High efficiency ethanol-diesel dual-fuel combustion: A comparison against conventional diesel combustion from low to full engine load

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Keywords

Ethanol; low carbon fuel; dual-fuel combustion; diesel engine; greenhouse gas emissions, tank-to-wheels.

Highlights

- High efficiency dual-fuel combustion was demonstrated between 0.3 and 2.4 MPa IMEP.
- Up to 4.4% higher net indicated efficiency than conventional diesel combustion.
- Up to 90% lower nitrogen oxides emissions using identical engine testing conditions.
- The use of ethanol as a substitute for diesel can reduce greenhouse gas emissions.

Abstract

Comparisons between dual-fuel combustion and conventional diesel combustion (CDC) are often performed using different engine hardware setups, exhaust gas recirculation rates, as well as intake and exhaust manifold pressures. These modifications are usually made in order to curb in-cylinder pressure rise rates and meet exhaust emissions targets during the dual-fuel operation. To ensure a fair comparison, an experimental investigation into dual-fuel combustion has been carried out from low to full engine load with the same engine hardware and identical operating conditions to those of the CDC baseline. The experiments were executed on a single cylinder heavy-duty diesel engine at a constant speed of 1200 rpm and various steady-state loads between 0.3 and 2.4 MPa net indicated mean effective pressure (IMEP). Ethanol was port fuel injected while diesel was direct injected using a high pressure common rail injection system. The start of diesel injection was optimised for the maximum net indicated efficiency in both combustion modes. Varied ethanol energy fractions and adaptive diesel injections were required to control the in-cylinder pressure rise rate and achieve highly efficient and clean dual-fuel operation. In terms of performance, the dual-fuel combustion attained higher net indicated efficiency than the CDC mode from 0.6 to 2.4 MPa IMEP, with a maximum of 47.2% at 1.2 MPa IMEP. The comparison also shows the use of ethanol resulted in 26% to 90% lower nitrogen oxides (NOx) emissions than the CDC operation. At the lowest engine load of 0.3 MPa IMEP, the dual-fuel operation led to simultaneous low NOx and soot emissions at the expense of a relatively low net indicated efficiency of 38.9%. In particular, the reduction in NOx emissions introduced by the utilisation of ethanol has the potential to decrease the engine running costs via lower consumption of aqueous urea solution in the selective catalyst reduction system. Moreover, the dual-fuel combustion with a low carbon fuel such as ethanol is an effective means of decreasing the use of fossil fuel and associated greenhouse gas emissions.

1. Introduction

Heavy-duty (HD) vehicles are typically powered by diesel engines due to their cost-effectiveness and high fuel conversion efficiency. However, there is a lot of concern over the greenhouse gas (GHG) emissions produced from the combustion of diesel and other fossil fuels [1]. This is due to a recent increase in the atmospheric concentration of GHGs such as carbon dioxide (CO_2) [2], which can lead to irreversible changes in climate and cause impacts on natural and human systems on all continents and across the oceans [1].

In 2010, HD vehicles were responsible for approximately 34% of the GHGs emitted by the global transport sector and 46.5% of the road transport CO_2 emissions [3]. The disproportionate contribution is highlighted by the fact the HD fleet represents only 11% of the world motor vehicles [4]. Substantial and sustained reductions in fossil fuel energy use and GHG emissions have to be attained in order to address the transport sector's impact on the environment.

Additionally, conventional diesel combustion (CDC) incurs a wide range of local in-cylinder gas temperatures and fuel/air equivalence ratios that can lead to the formation of noxious emissions, such as NOx and soot [5,6]. NOx emissions are mainly formed in near-stoichiometric high temperatures regions close to the diesel diffusion flame [7]. Soot formation occurs in high fuel/air equivalence ratio and intermediate temperature zones within the diesel spray [8,9]. These pollutants are linked to premature deaths caused by cardiovascular and respiratory diseases [10,11].

Stringent fuel conversion efficiency and exhaust emissions regulations have been implemented to limit the levels of GHG and noxious emissions from HD vehicles [12][13][14][15]. Manufactures are incorporating costly engine design elements [16–20] and aftertreatment technologies [21,22] to comply with these emissions standards while achieving the GHG reduction targets [12][13]. Some examples are the use of more robust selective catalyst reduction (SCR) systems for NOx mitigation, flexible and high pressure diesel injection equipment, as well as high efficiency turbocharging and air handling systems.

A balance between engine running costs and exhaust emissions can represent a challenge for HD engine manufactures with the use of both advanced in-cylinder and aftertreatment measures [16][17]. An improvement of 1% in fuel conversion efficiency can increase the levels of engine-out NOx from 10 g/kWh to 14 g/kWh [18]. This adversely affects the total cost of ownership due to a higher consumption of aqueous urea solution in the SCR system [23–26]. On the other hand, CDC operation with very low engine-out NOx emissions can result in low fuel conversion efficiency and excessive levels of soot due the different formation mechanisms [27,28].

Previous studies into dual-fuel compression ignition combustion have demonstrated the strategy has the potential to tackle these issues, increasing the fuel conversion efficiency while decreasing both the NOx and soot emissions [6][29][30][31][32]. This has been attributed to simultaneous reductions in local fuel/air equivalence ratios, combustion temperatures, and heat transfer losses [6][32].

Figure 1 shows an example of a dual-fuel system, which can be achieved by the installation of a port fuel injection system of a low reactivity fuel such as gasoline [32], ethanol [33], or natural gas [34] on a diesel engine. The ignition of the premixed charge is generally triggered by direct injections of diesel [6][35]. It should be noted that the use of a low carbon fuel like ethanol [36][37][38][39] can help decrease the dependence on fossil fuels and minimise GHG emissions from the global transport sector [40].

Despite the advantages of the dual-fuel operation, it is often challenging to obtain direct comparisons against the CDC mode from low to high engine loads (e.g. above 2.0 MPa IMEP).

This is due to modifications in engine hardware and/or test conditions that help control the emissions of NOx and the in-cylinder pressure rise rates from dual-fuel combustion. These alterations typically include the use of a different piston design and/or compression ratio [41][42] as well as changes in the levels of exhaust gas recirculation [43].

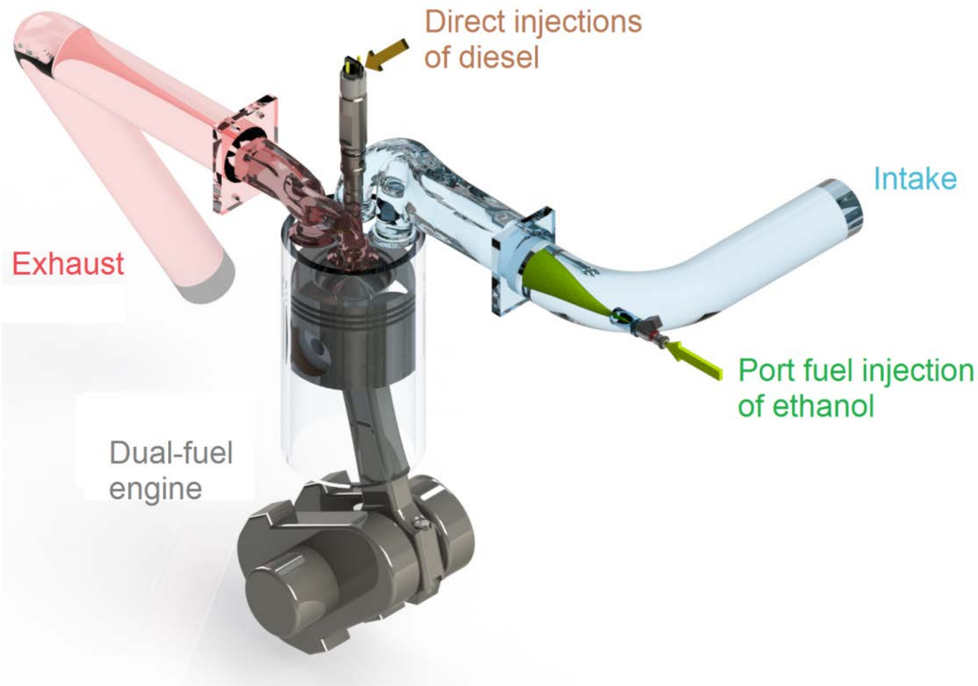

Figure 1 – Schematic diagram of a dual-fuel engine with direct injections of diesel and port fuel injection of ethanol.

This study aims at exploring the potential of dual-fuel combustion to achieve high fuel conversion efficiency and low exhaust emissions using the same combustion system and identical engine testing conditions to those employed by the CDC baseline.

To the best of our knowledge, this is the first attempt to experimentally compare the controllability, emissions, and fuel conversion efficiency of ethanol-diesel dual-fuel operation to those of the CDC mode from low (0.3 MPa IMEP) to full engine load (2.4 MPa IMEP). Moreover, practical considerations have been raised and the potential CO_2 reduction has been discussed on both a tank-to-wheels and well-to-wheels basis [37][44].

The investigation was performed on a single cylinder HD diesel engine at a steady-state speed of 1200 rpm. The diesel injection timings and the number of injections per cycle were optimised in both the combustion modes in order to maximise the fuel conversion efficiency, which was given by the net indicated efficiency. In addition, the dual-fuel operation was carried out using ethanol energy fractions that achieved the highest net indicated efficiency with minimal NOx and soot emissions, as determined in our previous studies [29][30][31][45][46].

