upper blending limit with petroleum fuels. The water affinity also introduces challenges in the distribution infrastructure, which must prevent water contamination [3]. One of the challenges of ethanol fuel blend is phase separation due to the hydrophilic nature of ethanol. To avoid phase separation of ethanol/gasoline blends, either anhydrous ethanol or blending agents must be applied [17]. Ethanol is barely miscible with diesel. The solubility depends on the chemical composition of the diesel used, the temperature at which the blend is being made, and the percentage of ethanol present in the blend [18]. It requires the addition of either an emulsifier to suspend small droplets of ethanol within the diesel or a co-solvent that functions as intermediary via molecular compatibility and bonding to yield homogeneous mixture [19]. Moreover, the poor auto-ignition capacity of ethanol decreases the reactivity of the fuel blends and brings a cold start problem, misfire and knock when operated in a diesel engine [20]. Thus, a cetane improver is required when a high portion of ethanol is applied [14]. Last but not least, it was revealed in [21,22] that blending a volatile fuel like ethanol into diesel can potentially lead to a flammable headspace.

A fourth category is the utilization of ethanol in dual-fuel operation. Given that the ratio of port injected fuel and directly injected fuel can be controlled online based on the running conditions, this application can benefit from improved combustion and emission characteristics by combining both the merits of SI engines and CI engines. Yet, due to the complication of two fueling systems and the proper phasing of combustion, the operating conditions need to be fully understood. The effect of parameters such as engine load, speed, direct injection timing, port/direct fuel ratio, engine compression, and inlet conditions are well illustrated in the literature [23,24]. More recently, as is proposed by Reitz et al., the reactivity controlled compression ignition (RCCI) approach which shows more than 56% gross indicated thermal efficiency and near-zero NOx and PM emissions with port-injected (PFI) E85 and direct-injected (DI) cetane improved (addition of 3 vol% 2-EHN) gasoline in an HD engine [25]. The so-called reactivity gradient (the difference between octane number of PFI and DI fuel) is found to be crucial [26]. Considering the volatility of ethanol and the high reactivity gradient between ethanol and diesel, ethanol/diesel dual-fuel is also a possible option. In [27], ethanol/GTL dual-fuel operation was claimed to mitigate the NOx-soot tradeoff on a single-cylinder diesel engine whilst sacrificing HC and CO emissions. Pedrozo et al. optimized ethanol/diesel dual-fuel combustion in an HD engine at 1200 RPM, 6.15 bar gIMEP [28]. It was reported that split diesel injection adjusted the mixture flammability and promoted combustion efficiency. High ethanol substitution caused poor combustion efficiency particularly at low inlet temperature while low ethanol substitution did not demonstrate benefits in terms of GIE and emissions reduction. The results also showed that high injection pressure led to longer ignition delay and faster combustion, higher PRR and NOx emissions, while increased inlet pressure can effectively reduce local in-cylinder temperature thus lower NOx, but higher HC and CO emissions. Padala et al. investigated the effects of ethanol/diesel fraction on a single-cylinder automotive-size CI engine running at 9 bar gIMEP [29]. Up to 60% energy fraction of diesel can be replaced by ethanol with a 10% gain in efficiency compared to diesel-only operation. Specifically, the efficiency kept increasing with the increasing ethanol ratio because of a faster burning rate until the ignition delay became so long that combustion phasing was retarded so much that misfire started to occur. At a fixed ethanol ratio, advancing DI timing can further increase the efficiency due to the proper positioning of combustion. Negligible soot emissions were reported, but NOx, HC, and CO emissions were observed to increase.

The development of alternative fuels such as biofuel and alcohol...
fuel should be driven by existing engine technology. Thus, the possibility of applying ethanol with a commercially available injection system without jeopardizing the existing configuration is crucial for future implementation. These aforementioned studies have shown the advantages of ethanol utilization in CI engines both in terms of efficiency and emissions. Among them, ethanol/diesel dual-fuel operations show more flexibility in terms of operating load range compared to other strategies. Nevertheless, most of these promising results from ethanol/diesel dual-fuel operations require an optimized piston, combustion chamber, inlet heating or a completely different diesel injection control. Furthermore, they focus mainly on a very limited operational load range. There are very limited studies exploring the use of ethanol with conventional fuels without a need to change the standard, commercially available engine components and calibration. In order to investigate this knowledge gap, an experimental investigation has been carried out to study the potential of retrofitting a commercial HD diesel engine to allow dual-fuel operation. The main goal of this work is to investigate the maximum ethanol substitution ratio at different loads when the engine is running at the original diesel injection timings.

2. Experimental setup and methodology

2.1. Engine and apparatus

Fig. 1 shows the experimental setup, which is a single-cylinder research engine modified from a 6-cylinder HD diesel engine from the DAF. The specification of the test engine is displayed in Table 1. Cylinder one is the test cylinder which is isolated from the other cylinders except for the camshaft and crankshaft. Cylinder 2 and 3 are disabled, while Cylinder 3, 4 and 5 function as propelling cylinders to keep the engine at a steady speed with help of a hydraulic dynamometer, Schenk W450. The test cylinder is equipped with an external air compressor and electronic heater to control the inlet conditions. Fuel pressure is supplied by a double-acting air-drive Resato HPU200-625-2 pump, which can theoretically boost the fuel pressure up to 4000 bar. However, due to the restriction of the fuel line and injector, the pressure is limited to 2500 bar. The exhaust gas is first cooled and then mixed with fresh air in the surge tank. The other two tanks are used to reduce the pressure oscillation to achieve a steady flow. The gaseous emissions HC, CO, NOx, and CO2 are measured by a Horiba Meca 7100 DEGR emissions analyzer. Particulate matter is detected by an AVL 415s measuring filter smoke number on 3 samples in 60 s, which are first averaged and converted into g/kWh by applying the correlation proposed by Christian et al. [30]. The direct injection of diesel is via an injector provided by Delphi (Type: F2P). After the intake valve is open and the exhaust valve is closed (−300°CA aTDC), ethanol is injected through a port injector (Type: Vialle28) which is placed in the intake manifold with an angle of 120° resulting in a spray targeting on the intake valve. Port fuel pressure is set to 5 bar through the fuel pump in the port fuel tank.

Fig. 1. Schematic of the experimental setup.

Table 1

<table>
<thead>
<tr>
<th>Test engine specification.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Base engine</strong> DAF XE355c</td>
</tr>
<tr>
<td><strong>Stroke</strong></td>
</tr>
<tr>
<td><strong>Bore</strong></td>
</tr>
<tr>
<td><strong>Connecting Rod</strong></td>
</tr>
<tr>
<td><strong>Compression ratio</strong></td>
</tr>
<tr>
<td><strong>Number of Valves</strong></td>
</tr>
<tr>
<td><strong>Exhaust valve close (EVC)</strong></td>
</tr>
<tr>
<td><strong>Intake valve close (IVC)</strong></td>
</tr>
<tr>
<td><strong>Exhaust valve open (EVO)</strong></td>
</tr>
<tr>
<td><strong>Intake valve open (IVO)</strong></td>
</tr>
<tr>
<td><strong>Valve overlap</strong></td>
</tr>
<tr>
<td><strong>Injector</strong></td>
</tr>
<tr>
<td><strong>Actuating type</strong></td>
</tr>
<tr>
<td><strong>Nozzle holes</strong></td>
</tr>
<tr>
<td><strong>Nozzle diameter</strong></td>
</tr>
<tr>
<td><strong>Included spray angle</strong></td>
</tr>
</tbody>
</table>

The investigated fuels are ethanol with 2% cyclohexane and normal diesel, both of which are supplied by Shell. The major properties are tested and listed in Table 2. Testing is performed at a constant engine speed of 1200 RPM based on the European Stationary Cycle. Four different engine loads (A10, A30, A50, and A70) are selected which cover the range from low to high load. Before each test, the engine is warmed up until the oil and water temperature is above 80 °C. In-cylinder pressure is measured at 0.1°CA intervals by an AVL GU21C uncooled pressure transducer. 70 cycles of raw pressure data are sampled in each measurement. Crank angle, fuel pressure, injector current, and in-cylinder pressure traces are recorded and processed by a SMETEC Combi.
data-acquisition system. Data from inlet and exhaust sensors, together with air and fuel flows and emissions levels are recorded at 20 Hz for 40s employing an in-house data acquisition system (TUeDACs).

In the first part of this investigation, pure diesel (E0) is used, then the ethanol mass ratio (EMR) is varied from E0 to the maximum achievable at each load. The maximum EMR is defined by either misfire (in low load cases) or excessively high peak pressure and high PRMax (in high load cases). Since a retrofit solution is the major objective in this study, a single direct injection strategy of diesel is applied, where the start of diesel injection is based on the original engine setting. The injection duration is adjusted such that a constant engine load is maintained while the EMR increases. Similarly, the other operating parameters such as inlet conditions, exhaust pressure, EGR rate, and injection pressure are set according to the original calibration of this engine. This requirement is relaxed in the second part of this investigation, since the A50 and A70 cases experience a high PRMax when the EMR is 60% and 50% respectively. Thus, the DI timing of the diesel is retarded compared to the original calibration. The detailed settings of the test matrix are shown in Tables 3 and 4.