2. Experimental setup

2.1. Experimental facilities

A schematic diagram of the single cylinder HD engine experimental setup is shown in Figure 2. A Froude Hofmann AG150 eddy current dynamometer was used to absorb the power produced by the engine. Fresh intake air was supplied to the engine via an AVL 515 sliding vanes compressor with a closed loop control for the boost pressure. A throttle valve located upstream of a large-volume surge tank provided fine control over the intake manifold pressure. The fresh air mass flow rate ($\dot{m}_{air}$) was measured with an Endress+Hauser Proline t-mass 65F thermal mass flow meter.

Another surge tank was installed in the exhaust manifold to damp out pressure fluctuations prior to the exhaust gas recirculation (EGR) circuit. An electronically controlled butterfly valve located downstream of the exhaust surge tank was used to set the required back pressure (e.g. exhaust manifold pressure). High-pressure loop cooled external EGR was supplied to the engine intake system by opening a pulse width modulation-controlled EGR valve. Boosted intake air and external EGR temperatures were controlled using water cooled heat exchangers.

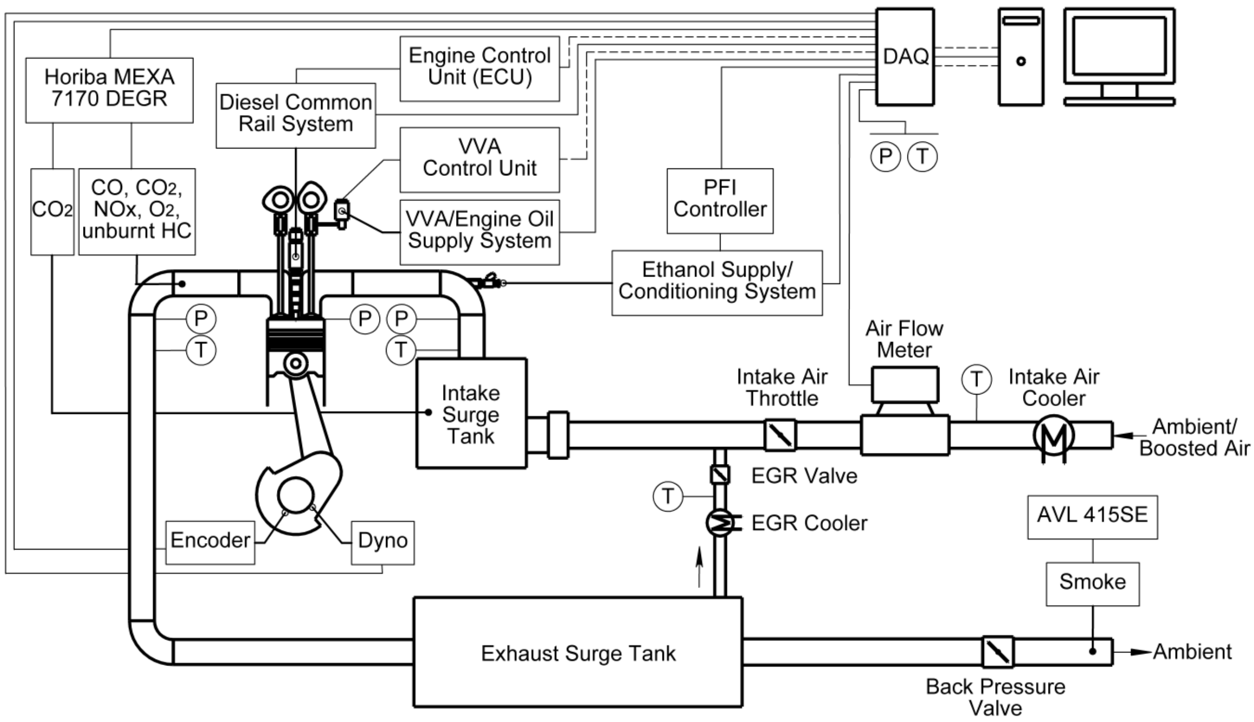

Figure 2 – Schematic diagram of the engine experimental setup.

2.2. Engine specifications

The single cylinder HD engine was equipped with port fuel injection of ethanol and high pressure common rail direct injection of diesel. The combustion system consisted of a 4-valve cylinder head and a stepped-lip piston bowl design with a geometric compression ratio of 16.8. Base hardware specifications are outlined in Table 1.

The diesel injections were controlled via a dedicated engine control unit (ECU) with the ability to support up to three shots per cycle. The intake valve lift profile was adjusted via a lost-motion variable valve actuation (VVA) system based on a normally open high-speed solenoid valve assembly and a special intake cam design [47].

Coolant and oil pumps were not coupled to the engine and were driven by separate electric motors. Engine coolant and oil temperatures were set to 353 ± 3 K. The oil pressure was held at 450 ± 10 kPa throughout the experiments.

Table 1 – Single cylinder HD engine specifications.

Parameter	Value
Displaced volume	2.026 dm^3
Stroke	155 mm
Bore	129 mm
Connecting rod length	256 mm
Number of valves	4
Piston type	Stepped-lip bowl
Geometric compression ratio	16.8
Peak in-cylinder pressure (P_{max}) limitation	18 MPa
Diesel Injection System	Bosch common rail, injection pressure of 50–220 MPa, 8 holes with nominal diameter of 0.176 mm, included spray angle of 150°
Ethanol Injection System	PFI Marelli IWP069, included spray angle of 15°

2.3. Fuel properties and delivery

The relevant properties of the fuel used in this work are listed in Table 2. The diesel fuel was supplied to the engine using a high pressure common rail injection system. Two Endress+Hauser Promass 83A Coriolis flow meters were used to determine the diesel mass flow rate ($\dot{m}_{diesel}$) by measuring the total fuel supplied to and from the diesel high pressure pump and injector.

In order to allow for dual-fuel operation, an ethanol fuel injection system was designed and fitted to the engine. Ethanol was injected through a port fuel injector (PFI) installed in the intake manifold. An in-house injector driver controlled the injector pulse width, which was adjusted according to the desired ethanol energy fraction.

The ethanol mass flow rate ($\dot{m}_{ethanol}$) was measured using an Endress+Hauser Proline Promass 80A Coriolis flow meter, allowing for measurements with an accuracy of 0.15%. The injection pressure was continuously monitored by a pressure transducer, so that a constant relative pressure of 300 kPa could be maintained across the injector.

Table 2 – Fuel properties.

Property	Diesel	Ethanol
Product name	Red diesel (gas oil)	Absolute ethanol 100
Standard/specification	BS 2869 Class A2	Anhydrous ethanol
Density at 293 K (ρ_{fuel})	0.827 kg/dm^3	0.790 kg/dm^3 [48]
Cetane number	> 45	n/a
Research octane number (RON)	n/a [49]	~107 [49]
Alcohol content in volume	n/a	99.9%
Water content	< 0.20 g/kg [50]	1.7 g/kg [48]
Sulphur content	< 0.01 g/kg	n/a
Heat of vaporisation	270 kJ/kg [49]	840 kJ/kg [49]
Carbon mass content ($\%C_{fuel}$)	86.6%	52.1% [49]
Hydrogen mass content ($\%H_{fuel}$)	13.2%	13.1% [49]
Oxygen mass content ($\%O_{fuel}$)	0.2%	34.8% [49]
Normalised molecular composition	$CH_{1.825}O_{0.0014}$	$CH_3O_{0.5}$
Lower heating value (LHV_{fuel})	42.9 MJ/kg	26.9 MJ/kg [49]

2.4. Exhaust emissions measurements and analysis

An AVL 415SE smoke meter was used for soot emissions measurements downstream of the back pressure valve. Gaseous emissions such as NOx, CO_2, carbon monoxide (CO), oxygen (O_2), and unburnt hydrocarbon (HC) were taken with a Horiba MEXA-7170 DEGR emissions analyser. The EGR rate was determined by calculating the ratio of the intake to the exhaust manifold CO_2 concentration measured by the same emissions analyser. A high pressure module allowed for high-pressure sampling upstream of the back pressure valve while a heated line was used to prevent condensation.

The measurement of unburnt HCs was performed on a wet basis by the Horiba's heated flame ionisation detector (FID). However, the HC emissions measured with the FID can lead to misinterpretation of the results due to the relative insensitivity of the device towards alcohols and aldehydes [51,52]. Therefore, the FID response was corrected by the method developed by Kar and Cheng [51] with an updated response factor of 0.68 for the oxygenated organic species resultant from ethanol combustion [52]. This procedure has been reported in our previous work [31] and allows for the determination of the actual unburnt HC emissions.

Finally, the exhaust emissions measurements were converted to net indicated specific emissions using the methodology described in the Regulation number 49 of the Economic Commission for Europe of the United Nations [50]. The concentrations of CO and NOx were converted to a wet basis by applying a correction factor for the raw exhaust gas according to the in-cylinder fuel mixture composition.

2.5. Data acquisition

The in-cylinder pressure was measured by a Kistler piezoelectric pressure sensor Type 6125C. Under mechanical load, crystals in the sensor produced an electrostatic charge, which was converted into an electric potential difference by means of an AVL FI Piezo charge amplifier. Intake and exhaust manifold pressures were measured by two Kistler water cooled piezoresistive absolute pressure sensors Type 4049A coupled to Kistler amplifiers Type 4622A. Temperatures and pressures at relevant locations were measured by K-type thermocouples and pressure gauges, respectively.

Two National Instruments data acquisition (DAQ) cards and a personal computer were used to acquire the signals from the measurement device. An USB-6251 high speed DAQ card received the crank angle resolved data synchronized with an optical encoder of 0.25 crank angle degrees (CAD) resolution. An USB-6210 low speed DAQ card acquired the low frequency engine operation conditions. These data were displayed live by an in-house developed DAQ program and combustion analyser.

2.6. Data analysis

A relevant parameter for the dual-fuel operation was the ethanol energy fraction (EF), which was defined as the ratio of the energy content of the ethanol to the total fuel energy supplied by

$$EF = \frac{\dot{m}_{ethanol}\, LHV_{ethanol}}{(\dot{m}_{ethanol}\, LHV_{ethanol}) + (\dot{m}_{diesel}\, LHV_{diesel})} \tag{1}$$

The excess of fuel in the exhaust gas was given by the global fuel/air equivalence ratio (Φ), which was calculated as

$$\phi = \frac{(14.5\, \dot{m}_{diesel} + 9.0\, \dot{m}_{ethanol})}{\dot{m}_{air}} \tag{2}$$

Crank angle based in-cylinder pressure traces were averaged over 200 consecutive cycles for each operating point and used to calculate the IMEP and the apparent net heat release rate (HRR). The pressure rise rate (PRR) was represented by the average of the maximum pressure variations of 200 cycles of cylinder pressure versus crank angle. Combustion and in-cylinder flow stability were monitored by the coefficient of variation of IMEP (COV_IMEP) and P_{max} (COV_P_{max}) over the sampled cycles.

Since the absolute value of the heat released is not as important to this study as the bulk shape of the curve with respect to crank angle, a constant ratio of specific heats (γ) of 1.33 was assumed throughout the engine cycle. The mass fraction burnt (MFB) was given by the ratio of the integral of the HRR and the maximum cumulative heat release. Combustion phasing was determined by the crank angle of 50% (CA50) cumulative heat release. Combustion duration was represented by the period of time between the crank angles of 10% (CA10) and 90% (CA90) cumulative heat release.

A current probe was used to acquire the electric current signal sent from the ECU to the diesel injector solenoid. The signal was corrected by adding the respective energising time delay, which was previously measured in a constant volume chamber [53]. The resulting diesel injector current signal allowed for the determination of the actual start of diesel injection.

Ignition delay was defined as the period of time between the actual start of main diesel injection (SOI_main) and the start of combustion (SOC), set to 0.3% MFB point of the averaged cycle. After the calculation of the combustion characteristics (e.g. CA50) and ignition delay, the average in-cylinder pressure and the resulting HRR were smoothed using a Savitzky-Golay filter.

Net indicated efficiency was determined by calculating the ratio of the work done to the rate of fuel energy supplied to the engine. Combustion efficiency calculations were based on the emissions products not fully oxidised during the combustion process except soot.

3. Methodology

Figure 3 shows the location of the test points over an estimated speed and load map of a HD diesel engine. Testing was carried out under a steady-state engine speed of 1200 rpm over a range of loads from 0.3 to 2.4 MPa IMEP. A PRR of 2.0 MPa/CAD and a P_{max} of 18 MPa were considered as the upper bounds for calibration. Stable engine operation was quantified by COV_IMEP values less than 5%.

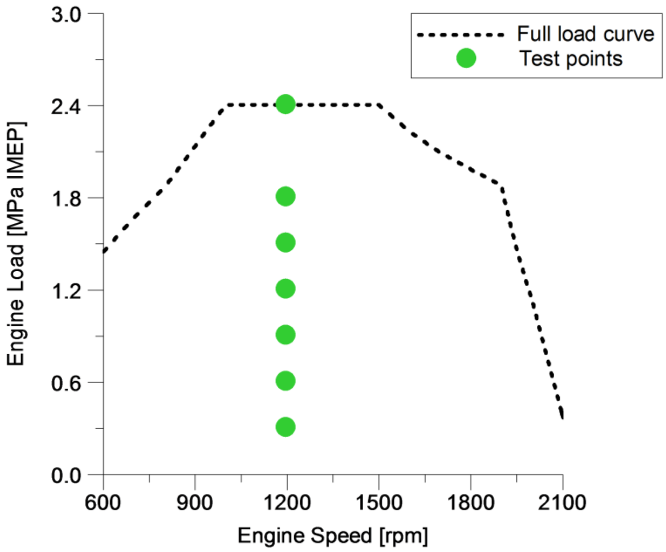

Figure 3 – Experimental test points over an estimated HD diesel engine speed-load map.

Table 3 summarises the test conditions for the CDC and ethanol-diesel dual-fuel operating modes. The experiments were performed using a pressure-based effective compression ratio of 16.8 [46]. The expansion ratio remained constant as a result of the fixed exhaust camshaft timing.

Table 3 – Operating conditions for the CDC and ethanol-diesel dual-fuel operation from low to full engine load at 1200 rpm.

Engine load	Intake manifold pressure	Exhaust manifold pressure	Intake manifold air temp.	EGR rate	Diesel injection pressure (CDC)	Diesel injection pressure (Dual-fuel)
MPa IMEP	kPa	kPa	K	%	MPa	MPa
0.3	115	125	307	25	105	50
0.6	125	135	310	25	125	90
0.9	155	165	315	25	140	110
1.2	190	200	319	25	155	125
1.5	230	240	324	25	170	140
1.8	260	270	324	20	190	160
2.4	300	310	323	11	220	190

The intake manifold pressure set point was taken from a Euro V compliant multi-cylinder HD diesel engine in order to provide a sensible starting point, since an external boosting device was used in place of a turbocharger. The exhaust manifold pressure was varied to maintain a constant pressure differential across the cylinder of 10 kPa. This allowed for exhaust gas recirculation, which was used to curb NOx formation.

The EGR rate was limited at 25% between 0.3 and 1.5 MPa IMEP to avoid excessive smoke and a decrease in net indicated efficiency. At 1.8 and 2.4 MPa IMEP, the EGR rate was reduced to 20% and 11%, respectively. This was essential in order to achieve lean and efficient high load operations using the same levels of boost pressure of the multi-cylinder engine.

Diesel injection pressures were set to be 30 to 55 MPa higher in the CDC mode than those in the dual-fuel combustion due to the relatively higher diesel flow rates and longer injection durations at a given engine load. This was necessary to minimise soot emissions from the CDC operation via improved diesel atomisation and enhanced the fuel-air mixing process.

All comparisons were carried out for the cases that attained the highest net indicated efficiencies after sweeps of diesel injection timings. Additionally, the diesel injection strategy (i.e. number of diesel injections per cycle) was optimised and varied as the engine load was increased.

In the dual-fuel mode, the ethanol energy fraction was also optimised for minimum NOx and soot emissions, as supported by our previous dual-fuel studies [29][30][31][45][46]. A maximum EF of 0.79 was achieved at 1.2 MPa IMEP. Advanced dual-fuel combustion control strategies such as the internal exhaust gas recirculation (iEGR) [31] and Miller cycle [46] were not explored in this study as they would require different test procedures.

4. Results and Discussion

4.1. Overview of the load sweep

Figure 4 depicts the effect of engine load on both the operating modes. CDC operation was characterised by longer mixing-controlled combustion phase as the load was increased. This was attributed to longer diesel injection periods and increased amount of fuel, which limited the fuel vapour-air mixing process [49,54]. The optimum CA50 in CDC mode varied as the engine load was increased, allowing for more advanced burn rates at mid-loads and delayed combustion events at high loads. The reasons behind this are described in the next subsection.