### 3. Data analysis

At each load, the total fuel energy input per cycle is fixed, which can be normalized by the so-called FuelMEP calculated with Eq. (1). The $m$ indicates the fuel mass per cycle, $LHV$ stands for the lower heating value and $V_d$ is the displacement of the test cylinder. The gross indicated mean effective pressure (gIMEP) is calculated based on Eq. (2). The calculation of EMR and ethanol energy ratio (EER) is based on Eqs. (3) and (4). At fixed inlet pressure, the increased ethanol mass could have two effects on the global lambda. On the one hand, since the energy density of ethanol is lower, the total fuel mass increases when the EMR increases to keep a fixed load. This decreases the air/fuel ratio. On the other hand, the stoichiometric air/fuel ratio of ethanol is much lower than that of diesel which decreases the overall stoichiometric air/ fuel ratio. The net result of these competitive effects leads to an overall increased global lambda for A10, A30, and A50 for an increasing EMR. The global lambda for A70 cases stays relatively stable as can be seen in Fig. 2.

\[
\text{FuelMEP} = \frac{m_{\text{ethanol}} \times LHV_{\text{ethanol}} + m_{\text{diesel}} \times LHV_{\text{diesel}}}{V_d}
\]  

\[
gIMEP = \frac{\int_0^{180} P \times dV}{V_d}
\]  

\[
EMR = \frac{m_{\text{ethanol}}}{m_{\text{ethanol}} + m_{\text{diesel}}} \times 100\%
\]  

\[
EER = \frac{EMR}{EMR + \frac{\text{LHV}_{\text{diesel}}}{\text{LHV}_{\text{ethanol}}}} \times 100\%
\]

The recorded cylinder pressure data is averaged for 70 cycles and filtered with a Savitzky–Golay filter (Order: 1, Frame length: 19). Both the pressure rise rate and the ROHR (Eq. (5)) are computed from this average and filtered pressure trace.

\[
ROHR = \frac{\gamma - 1}{\gamma - 1} \frac{dV}{d\theta} + \frac{1}{\gamma - 1} \frac{V}{\frac{dP}{d\theta}}
\]

where $\gamma$ is the specific heat capacity ratio determined using the Eq. (6) [31]:

\[
\gamma = 1.35 - 6 \times 10^{-4}T + 1.1 \times 10^{-8}T^2
\]

$T$ is the global temperature calculated from cylinder pressure based on the ideal gas law. Cylinder pressures are pegged at the bottom dead center (BDC) using the measured average inlet pressure. Combustion efficiency, GIE, and thermal efficiency are calculated based on Eqs (7)–(9) respectively.

\[
\eta_{\text{combustion}} = \left(1 - \frac{\text{ISHC} \times LHV_{\text{fuel}} + \text{ISCO} \times LHV_{\text{CO}} + \text{ISH}_{2} \times LHV_{\text{H}_2}}{\text{ISFC} \times LHV_{\text{fuel}}} \right) \times 100\%
\]  

\[
\eta_{\text{GIE}} = \frac{\text{gIMEP}}{\text{FuelMEP}} \times 100\%
\]  

\[
\eta_{\text{thermal}} = \frac{\eta_{\text{GIE}}}{\eta_{\text{combustion}}} \times 100\%
\]

In this paper, combustion phasing is related to CA50, the crank angle where 50% of accumulated heat is released. The start and end of combustion are referred to CA10 and CA90 respectively, the crank angles where 10% and 90% of heat is released. The start of injection (SOI) is determined by the start of actuation (SOA) with a 0.2 ms delay time. Ignition delay is defined as the crank angle difference between CA10 and SOI. While burn duration (BD) is defined as the crank angle difference between CA90 and CA10. The EGR rate is determined from the measured ratio of CO2 from the inlet air and exhaust gas. All measurements are done three times to assess the repeatability. The results shown in the following sections are the average value of these repetitions. Since the variation is found to be negligible, the error bar is eliminated for the figures in Sections 4.1 and 4.2 for the sake of clarity. For the figures in Section 4.3, the error bar is presented.

#### Table 2

<table>
<thead>
<tr>
<th>Properties</th>
<th>Ethanol + 2%cyclohexan</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>52.88 %wt (Shell proprietary in-house method)</td>
<td>86.5 %sm/m [ASTD5291(Method C)]</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>13.16 %wt (Shell proprietary in-house method)</td>
<td>13.5 %sm/m [ASTD5291(Method C)]</td>
</tr>
<tr>
<td>Oxygen</td>
<td>33.96 %wt (Shell proprietary in-house method)</td>
<td>0</td>
</tr>
<tr>
<td>Net calorific Value</td>
<td>27.42 MJ/kg (Shell proprietary in-house method)</td>
<td>43.053 MJ/kg [ASTD5291(Method C)]</td>
</tr>
<tr>
<td>Density at 15 °C</td>
<td>792.9 kg/m3 (ASTM D4052*)</td>
<td>–</td>
</tr>
</tbody>
</table>