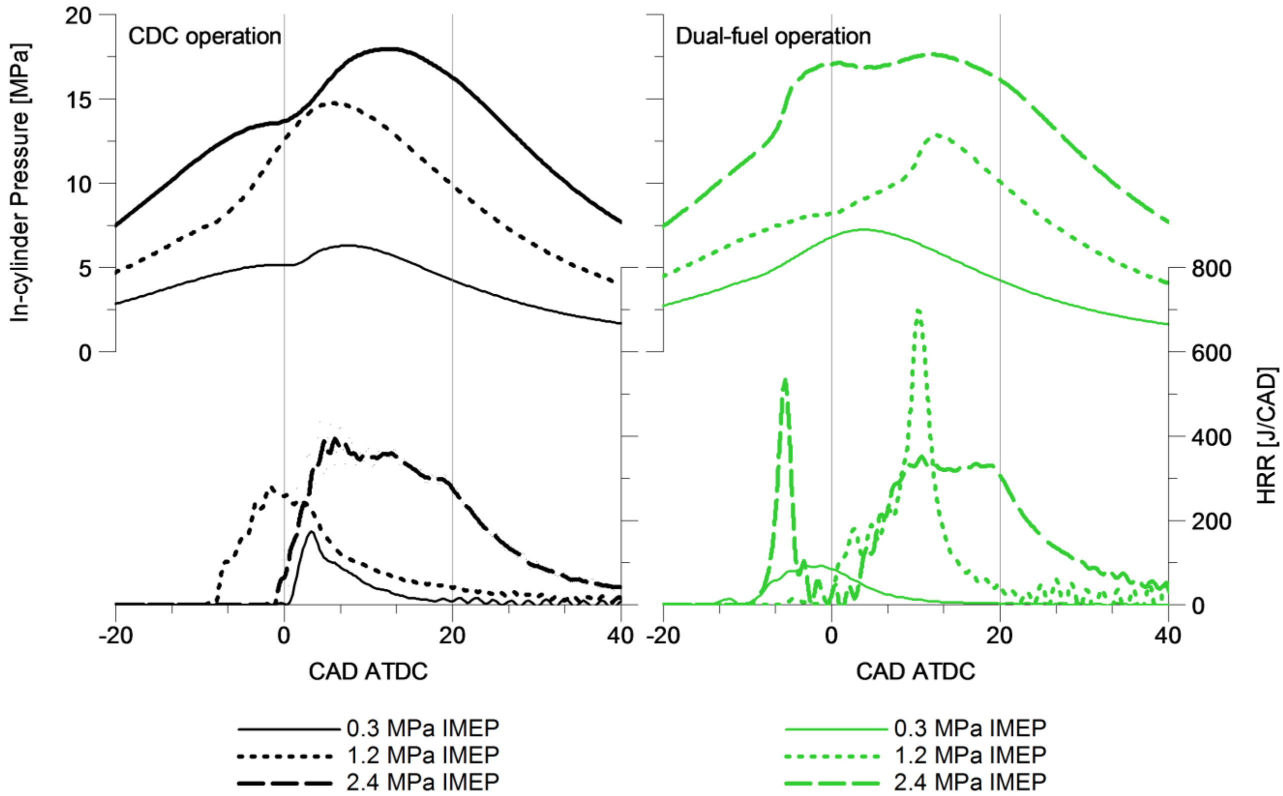

Figure 4 – The effect of engine load on CDC and ethanol-diesel dual-fuel operation at 1200 rpm.

The dual-fuel operation led to higher peak heat release than the CDC mode at all engine loads except 0.3 MPa IMEP. This required different diesel injection strategies and eventually later combustion process in order to control the PRRs as the engine load was increased. The combustion was triggered by and initiated after the diesel injection at low and medium load operations between 0.3 and 1.5 MPa IMEP. Higher compression pressures and temperatures accelerated the autoignition of the premixed ethanol fuel prior to the diesel injection at high engine loads of 1.8 and 2.4 MPa IMEP.

Figure 5 shows the optimum EF had to be rapidly reduced from 0.76 to 0.25 when increasing the engine load from 1.5 to 1.8 MPa IMEP. This was necessary in order to minimise the PRRs associated with the early autoignition of ethanol. It is important to bear in mind that modifications in the engine hardware (e.g. lower effective compression ratio via Miller cycle) and/or test procedure (e.g. lower intake manifold air temperature) can increase the maximum EF at higher loads [46].

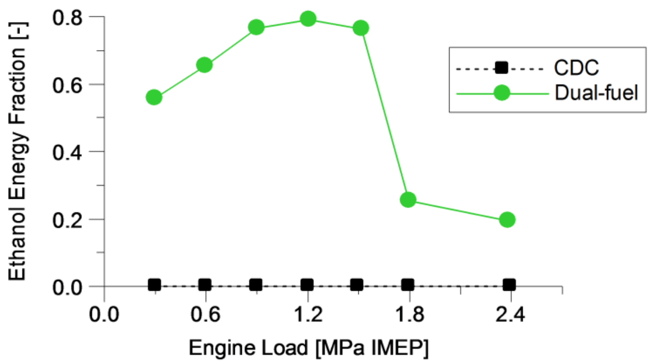

Figure 5 – Optimum ethanol energy fraction for varied engine loads at 1200 rpm.

4.2. Combustion control

Figure 6 shows the actual start of diesel pre-injection (SOI_pre), SOI_main, and in-cylinder pressure characteristics for optimised CDC and dual-fuel operation. In the CDC mode, a 3 mm^3 diesel pre-injection with a constant dwell time of 1 ms was used to reduce the levels of PRR [45] between the engine loads of 0.3 MPa IMEP and 1.5 MPa IMEP. The lower PRRs were associated with the shorter ignition delay produced by the combustion of the diesel pre-injection and likely formation of a hot and reactive mixture prior to the main diesel injection [55].

At high engine loads of 1.8 and 2.4 MPa IMEP, relatively shorter ignition delays introduced by lower EGR rates and higher in-cylinder pressures and temperatures allowed for the use of a single diesel injection near firing top dead centre (TDC). The maximum SOI_main advance was limited by the P_{max} while the PRRs were maintained within the limit of 2.0 MPa/CAD.

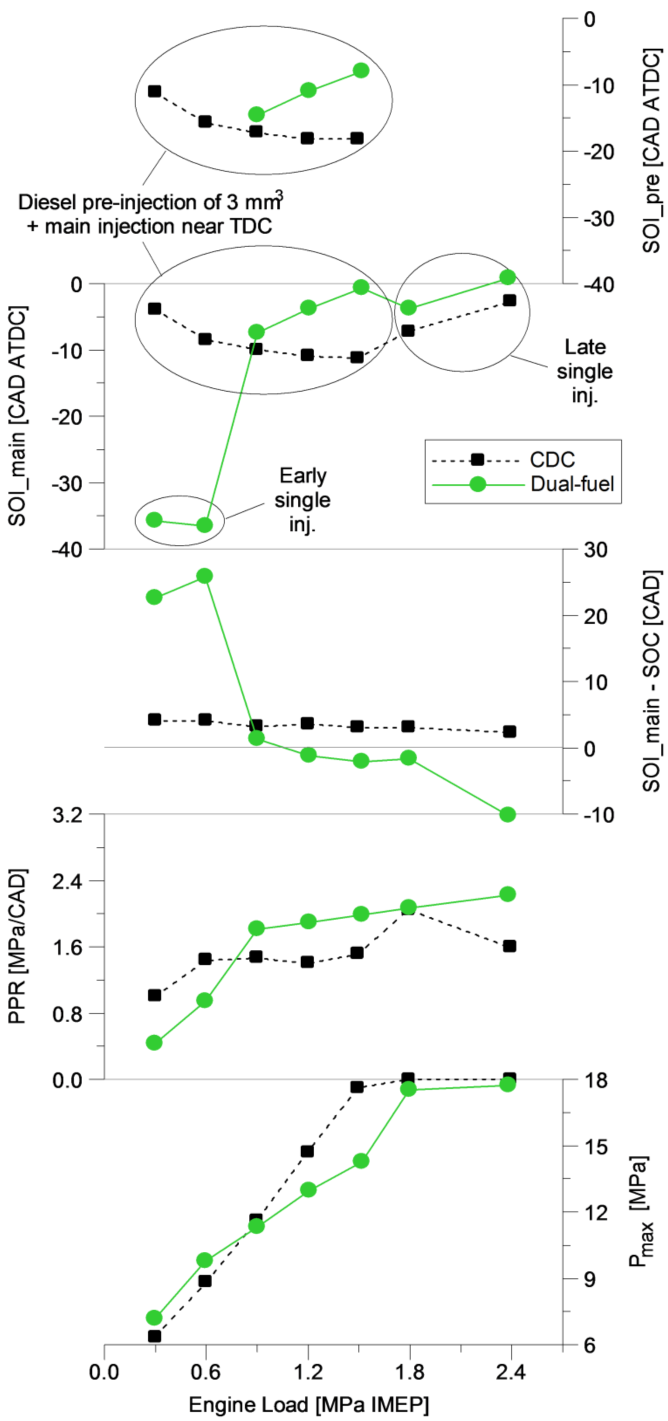

Figure 6 – Diesel injection timings and combustion characteristics for optimised CDC and ethanol-diesel dual-fuel operation at 1200 rpm.

In the dual-fuel operation, the combination of an early single diesel injection at about -36 CAD after top dead centre (ATDC) and EFs of 0.56 and 0.65 allowed for long ignition delays (SOI_main–SOC) and better mixture preparation at 0.3 and 0.6 MPa IMEP. This enhanced the combustion process via a more progressive and probably sequential combustion from high to low reactivity regions [8]. This has also been identified in computational simulations performed by Desantes et al. [56] and is supported by the low levels of PRR. However, the P_{max} was increased when compared to that of the CDC operation due to earlier CA50 and shorter combustion for the dual-fuel mode at these particular loads (see Figure 7).

At mid-loads between 0.9 and 1.5 MPa IMEP, less partially premixed diesel fuel could be used in order to prevent an early ignition of the in-cylinder charge. Therefore, the mass of the diesel was divided into two direct injections using the same strategy employed in the CDC cases. The injection of a small amount of diesel prior to the SOI_main was essential to mitigate excessive PRRs. This was a result of a shorter SOI_main–SOC period and elimination of the premixed combustion peak typically observed with a late single diesel injection strategy [45]. Despite the controlled levels of PRR, the diesel injection timings were delayed by up to 10.5 CAD when compared against those of the CDC operation, helping lower the P_{max} levels.

At 1.8 and 2.4 MPa IMEP, the premixed ethanol fuel autoignited prior to the diesel injection. Lowe ethanol energy fractions and a single diesel injection near TDC were used to control the burn rate as well as the resulting PRR and P_{max}. The introduction of a diesel pre-injection would increase the PRR levels at these loads due to simultaneous and early combustion of the ethanol and pre-injected diesel fuel.