#### Table 3

<table>
<thead>
<tr>
<th>Test Matrix</th>
<th>A10</th>
<th>A30</th>
<th>A50</th>
<th>A70</th>
</tr>
</thead>
<tbody>
<tr>
<td>gIMEP [bar]</td>
<td>3.4±0.05</td>
<td>9±0.05</td>
<td>14±0.05</td>
<td>19±0.2</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>1000 bar</td>
<td>1550 bar</td>
<td>1900 bar</td>
<td>1950 bar</td>
</tr>
<tr>
<td>Start of Actuation (Diesel)</td>
<td>aTDC</td>
<td>aTDC</td>
<td>aTDC</td>
<td>aTDC</td>
</tr>
<tr>
<td>EGR rate</td>
<td>25%</td>
<td>26%</td>
<td>24.5%</td>
<td>20%</td>
</tr>
<tr>
<td>Intake temperature (°C)</td>
<td>40</td>
<td>45</td>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>Intake pressure</td>
<td>1.1 bar</td>
<td>1.6 bar</td>
<td>2.3 bar</td>
<td>2.9 bar</td>
</tr>
<tr>
<td>Exhaust pressure</td>
<td>1.4 bar</td>
<td>1.75 bar</td>
<td>2.67 bar</td>
<td>3.2 bar</td>
</tr>
<tr>
<td>Engine speed</td>
<td>1200 RPM</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
4. Results and discussion

4.1. Combustion characteristics

Fig. 3a shows the tested the maximum EMR at each load. The operating loads, represented by the gIMEP, are kept nearly constant at different EMR. A10, A30, A50, A70 are the 4 different common loads for a typical HD truck, corresponding to 3.4 bar, 9 bar, 14.4 bar, and 19.9 bar gIMEP respectively. The detailed energy ratio, mass ratio, and amount of ethanol and diesel in each cycle is shown in Fig. 3b and c. At A10 and A30, the EMR can be as high as 80% without misfiring. This is confirmed by the coefficient of variance based on gIMEP (COV$_{gIMEP}$) shown in Fig. 4a. The engine actually runs stable and well within 5% COV$_{gIMEP}$ [32] for all cases and hardly dependent on the substitution ratio. Only the COV$_{gIMEP}$ at the lowest load, A10, keeps increasing as more ethanol is applied. This is mainly because, under these lean combustion conditions, a small change in the mixture can bring a large variation in the combustion process [33]. This is particularly the case in HCCI combustion, the prevalent case at high EMR values, where large cycle-to-cycle variations constrain the operation range [34]. As the load increases, the maximal EMR generally decreases due to the restriction of the high pressure rise rate and peak pressure which will be elaborated on later.

Apart from cyclic variation, another limitation for ethanol/diesel dual-fuel combustion is engine knock, which refers to end-gas auto-ignition before the combustion by flame propagation is finished in spark ignition (SI) engines. It causes localized rapid pressure rise that excites acoustic resonance frequencies in the cylinder [35]. The detection of knock is quite critical and requires well-trained hearing [36]. Thus, a dedicated criterion derived from the cylinder-pressure such as the pressure rise rate (PRR) is commonly employed to adjust the operating parameters to maintain the engine running in a smooth state. Generally, PRRmax should be kept below a certain level to prevent severe engine noise, deterioration of performance, and hardware failure. However, in a compression ignition engine where the combustion depends on the auto-ignition, much higher pressure rise rates are allowed due to more robust engine construction. Moreover, some studies report that PRR (bar/°CA) alone cannot provide consistent results when operating parameters such as speed or boost level change [35]. When knock happens in an HCCI-like engine, high-amplitude ripples appear on the in-cylinder pressure curve and the corresponding spectral power content increases. Hence the ringing intensity (RI) is calculated based on the equation proposed in [36], which correlates with the acoustic energy of the resonating pressure and quantifies the tendency to produce acoustic oscillations. At A10 and A30, PRRmax is not problematic, even at high EMR, as it remains at relatively low levels. As the load increases, high pressure rise rates are observed at high EMR. This correlates with the fact that more premixed combustion is observed and the burn duration becomes shorter. A10 though shows a distinctively different trend compared to the other three operating loads. PRRmax of A10 decreases as the EMR increases, while it increases for A30, A50, and A70. This is mainly because the CA50 of A10 retards while that of the other operating loads tend to advance (Fig. 6c). What is striking in Fig. 4b is that A50 and A70 show a high value of PRRmax even at medium (~40%) EMR. This is likely due to knock, and it limits the substitution ratio at high

### Table 4

Test matrix of optimization of combustion phasing.

<table>
<thead>
<tr>
<th></th>
<th>A50E60</th>
<th>A70E50</th>
</tr>
</thead>
<tbody>
<tr>
<td>gIMEP [bar]</td>
<td>14.4±0.05</td>
<td>19.9±0.2</td>
</tr>
<tr>
<td>Direct injection pressure [bar]</td>
<td>1900</td>
<td>1950</td>
</tr>
<tr>
<td>Diesel injection timing [%CA aTDC]</td>
<td>-6.6/-5.6/-4.5/-3.5/-2.5/-1.5</td>
<td>-5.5/-4/-2.8/-1.5/0/1.5</td>
</tr>
<tr>
<td>EGR rate [%]</td>
<td>24.5</td>
<td>20</td>
</tr>
<tr>
<td>Intake temperature [°C]</td>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>Intake pressure [bar]</td>
<td>2.3</td>
<td>2.9</td>
</tr>
<tr>
<td>Exhaust pressure [bar]</td>
<td>2.67</td>
<td>3.2</td>
</tr>
<tr>
<td>Engine speed [RPM]</td>
<td>1200</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 2. Global lambda vs premixed lambda at different operating loads.

Fig. 3. Operating load vs EMR (a), EER vs EMR (b), and diesel vs ethanol per cycle (c).
The computed ringing intensity shows the same trends as PRRmax. As is shown in Fig. 4c, A10 and A30 illustrate low ringing intensity under the well-accepted threshold 5 MW/m² [35]. When the load increases to A50 and A70, the ringing intensity exceeds 5 MW/m² after E40. Since both PRRmax and RI increase at high EMR, it would seem to be prudent to restrict the EMR below 40% for A50 and A70 when operated at default diesel settings.

Fig. 5 demonstrates the cylinder pressure and ROHR profiles at different operating loads. It can be seen that the pressure traces before combustion starts are noticeably lower for high EMR for all load cases. This is caused by the ‘cooling effects’ of ethanol which has a high latent heat of vaporization. The global cylinder temperature before ignition decreases as EMR increases, as is in Fig. 14 in Appendix. It is well-understood that RCCI combustion is initiated by the highly reactive fuel and extends subsequently to the well-mixed port-injected low reactivity fuel [37]. The engines in typical RCCI studies are generally running with a high port injection ratio. Thus, it is most likely in this investigation that the homogeneously mixed ethanol is too lean to ignite by compression heat. Hence, followed by the auto-ignition of diesel, combustion propagates to the well-mixed ethanol/air mixture afterwards. Although burn durations are relatively short, which hints towards auto-ignition. It is difficult to distinguish whether this is flame propagation or auto-ignition. Nevertheless, it is noteworthy that the effects of EMR on combustion characteristics vary at different loads. At A10, the engine is running in a lean condition as is shown in Fig. 2. Both the global and premixed lambda are high even with an inlet pressure as low as 1.1 bar. Due to the low in-cylinder pressure and temperature, a complete separation between direct diesel injection and combustion events occurs. The induction of ethanol will decrease the cylinder temperature further before combustion, which decreases the reactivity of mixture and leads to longer mixing time, delayed combustion phasing and extended burn duration. Consequently, the in-cylinder

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**Fig. 4.** COVgIMEP (a), PRRmax (b), and ringing intensity (c) vs EMR.

**Fig. 5.** Cylinder pressure and ROHR at different EMR and operating load.
Increasing load leads to longer diesel injections, higher in-cylinder pressure, and temperature. Running diesel only, combustion starts before the injection is finished for A30, A50, and A70. This can be characterized by the ignition dwell which is defined as the crank angle difference between CA10 and end of diesel injection (EOI). A positive ignition dwell indicates a complete separation between diesel injection and combustion events. It can be noticed from Fig. 6b that ignition dwell generally decreases as the load increases, which is obvious as the injection duration increases. In addition, the ignition dwell increases as a function of EMR, regardless of the operating load, mainly for two reasons: DI injection becomes shorter and charge temperature becomes lower (Fig. 14 in Appendix). It is interesting to find that a distinct two-stage heat release phenomenon is found at A30 for the E70 case (Fig. 5). The rest of the ROHR profiles of A30 are shown in Fig. 15 in Appendix. Apparently, the remaining bulk ethanol-air charge auto-ignites shortly after the large pre-mixed diesel/ethanol/air mixture has increased the bulk temperature. Notwithstanding the widely different ROHR profiles, the combustion phasing appears to be unaffected until the ethanol ratio reaches 60%, after which the CA50 is delayed, and ignition dwell becomes positive as is shown in Fig. 6b, in the A30 cases. This starts where the dual peak behavior appears to become more prominent in the ROHR. The weight of the combustion shifts more and more to the second peak which ignites later in the cycle.

As the load increases further to A50 and A70, ignition dwell never becomes positive. Pure diesel operation at A50 and A70 (A50E0 and A70E0) only shows a small premixed burn and a large mixing-controlled combustion phase. For all other cases, the well-mixed ethanol and air mixture are still too lean and cold to be ignited before the injection of diesel even though the charge temperature during compression is higher than that of the A10 and A30 cases (as is shown in Fig. 14 in Appendix). When diesel is injected into these well-mixed mixtures of ethanol and air, it will ignite easier though than in the A30 and A10 cases. The premixed combustion increases significantly as the EMR increases, which leads to the decreased burn duration in Fig. 6d. The ROHR profile for A50E70 suggests that the bulk of the ethanol/air charge auto-ignites rapidly after the injection of the diesel, leading to a short burn duration. Compared to the A30 case, the bulk ignites closer to the diesel injection which is thought to be due to the higher reactivity of the bulk (higher temperature and ethanol concentration). Consequently, both the peak of ROHR and cylinder pressure increase as shown in Fig. 5. Particularly, 200 bar peak in-cylinder pressure is observed at A70E50, which almost reaches the upper limit of this engine configuration. For this reason, only ethanol substitutional mass ratios below 50% are investigated at A70.