4.3. Heat release analysis

Figure 7 depicts the heat release characteristics for the CDC and dual-fuel operation. The optimum CA50 for the maximum net indicated efficiency was initially advanced and then retarded in the CDC mode. The advance in the CA50 position was likely linked to the short CA10–CA90 period and relatively higher heat transfer losses at 0.3 MPa IMEP. The delay was associated with the peak in-cylinder pressure limitation at high load operations of 1.8 and 2.4 MPa IMEP. Additionally, lower levels of EGR and possibly higher combustion temperatures helped shorten the CA10–CA90 periods of the CDC operation at these high load conditions.

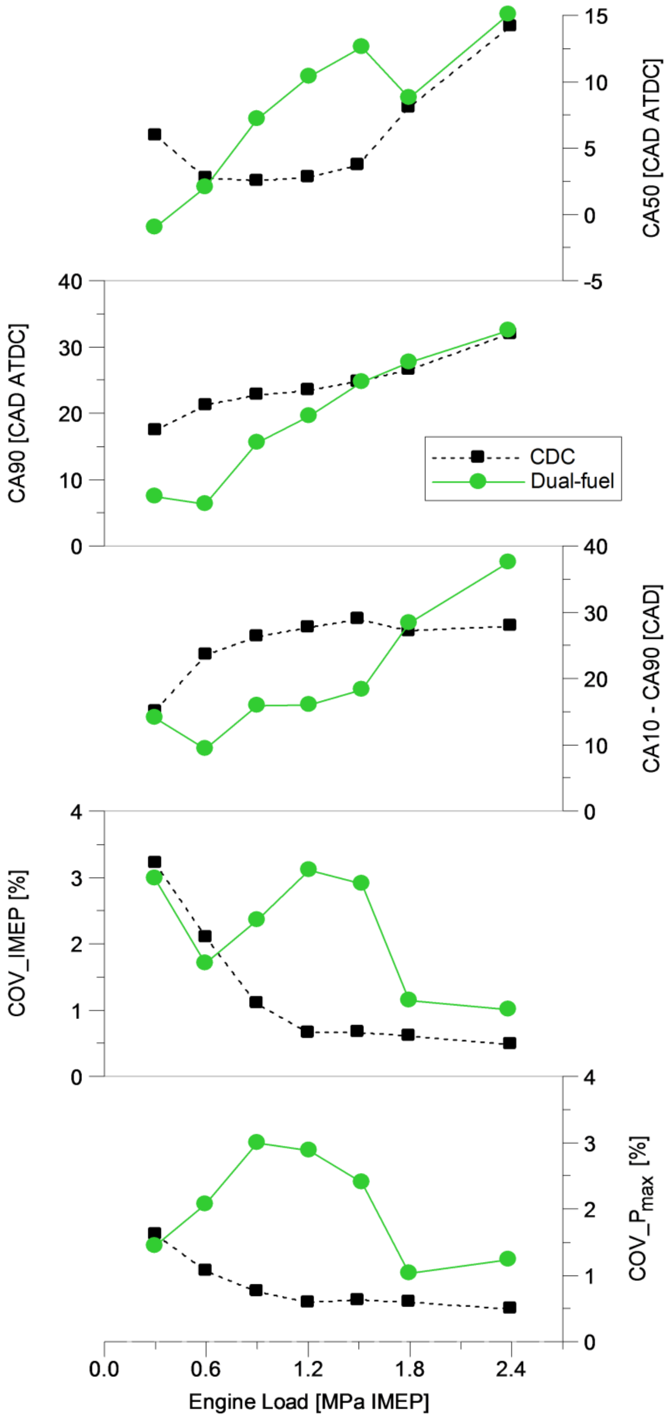

Figure 7 – Heat release characteristics for optimised CDC and ethanol-diesel dual-fuel operation at 1200 rpm.

In comparison, the dual-fuel operation often required later CA50s as the engine load was increased in order to avoid excessive PRRs. At high loads of 1.8 and 2.4 MPa IMEP, the CA50 and CA90 positions were similar for both the combustion modes due to the P_{max} limitation of 18 MPa and lower EFs used in the dual-fuel mode.

In general, the increase in engine load generally led to later CA90s and longer CA10–CA90 as a result of the higher fuel flow rates. The higher degree of premixed combustion in the dual-fuel mode was likely the cause for the relatively earlier CA90s and faster CA10–CA90 periods between 0.3 and 1.5 MPa IMEP. Nonetheless, the early ignition of the ethanol fuel produced longer burn rates than the CDC operation at 1.8 and 2.4 MPa IMEP.

In terms of combustion stability, the mixing-controlled combustion of the CDC operation effectively decreased the COV_IMEP and COV_P_{max} to 0.5% as the engine load was increased to 2.4 MPa IMEP. In the dual-fuel mode, later CA50s and a more premixed combustion yielded higher levels of COV_IMEP between 0.9 and 2.4 MPa IMEP. In addition, the dual-fuel operation resulted in higher COV_P_{max} at all engine loads except 0.3 MPa IMEP. Nevertheless, the COV_IMEP and COV_P_{max} could be controlled between 1.0% and 3.0%.

4.4. Engine-out emissions

Figure 8 shows the net indicated specific emissions for the optimum cases over a sweep of load. An EGR rate of 25% was used to minimise NOx emissions at engine loads up to 1.5 MPa IMEP. This allowed for a CDC operation with net indicated specific emissions of NOx (ISNOx) of 3.9 g/kWh, on average, between 0.3 and 1.5 MPa IMEP. The use of lower EGR rates of 20% and 11% increased the combustion temperatures at 1.8 and 2.4 MPa IMEP, yielding higher ISNOx of 4.4 and 5.7 g/kWh, respectively.

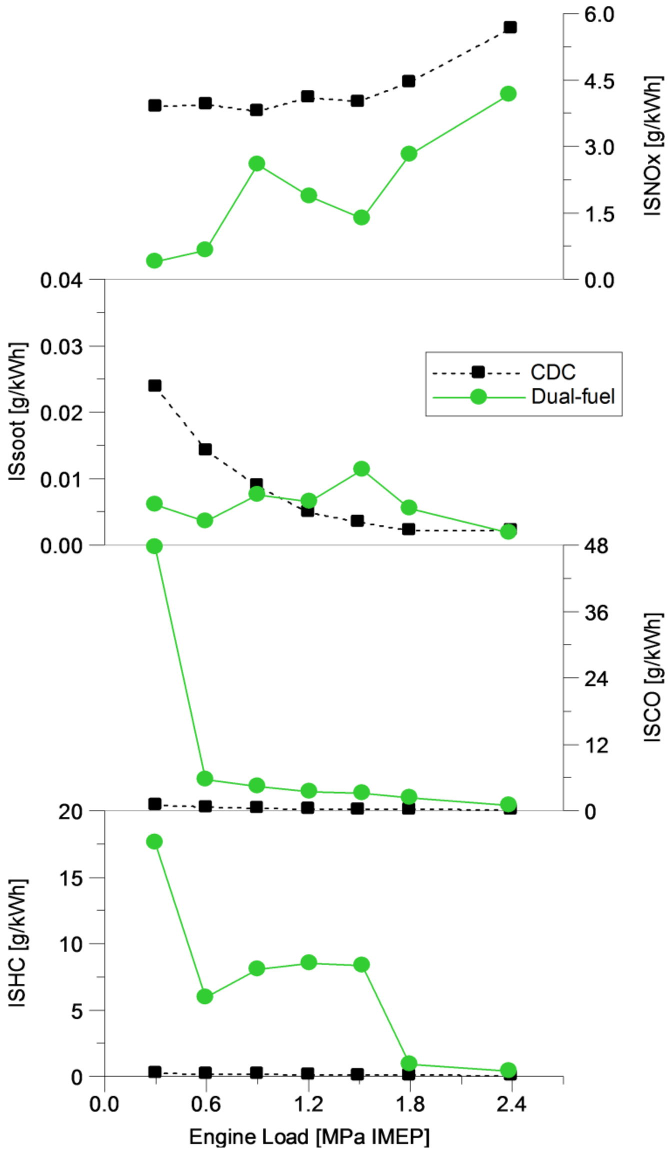

Figure 8 – Net indicated specific emissions for optimised CDC and ethanol-diesel dual-fuel operation at 1200 rpm.

Alternatively, the optimised dual-fuel operation achieved lower ISNOx than the CDC mode at all engine loads. This was linked to the premixed ethanol fuel, which probably helped decrease the amount of in-cylinder regions of high combustion temperature. Reductions in NOx emissions varied from 26% at 2.4 MPa IMEP up to 90% at 0.3 MPa IMEP for EFs of 0.19 and 0.56, respectively.

The lowest levels of ISNOx were attained at 0.3 and 0.6 MPa IMEP due to longer ignition delays and relatively more homogenous combustion process when compared against the other dual-fuel cases with diesel injections closer to TDC. NOx emissions were decreased when increasing the engine load from 0.9 to 1.5 MPa IMEP due to later optimum CA50s and potentially lower combustion temperatures. At high loads of 1.8 and 2.4 MPa IMEP, the ethanol autoignition process and shorter diesel mixing-controlled combustion helped reduce the peak in-cylinder gas temperatures [46], decreasing the ISNOx when compared to the CDC operation.

In the CDC mode, higher diesel injection pressures and in-cylinder gas temperatures helped curb soot emissions as the engine load was increased. In comparison, net indicated specific emissions of soot (ISsoot) were maintained consistently low in the dual-fuel operation because of reduced regions of fuel rich combustion, particularly at 0.3 and 0.6 MPa IMEP. This is a significant improvement over the CDC cases considering the dual-fuel combustion employed lower diesel injection pressures, as explained in Section 3.

At a mid-load of 1.5 MPa IMEP, the dual-fuel operation yielded an ISsoot of 0.011 g/kWh, which was significantly higher than the 0.003 g/kWh for the CDC case. This can be explained by the late CA50 position and short ignition delay, which potentially reduced combustion temperatures and increased local fuel/air equivalence ratios.

CO and unburnt HC emissions increased significantly in the dual-fuel combustion when compared against the CDC operation. This was probably a result of premixed fuel trapped in the crevice volumes of the stock diesel piston as well as lower local in-cylinder gas temperatures [6].

High net indicated specific emissions of CO (ISCO) and unburnt HC (ISHC) were measured for the dual-fuel operation at 0.3 MPa IMEP. This can be attributed to excessively low combustion temperatures and overly lean regions that did not release enough heat in order to effectively oxidise the fuel [6]. At 1.8 and 2.4 MPa IMEP, the use of lower EFs as well as lower EGR rates likely increased combustion temperatures, decreasing CO and unburnt HC emissions.

4.5. Engine performance

Figure 9 depicts the engine performance metrics for optimised CDC and ethanol-diesel dual-fuel operation. The global fuel/air equivalence ratio (Φ) of the dual-fuel combustion was either comparable or lower than that of the CDC mode at a given engine load. This was attributed to minor variations in the intake air flow rate (within 3% and not showed for the sake of brevity) and improvements in net indicated efficiency. Differences in LHV_{fuel} probably balanced out changes in stoichiometric air/fuel ratio.

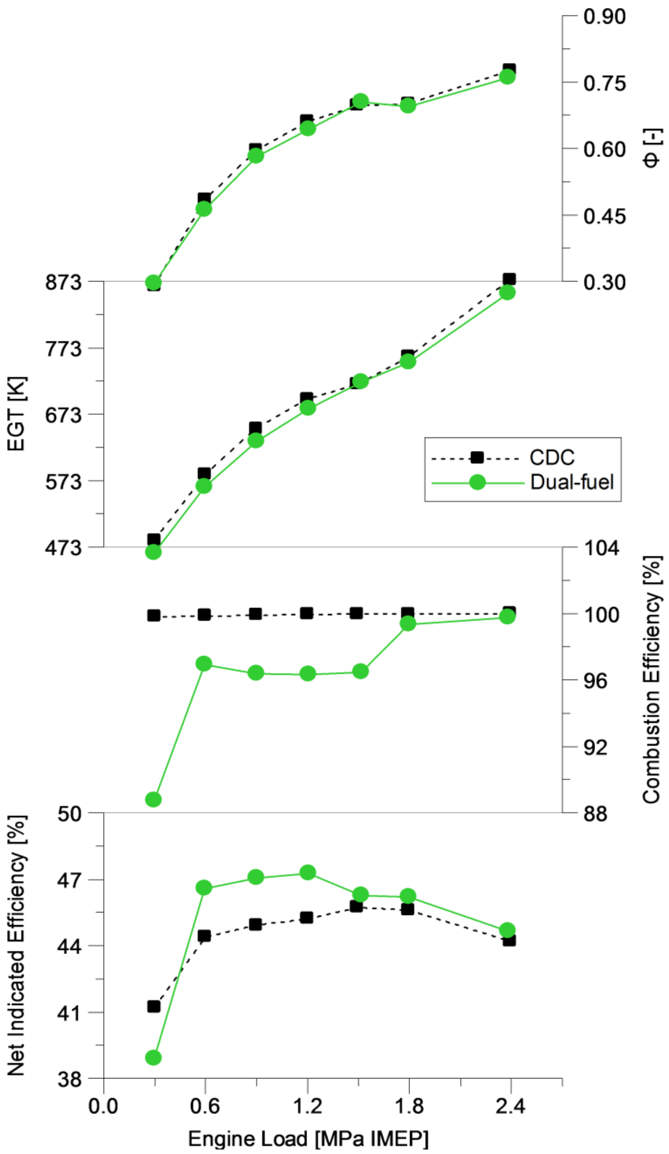

Figure 9 – Engine performance for optimised CDC and ethanol-diesel dual-fuel operation at 1200 rpm.

The exhaust gas temperature (EGT) increased with the engine load due to later CA90s and higher levels of fuel energy supplied. However, the dual-fuel operation incurred EGTs up to 20 K lower than those of the respective CDC case. This was possibly a result of a more homogenous and lower temperature combustion process for an engine operation with premixed ethanol fuel [31][46].

The dual-fuel mode also yielded lower combustion efficiencies than the CDC cases as supported by the ISCO and ISHC in Figure 8. At medium and high engine loads, combustion efficiency ranged between 96.3% and 99.7% despite the use of high EFs. This was attributed to relatively higher Φ and local in-cylinder gas temperatures.

At the lowest load of 0.3 MPa IMEP, the combination of a low combustion efficiency of 88.7% and an EGT of 463 K can represent a challenge for HD engine manufactures. This is due to a reduction in the effectiveness of the oxidation catalyst in reducing CO and unburnt HC emissions [57][58]. In-cylinder control strategies such as intake throttling and iEGR can help increase the EGT while simultaneously minimising the levels of ISCO and ISHC [31]. Moreover, the low combustion efficiency adversely affected the performance of the dual-fuel operation at 0.3 MPa IMEP, limiting the net indicated efficiency to 38.9%.

Nonetheless, the ethanol-diesel dual-fuel combustion resulted in higher net indicated efficiencies than the CDC operation between 0.6 and 2.4 MPa IMEP. A peak net indicated efficiency of 47.2% was attained at 1.2 MPa IMEP and represented an increase of 4.4% over the 45.2% of the CDC mode. The maximum net indicated efficiency achieved by the CDC operation was 45.7% at 1.5 MPa IMEP.

The ethanol autoignition process likely helped decrease the combustion temperatures and thus the heat transfer losses [49], as supported by the NOx reduction in Figure 8. However, the use of a late CA50 at 1.5 MPa IMEP and low EFs at 1.8 and 2.4 MPa IMEP limited improvements in the net indicated efficiency of the dual-fuel operation. This was necessary in order to control the PRRs below 2.0 MPa/CAD.

4.6. Additional practical considerations

Additional practical aspects for ethanol-diesel dual-fuel operation were assessed in order to evaluate whether the combustion strategy can be successfully used in a Euro VI HD engine. The analysis focused on the total fuel flow rate, the estimated consumption of aqueous urea solution in the SCR system ($\dot{m}_{urea}$) to meet the Euro VI NOx limit of 0.4 g/kWh, and the SCR corrected net indicated efficiency ($Net\ Indicated\ Eff._{SCR\ corr.}$). The methodology for the calculation of these performance metrics has been described in our previous study [45].

Figure 10 shows the optimised ethanol-diesel dual-fuel combustion increased the total fuel consumption by up to 45.8% in comparison with the CDC mode (8.12 kg/h vs. 5.57 kg/h at 1.5 MPa IMEP). This is attributed to the relatively lower density ($\rho_{ethanol}$) and energy content ($LHV_{ethanol}$) of the ethanol fuel. Appropriate volumes of diesel and ethanol fuel tanks will have to be designed according to the application of the engine and duty cycle.

In terms of NOx aftertreatment, the ethanol-diesel dual-fuel combustion attained lower levels of ISNOx than the CDC operation, effectively decreasing the $\dot{m}_{urea}$ requirements. Higher $\dot{m}_{urea}$ were estimated for both the combustion modes as the engine load was increased. This was due to an increase in the production of NOx emissions (in g/h) as well as the reduction in the EGR rate at 1.8 and 2.4 MPa IMEP.

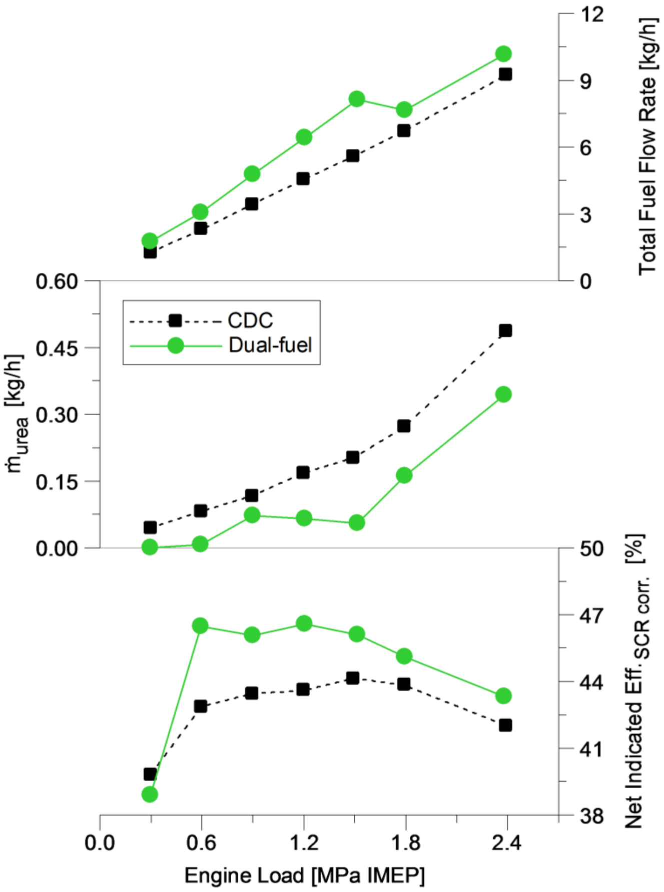

Figure 10 – Practical considerations for optimised CDC and ethanol-diesel dual-fuel operation on a Euro VI HD engine.