### 4.2. Emissions and efficiency

Fig. 7 shows the indicated specific emissions under the current operating conditions. NOx emissions are quite sensitive to local temperature both in terms of magnitude and the time spent in a high temperature area. It is generally accepted that NOx emissions can be controlled by EGR which brings down the combustion temperature. In this investigation, a moderate EGR rate is applied based on the default calibration. It can be seen that there is no consistent or clear trend concerning the influence of ethanol ratio on NOx emissions. Specifically, at A10, NOx emissions decrease as more ethanol is induced. This could be mainly attributed to the increased ignition delay and dwell at high EMR. It causes the mixture to approach more and more globally
homogeneous thus, therefore largely avoid the formation of NO\textsubscript{x}. In addition, CA50 gets delayed at high EMR, which decreases the local temperature and the time available at high temperature. For A30, A50, and A70, NO\textsubscript{x} emissions tend to decrease first and increase afterward with EMR. This is also likely to be caused by the local cylinder temperature which firstly decreases due to evaporation of ethanol and then increases again due to a larger proportion of premixed combustion and advanced CA50. NO\textsubscript{x} emissions of A30E80 and A50E70 increase by a maximum of 25% compared with pure diesel combustion. NO\textsubscript{x} emissions, however, are expected to be lower when a high EGR rate is applied. The enhancement of EGR rate also decreases soot oxidation and increases engine-out soot emissions. Thus, the key to overcome the soot-NO\textsubscript{x} tradeoff under a high EGR rate is to avoid soot formation. At the default engine calibration, pure diesel operation already shows negligible engine-out soot emissions due to high injection pressure. As is displayed in Fig. 7b, the addition of ethanol can decrease soot emissions even further regardless of the load. When more fuel is introduced via port injection during the intake stroke, more and more fuel burns in a fully pre-mixed mode which is always lean resulting in less soot regardless of the operating load. Soot can be only be formed due to the direct injection of diesel fuel, which will become less prominent as EMR increases. Furthermore, as can be seen in Fig. 2, the premixed lambda has a minimum of 2.4 due to the oxygen contained in the ethanol molecule. It is noteworthy that this makes ethanol a perfect alternative fuel to be applied with high EGR rates to achieve the simultaneous reduction of soot and NO\textsubscript{x} emissions. Investigation of this effect is however beyond the scope of this paper.

An important finding from Fig. 7 is that HC and CO emissions increase significantly at high EMR. Notably, HC emissions increase almost linearly with EMR from A30 to A70. This is expected since more fuel can be trapped in the crevice volume when the EMR increases. Thus, incomplete combustion occurs due to the flame not penetrating these areas. The unburnt mixture in the crevice is then released in the exhaust stroke and is one of the major sources of unburnt HC [38]. At A10, HC emissions increase much more rapidly. In addition to fuel capture in the crevice volume, the combustible mixture at A10 is more likely to experience bulk quenching due to the overall very lean ambient condition and low in-cylinder temperature. As is suggested in [39], unburnt HC originating from the cylinder wall and the piston top quench layer are of great importance. Also, the retarded combustion phasing of A10 decreases the cylinder temperature even further as the EMR increases. This leads to an even larger increase in HC emissions at A10. In particular at A10E80, the HC emissions almost reach the limit of the Horiba system, which prevents data acquisition at higher ethanol substitutional ratios. HC emissions are observed to decrease at high load because of the increased cylinder temperature but are still much higher as compared to pure diesel combustion. CO emissions are formed mainly in fuel-rich conditions due to incomplete combustion. The dual-fuel operation can induce a high gradient of both reactivity and equivalence ratio, which makes the direct-injected diesel burn in a fuel-rich condition. Consequently, CO emissions increase as a function of EMR. Furthermore, due to the low cylinder temperatures for the A10 case, poor oxidation of CO to CO\textsubscript{2} becomes a major issue. Interestingly, the trend found for the CO emissions of A30, A50, and A70 is opposite to those found for NO\textsubscript{x}, both being explained by the variation of the cylinder temperature. Firstly, CO emissions increase due to a decreased cylinder temperature and then decrease due to improved oxidation at an increased cylinder temperature.
Further to note, it was pointed out in [40] that a higher light-off temperature of the diesel oxidation catalyst (DOC) is required in RCCI operation than in traditional diesel operation due to a higher level of HC and CO emissions and a different chemical composition of RCCI hydrocarbons. As is shown in Fig. 8, the exhaust temperature of A10 is above 250 °C. As the load increases to A30 and above, the exhaust temperature is higher than 380 °C. Most studies have proven that the conversion efficiency of the catalyst for HC and CO emissions is above 90% when the exhaust temperature is higher than 300 °C [40,41]. It is believed that most of the HC and CO emissions from A30 to A70 can be effectively removed by the DOC in real engine situations where HD trucks are equipped with this after-treatment system. It is important to note that the exhaust temperature is in this investigation is measured about 15 cm away from the cylinder head in the exhaust manifold. The temperature of exhaust gas would be lower when it enters the after-treatment system due to thermal energy loss in the turbocharger in a real engine situation. More experimental and simulation work are necessary to understand the conversion rates of HC and CO emissions, especially at the A10 load condition.

The combustion efficiency is calculated based on measured emissions according to Eq. (7). Following the increase of HC and CO emissions, Fig. 9 shows that combustion efficiency decreases with increasing EMR. Specifically, for A10, there is a substantial reduction in combustion efficiency (from 99 to 85.7%). It indicates that ethanol/diesel dual-fuel combustion at low load suffers from incomplete combustion under the default calibration mainly due to a retardation of combustion (CA50 changes from 5 to 10°CA aTDC). For the same reason, the effective expansion ratio decreases and consequently thermal efficiency is observed to decrease as well. Possible ways to improve the combustion process of ethanol/diesel dual-fuel combustion at low load could be by applying inlet heating, adapting the EGR cooling strategy or advancing the direct injection of diesel (advanced combustion phasing) [29,42]. Given the poor performance of ethanol/diesel dual-fuel combustion at low load under the default diesel calibration, it may be advisable to apply a low ethanol substitution ratio or even revert to pure diesel operation at cold start, idle and low load. Under these operating conditions, pure diesel operation shows much better robustness and higher efficiency. In other words, the flexibility of tuning the ratio of port-injected ethanol and direct-injected diesel according to operating conditions is one of the advantages of dual-fuel combustion over other combustion concepts.

As the load increases, the dual-fuel operation starts to show some output gain over pure diesel operation. Despite the lower combustion efficiency, a steady rise in thermal efficiency is shown at A30, from 49.1% (E0) to 50% (E80), as is displayed in Fig. 9b. The thermal efficiency of A50 stands out by increasing from 49.1% (E0) to 50.7% (E60), 1.6% gain. At E70 the thermal efficiency of A50 cases decreases. Under this operating load, increasing EMR shifts CA50 earlier and closer to TDC. Early combustion phasing could contribute to a higher effective expansion ratio and faster heat release. However, the aforementioned high PRMax at high EMR of A50 cases induces a rapid increase of PRR, causing a substantial increase in heat transfer loss and a decrease in thermal efficiency [43,44]. It can be seen from the energy balance analysis (Fig. 9c) that the heat transfer loss indeed gradually increases as EMR increases. These competing factors indicate an objective calibration is required if optimum thermal efficiency is to be pursued. In other words, combustion phasing needs to be shifted to a later position to control PRR not only for avoiding knock but also maximizing the work extraction potential. For the same reason, an increasing and then decreasing trend of thermal efficiency is found at A70. As is demonstrated in Fig. 9b, a decrease is observed after A70E20, wherein thermal efficiency rapidly decreases from 49.7% (E20) to ~48% (E30-E50).

4.3. Effects of diesel injection timing

As is postulated, ethanol/diesel dual-fuel combustion at high load appears to suffer from high PRMax and ringing intensity. This knocking tendency limits the EMR at high load mainly due to the improper combustion phasing. To address this issue more in detail, the default diesel injection timing in this section is adjusted to achieve suitable combustion phasing with A50E60 and A70E50. As is shown in Fig. 10, the actuation of the DI pulse (SOA) of A50E60 and A70E50 is swept from −6.6 to −1.5°CA aTDC and from −5.5 to 1.5°CA aTDC respectively. As expected, CA50 responds almost linearly to the delay of the diesel injection which shows that the control of combustion phasing can be manipulated easily through the direct injection pulse of the diesel. This is another advantage for ethanol/diesel dual-fuel combustion over ethanol utilization in an HCCI strategy where the combustion is fully kinetically controlled. Furthermore, the sensitivity of the
response of combustion phasing also contributes to an optimum running condition during a transient state.

Fig. 11 shows the in-cylinder pressure and ROHR profiles of A50E60 and A70E50 at different combustion phasing. Both the peak of in-cylinder pressure trace and ROHR profiles decrease and the timing of the peak is delayed. Specifically, the peak cylinder pressure of A70E50 decreases from an extremely high 200 bar (CA50: 4.1°CA aTDC) to 153 bar (CA50: 12.7°CA aTDC). Additionally, the slower heat release (wider ROHR profile) at late combustion elongates the burn duration as is shown in Fig. 10b. This late combustion phasing also has a dramatic effect on PRRmax and ringing intensity. As is shown in Fig. 12, both PRRmax and ringing intensity can be controlled below the safety threshold when the combustion phasing is after 7.8°CA aTDC. It is noted that in [29], where misfire due to over-retarded combustion phasing was the major limiting factor. It was reported a higher EMR can be achieved by advancing the diesel injection timing at 9 bar IMEP. Nevertheless, both results indicate that with adjusted diesel injection timing, the engine can be operated safely with a high EMR at a specific operating load.