The lower urea consumption in the dual-fuel mode allowed for higher $Net\ Indicated\ Eff._{SCR\ corr.}$ between 0.6 and 2.4 MPa IMEP. The maximum $Net\ Indicated\ Eff._{SCR\ corr.}$ of 46.5% was achieved at 0.6 MPa IMEP and represented an increase of 8.4% over the CDC mode. Impaired combustion efficiency limited the $Net\ Indicated\ Eff._{SCR\ corr.}$.of the dual-fuel mode at 0.3 MPa IMEP, despite the low engine-out NOx of 0.4 g/kWh and $\dot{m}_{urea} = 0$.

These improvements can reduce the engine running costs depending on the volumetric price ratio between ethanol and diesel fuel [45] as well as the cost of aqueous urea solution. Nevertheless, the implementation of this dual-fuel combustion strategy on a HD engine would have to weigh the higher efficiency and lower NOx emissions against the additional complexity and upfront cost of a port fuel injection system and extra fuel tank.

4.7. Potential CO_2 reduction

Table 4 reveals the complete combustion of ethanol can reduce the emissions of CO_2 by ~4% when compared to the combustion of diesel at a given energy input. However, practical ethanol energy fractions in dual-fuel mode vary between 0.00 and ~0.80 while the actual fuel energy consumption changes with the net indicated efficiency.

Table 4 – Hypothetical CO_2 emissions for diesel and ethanol combustion.

Property	Diesel	Ethanol
Normalised molecular composition	$CH_{1.825}O_{0.0014}$	$CH_3O_{0.5}$
Lower heating value (LHV_{fuel})	42.9 MJ/kg	26.9 MJ/kg [49]
Normalised molar mass (M_{fuel})	13.87 g/mol	23.03 g/mol
Mass of CO_2 emissions per mole of fuel	44.01 gCO_2/mol	44.01 gCO_2/mol
Mass of CO_2 emissions per mass of fuel	3.17 gCO_2/g	1.91 gCO_2/g
Mass of CO_2 emissions per MJ of fuel	73.9 gCO_2/MJ	71 gCO_2/MJ
Specific CO_2 emissions reduction	n/a	~4%

The use of the engine-out CO_2 emissions in the calculation of net indicated specific emissions of CO_2 ($ISCO_2$) would result in incorrect trends for a dual-fuel operation, with significant reductions at all engine loads. This is because of the partial oxidation of hydrocarbons and formation of CO. To remove the effect of incomplete combustion, the $ISCO_2$ (in g/kWh) was estimated using the Equation 3, which assumed a complete oxidation of the fuel injected to CO_2, either in-cylinder or in the aftertreatment system.

$$ISCO_2 = \left(\frac{\dot{m}_{diesel}}{M_{diesel}} + \frac{\dot{m}_{ethanol}}{M_{ethanol}}\right)\left(\frac{M_{CO_2}}{P_{ind}}\right) \times 10^3 \qquad (3)$$

where M_{CO_2} is the molar mass of CO_2 of 44.01 g/mol [50] and P_{ind} is the net indicated power in kW.

Figure 11 shows the optimised dual-fuel operation can achieve lower $ISCO_2$ than the CDC mode from 0.6 to 2.4 MPa IMEP. The potential CO_2 reduction introduced by the ethanol-diesel dual-fuel combustion varied between 1.8% and 7.5%. This improvement was a result of the increase in net indicated efficiency combined with higher hydrogen to carbon ratio of the ethanol fuel [59,60]. The low net indicated efficiency impaired the CO_2 reduction at 0.3 MPa IMEP, increasing the $ISCO_2$ by 3.7% when compared to the CDC mode.

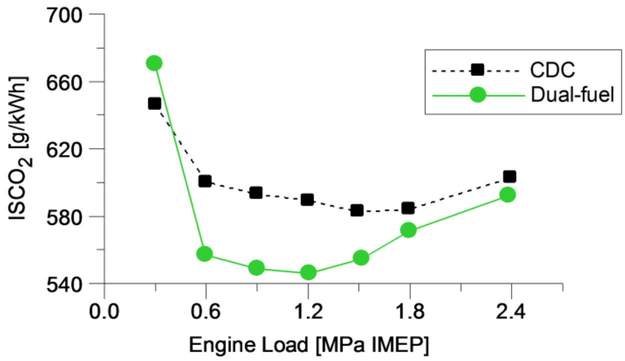

Figure 11 – Estimated $ISCO_2$ for optimised CDC and ethanol-diesel dual-fuel operation at 1200 rpm.

In order to provide an additional insight into the CO_2 reductions, a tank-to-wheels (TTW) analysis was performed by calculating the ratio of the estimated mass of CO_2 emissions to the total fuel energy supplied to the engine (in MJ) as

$$TTW\ CO_2 = \frac{ISCO_2\ P_{ind}}{(\dot{m}_{diesel}\ LHV_{diesel} + \dot{m}_{ethanol}\ LHV_{ethanol})} \tag{4}$$

Figure 12 reveals the optimised ethanol-diesel dual-fuel combustion decreased the levels of TTW CO_2 emissions by up to 3.2% when compared against a constant 73.9 g/MJ produced by the CDC operation. This was attributed to the presence of the ethanol fuel, as the TTW CO_2 emissions are heavily dependent on the in-cylinder fuel characteristics (e.g. M_{fuel} and LHV_{fuel}).

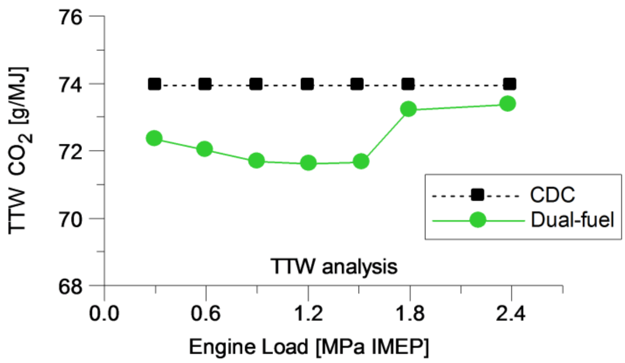

Figure 12 – Estimated TTW CO_2 emissions for CDC and ethanol-diesel dual-fuel operation.

It is important bear in mind that the data showed in Figure 11 and Figure 12 were obtained by assuming complete conversion of the fuel into $ISCO_2$. Additionally, the analysis neglected the CO_2 emissions produced by aqueous urea solution reactions in the SCR system [26], which were calculated [50] to be smaller than 0.4% of the estimated $ISCO_2$. For a more comprehensive analysis, the actual CO_2 emissions should be measured downstream of the aftertreatment system during the appropriate engine/vehicle test cycle.

4.8. Theoretical well-to-wheels analysis

A well-to-wheels (WTW) analysis can be used to assess the GHG emissions and energy expended over the production and use of a given fuel [37,44]. The methodology combines the TTW results to the well-to-tank (WTT) contribution, which takes into consideration the GHGs emitted during the extraction or cultivation of raw materials, processing, transportation, and other processes necessary to get the fuel into the fuel tank.

The levels of GHGs were expressed as grams of CO_2 equivalent (CO_{2eq}) emissions per MJ of fuel injected. This was required because of the higher global warming potentials (GWPs) for methane (CH_4) and nitrous oxide (N_2O) compounds, which have GWPs equivalent to 25 and 298 times that of the CO_2 over a time horizon of 100 years [61].

If one considers the CO_2 emissions from bioethanol combustion can be absorbed by plants during photosynthesis [37,44], the TTW CO_{2eq} emissions for a bioethanol-diesel dual-fuel engine will be determined by those emitted from diesel combustion only as

$$TTW\ CO_{2eq} = 73.9\,(1 - EF) \tag{5}$$

From Equation 5, the WTW CO_{2eq} emissions were calculated as

$$WTW\ CO_{2eq} = [WTT_{diesel}(1 - EF) + WTT_{ethanol}(EF)] + TTW\ CO_{2eq} \tag{6}$$

where WTT_{diesel} is the WTT CO_{2eq} emissions for fossil diesel fuel of 15.4 g/MJ [38][39], and $WTT_{ethanol}$ is the WTT CO_{2eq} emissions for sugarcane ethanol of 24.8 g/MJ [38][39].

The $WTT_{ethanol}$ excluded CO_{2eq} emissions produced by indirect land use change (iLUC) due to the uncertainties over the predictions [62–64][65] and the possibility of a bonus if biomass is obtained from restored degraded land [36].