The increased engine stability with the delayed combustion phasing does not come without sacrifices. Fig. 13 shows that combustion efficiency decreases gradually as CA50 shifts to a later position. This is due to the lower charge temperature and HC and CO emissions increase when combustion moves more into the expansion stroke. As mentioned in the previous section, thermal efficiency increases first and then decreases as CA50 is delayed. Both of the selected test points show an optimum CA50. Thus, dedicated calibration is necessary to balance the tradeoff between the limitation of pressure rise rate and the potential loss in efficiency. Obviously, ethanol/diesel dual-fuel combustion presents more freedom to manipulate the combustion phasing concerning efficiency, emissions, and PRRmax as compared to the other strategies of applying ethanol. However, compared with pure diesel operation at A50 (GIE: 49.3%) and A70 (GIE: 49%), delayed diesel injection timings of A50E60 and A70E50 results in lower GIE when safety limits are considered. To be specific, the GIE of A50E60 decreases from 50.2% (CA50: 5°CA aTDC) to 49.2% (CA50: 8°CA aTDC). While GIE of A70E50 decreases from 48.2% (CA50: 5.7°CA aTDC) to 47.8% (CA50: 8.9°CA aTDC) (Fig. 15).

5. Conclusions

This work investigated the ethanol/diesel dual-fuel combustion in a single cylinder of an HD commercially available engine. The main goal of this paper is to explore the operating range and the maximum ethanol substitution ratio that could be achieved without modification of the standard engine specifications and calibration. In essence, all tests were performed with the default diesel calibration with respect to intake conditions and DI timing. Four different operating loads from low to high were tested with different EMRs. Pure diesel operation was also tested to provide a reference case. Based on the combustion and emission characteristics, the following conclusions can be drawn:

1. At low load A10, the low in-cylinder temperature and lower charge density result in low reactivity of the ethanol-air mixture. The increased EMR prolongs ignition delay and retards combustion phasing due to the high latent heat of vaporization. This leads to a decreased effective expansion ratio, a longer burn duration and consequently a decreased thermal efficiency.
2. Although the EMR can be up to 80% at A10 and A30 without misfire, low EMR is suggested for low load conditions since ethanol/diesel dual-fuel combustion generally suffers from poor combustion completeness. This jeopardizes combustion efficiency significantly and constrains the maximal EMR. High HC and CO emissions occur due to ethanol trapped in the crevice volume. All tested points show extremely low soot emissions with a decreasing trend as a function of EMR in general because more fuel is homogenously mixed and, by definition, lean at higher EMR. Although rises and decreases are noted, no consistent effects of ethanol on NOx emissions can be demonstrated at different operating loads.

3. As the load increases, the global lambda decreases at the current engine calibration and less HC and CO emissions are generated. Due to the hotter and denser cylinder charge, combustion phasing shifts earlier at higher EMR for A50 and A70. The shortened burn duration and faster heat release are observed. This yields a higher PRRmax and ringing intensity, which limits the application of ethanol/diesel dual-fuel combustion to 40% EMR for A50 and A70 cases. The thermal efficiency mainly depends on combustion phasing and is load specific. At A30, a steady increase in thermal efficiency is noticed as EMR increases, while for A50 and A70, the thermal efficiency shows an optimum as a function of EMR.

4. Good control of combustion phasing via the direct injection timing of diesel is shown for A50E60 and A70E50. The knocking tendency at high load can be mitigated by late diesel injection timing to maintain late CA50 with a small sacrifice in GIE (A50E60: 50.2% to 49.2%, A70E50: 48.2% to 47.8%). Both PRRmax and ringing intensity are below the threshold value when combustion phasing is after 7.8°CA aTDC, which indicates that a high EMR can be used even at high loads. However, a dedicated calibration is required to maximize the indicated efficiency yet maintain low PRRmax and ringing intensity.

From a practical point of view, HD trucks will still be powered by diesel engines in the short term due to its proven robustness, established technology, and well-developed infrastructure. The ability to retrofit available diesel engines with alternative fuels without significant modification becomes crucial in future applications. The work presented in this paper sheds some insights into the viability of ethanol/diesel dual-fuel combustion on an HD diesel engine. This application of ethanol appears to be promising in terms of thermal efficiency gain and soot reduction potential, most important of all, flexibility in operating range and good control of combustion phasing. Like most LTC concepts, high HC and CO emissions are shown to be the downside of this solution. As is postulated, the conversion efficiency of a DOC at low load conditions remain to be investigated and will inspire future work.

Author contributions

Roger Cracknell and Robert Wardle conceived the presented idea. Jinlin Han designed and carried out the experiments. The post-process and data analysis was mainly performed by Jinlin Han under the supervision of L.M.T. Somers. And Jinlin Han took the lead in writing the manuscript. All authors provided critical feedback and helped shape the research, analysis, and manuscript. Both Arndt Joedicke and Vivek Raja Raj Mohan contributed to the final version of the manuscript.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.
Appendix

Figs. 14 and 15.

Fig. 14. Averaged global temperature at different operating load and EMR.

Fig. 15. ROHR of A30.
References