Figure 13 shows the theoretical TTW CO_{2eq} and WTW CO_{2eq} emissions for CDC and bioethanol-diesel dual-fuel operation. The lowest TTW CO_{2eq} emissions were attained at mid-loads under the dual-fuel mode, where both the net indicated efficiency and EF were maximised. As a result, the bioethanol-diesel dual-fuel combustion decreased the levels of WTW CO_{2eq} by up to 57% when compared with the 89.3 g/MJ for a CDC operation. These improvements can help combat climate change and achieve a more sustainable transport sector.

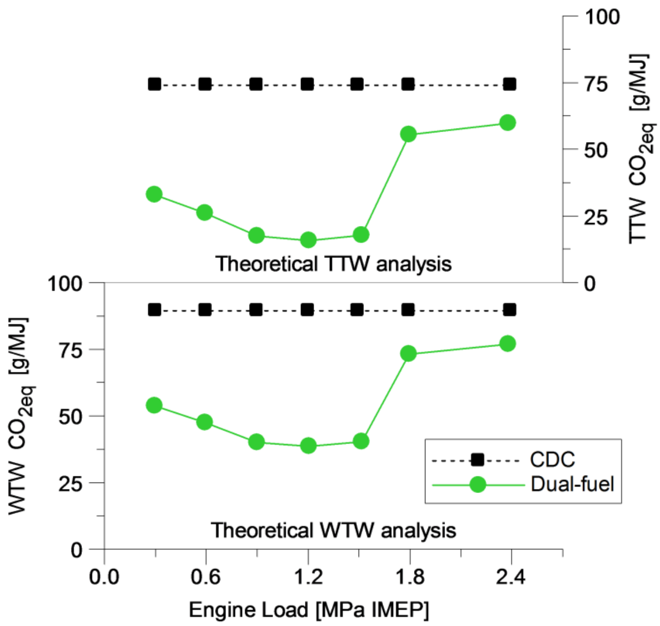

Figure 13 – Theoretical TTW and WTW CO_{2eq} emissions for CDC and bioethanol-diesel dual-fuel operation.

5. Conclusions

In this study, experiments were performed to compare the controllability, exhaust emissions, and fuel conversion efficiency of ethanol-diesel dual-fuel combustion to those of conventional diesel combustion (CDC). The investigation was conducted using identical operating conditions at a constant engine speed of 1200 rpm and different loads ranging between 0.3 and 2.4 MPa IMEP.

Testing was carried out on a HD diesel engine with a stock piston and original compression ratio of 16.8. Peak in-cylinder pressure and pressure rise rate (PRR) were limited at 18 MPa and 2.0 MPa/CAD, respectively. The diesel injection timings and fuel delivery were optimised for the maximum net indicated efficiency in both the combustion modes. The comparison against the CDC operation allowed for a better understanding of the potentials, requirements, and limitations of the ethanol-diesel dual-fuel combustion, which can be summarised as follows:

1. The dual-fuel combustion attained significantly lower NOx and soot emissions than the CDC cases at low engine loads of 0.3 and 0.6 MPa IMEP. This was attributed to the combustion of more homogeneous charge obtained by the use of an early single diesel injection at approximately -36 CAD ATDC and large percentages of premixed ethanol with energy fractions of 0.56 and 0.65.
2. The dual-fuel mode experienced a relatively low net indicated efficiency of 38.9% at the lowest load of 0.3 MPa IMEP. This was associated with the reduced combustion efficiency of 88.7% caused by excessively lean and low temperature combustion. This region of engine speed-load map also suffered from a low exhaust gas temperature of 463 K, which can adversely affect the effectiveness of the oxidation catalyst.
3. Higher ethanol energy fractions up to 0.79 and adaptive diesel injections were required as the engine load was increased from 0.6 to 1.5 MPa IMEP. A transition zone was observed between 0.6 and 0.9 MPa IMEP where less diesel fuel could be partially premixed in order to avoid early ignition and control the levels of PRR.
4. At mid-loads of 0.9, 1.2, and 1.5 MPa IMEP, optimised dual-fuel combustion was achieved with a 3 mm^3 diesel pre-injection prior to the main diesel injection. The relatively higher degree of fuel stratification increased the levels of NOx and soot when compared to those obtained at low engine loads. Nevertheless, mid-load dual-fuel operation attained lower NOx emissions and up to 4.4% higher net indicated efficiencies than the CDC cases.
5. At high engine loads of 1.8 and 2.4 MPa IMEP, early autoignition of the ethanol fuel increased the PRRs and limited the maximum ethanol energy fractions at 0.25 and 0.19, respectively. This was linked to the high in-cylinder gas temperatures and pressures prior to the start of combustion. Nonetheless, the ethanol compression ignition combustion helped increase the net indicated efficiency and reduce NOx emissions in comparison with the CDC mode. This was primary due to a shorter diesel mixing-controlled combustion.

6. The ethanol-diesel dual-fuel combustion increased the total fuel flow rate by up to 45.8% when compared against the CDC operation. This was a result of differences in fuel characteristics (e.g. LHV_{fuel}) and will require the design of appropriate fuel tank volumes.

Overall, the optimisation of the diesel injection strategy and ethanol energy fraction was a key enabler for controlling the PRRs. This allowed for a dual-fuel combustion with higher net indicated efficiencies than the CDC operation between 0.6 and 2.4 MPa IMEP, with a peak of 47.2% at 1.2 MPa IMEP. Furthermore, the ethanol-diesel dual-fuel combustion attained lower NOx emissions (up to 90%) than the CDC mode from low to full engine load. This can decrease the consumption of aqueous urea solution in the exhaust aftertreatment system and help to lower the engine running cost. Finally, the substitution of diesel with bioethanol (e.g. produced from sugarcane) can reduce the use of fossil fuel and effectively minimise the GHG emissions of future HD engines, as supported by the lower tank-to-wheels and theoretical well-to-wheels CO_2 equivalent emissions.

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Nomenclature

ATDC, After Firing Top Dead Centre; CA10, Crank Angle of 10% Cumulative Heat Release; CA10–CA90, Combustion Duration or 10–90% Cumulative Heat Release; CA50, Crank Angle of 50% Cumulative Heat Release; CA90, Crank Angle of 90% Cumulative Heat Release; CAD, Crank Angle Degree; CDC, Conventional Diesel Combustion; CH_4, Methane; CO, Carbon Monoxide; CO_2, Carbon Dioxide; CO_{2eq}, CO_2 Equivalent; COV_IMEP, Coefficient of Variation of

IMEP; COV_P_{max}, Coefficient of Variation of P_{max}; DAQ, Data Acquisition; ECU, Engine Control Unit; EF, Ethanol Energy Fraction; EGR, Exhaust Gas Recirculation; EGT, Exhaust Gas Temperature; FID, Flame Ionisation Detector; GHG, Greenhouse Gas; GWP, Global Warming Potential; HC, Hydrocarbons; HD; Heavy-duty; HRR, Apparent Net Heat Release Rate; iEGR, Internal EGR; iLUC, Indirect Land Use Change; IMEP, Net Indicated Mean Effective Pressure; ISCO, Net Indicated Specific Emissions of CO; $ISCO_2$, Net Indicated Specific Emissions of CO_2; ISHC, Net Indicated Specific Emissions of Actual Unburnt HC; ISNOx, Net Indicated Specific Emissions of NOx; ISsoot, Net Indicated Specific Emissions of Soot; LHV_{diesel}, Lower Heating Value of Diesel; $LHV_{ethanol}$, Lower Heating Value of Ethanol; LHV_{fuel}, Lower Heating Value; $\dot{m}_{air}$, Fresh Air Mass Flow Rate; $\dot{m}_{diesel}$, Diesel Mass Flow Rate; $\dot{m}_{ethanol}$, Ethanol Mass Flow Rate; $\dot{m}_{urea}$, Estimated Consumption of Aqueous Urea Solution in the SCR System; M_{CO_2}, Normalised Molar Mass of CO_2; M_{diesel}, Normalised Molar Mass of Diesel; $M_{ethanol}$, Normalised Molar Mass of Ethanol; M_{fuel}, Normalised Molar Mass; MFB, Mass Fraction Burnt; N_2O, Nitrous Oxide; $Net\ Indicated\ Eff._{SCR\ corr.}$, SCR Corrected Net Indicated Efficiency; NOx, Nitrogen Oxides; O_2, Oxygen; P_{ind}, Net Indicated Power; PFI, Port Fuel Injector; P_{max}, Peak In-cylinder Gas Pressure; PRR, Pressure Rise Rate; RON, Research Octane Number; SCR, Selective Catalyst Reduction; SOC, Start of Combustion; SOI_main, Actual Start of Main Diesel Injection; SOI_mai–SOC, Ignition Delay; SOI_pre, Actual Start of Diesel Pre-injection; TDC, Firing Top Dead Centre; TTW, Tank-to-wheels; VVA, Variable Valve Actuation; WTT, Well-to-tank; WTT_{diesel}, WTT CO_{2eq} Emissions for Fossil Diesel; $WTT_{ethanol}$, WTT CO_{2eq} Emissions for Ethanol; WTW, Well-to-wheels; $\%C_{fuel}$, Carbon Mass Content; $\%H_{fuel}$, Hydrogen Mass Content; $\%O_{fuel}$, Carbon Mass Content; γ, Ratio of Specific Heats; ρ_{fuel}, Density; ρ_{diesel}, Diesel Density; $\rho_{ethanol}$, Ethanol Density; Φ, Global Fuel/Air Equivalence Ratio.

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