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

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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 different diesel injection strategies were required to control the in-cylinder pressure rise rate and achieve highly efficient and clean dual-fuel operation. In terms of performance, dual-fuel combustion attained higher net indicated efficiency than the CDC mode from 0.6 to 2.4 MPa IMEP, with a maximum value 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 cause irreversible

Abbreviations: 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; CH4, methane; CO, carbon monoxide; CO2, carbon dioxide; CO2eq, CO_2 equivalent; COV_IMEP, coefficient of variation of IMEP; COV_Pmax, coefficient of variation of P_{max}; DAQ, data acquisition; ECR, effective compression ratio; ECU, engine control unit; EF, ethanol energy fraction; EGR, exhaust gas recirculation; EGT, exhaust gas temperature; FID, flame ionisation detector; FSN, filter smoke number; 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;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; $Net\ Indicated\ Eff_{.SCRcorr.}$, 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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climate change and negatively impact the health of living organisms across the globe [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]. This disproportionate contribution is highlighted by the fact the HD fleet represents only 11% of the world motor vehicles [4]. Substantial and continuous reduction in fossil fuel energy use and GHG emissions must be achieved in order to address the transport sector's impact on the environment.

Additional cause for concern is that CDC is prone to 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 [12–15]. Manufacturers are incorporating costly engine design elements [16–20] and aftertreatment technologies [21,22] to comply with the aforementioned emissions standards and GHG targets [12,13]. The use of improved 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 are some of the ways in which this is being achieved.

Reaching a balance between engine running costs and exhaust emissions can represent a challenge for HD engine manufacturers [16,17]. Both in-cylinder and aftertreatment measures are considered and are often linked. 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 that the strategy has the potential to resolve these issues, increasing the fuel conversion efficiency while decreasing both the NOx and soot emissions [6,29–32]. This has been attributed to simultaneous reductions in local fuel/air equivalence ratio and temperature, shorter combustion duration, and lower heat transfer losses [6,32].

Fig. 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

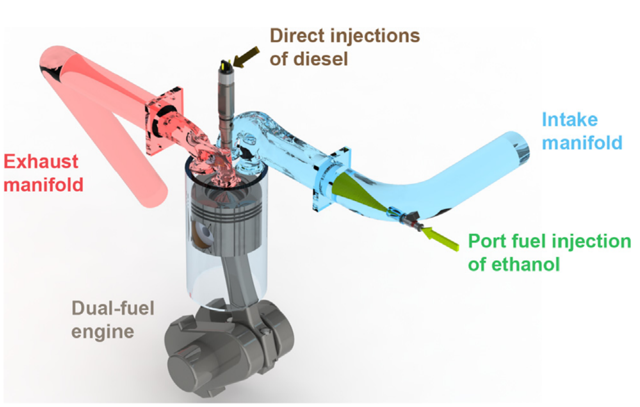

Fig. 1. Schematic diagram of a dual-fuel engine with direct injections of diesel and port fuel injection of ethanol.

triggered by direct injections of diesel [6,35]. It should be noted that the use of a low carbon fuel like ethanol [36–39] can help decrease the dependence on fossil fuels and minimise GHG emissions from the global transport sector [40].

Despite the advantages of dual-fuel operation, it is challenging to obtain direct comparisons against the CDC mode from low to high engine loads (e.g. above 2.0 MPa IMEP). This is often due to modifications in engine hardware and/or test conditions that are used to 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].

This study aims to explore the potential of ethanol–diesel 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 of 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 the above-mentioned combustion modes from low to full engine load (e.g. 0.3–2.4 MPa IMEP). In addition to this, 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. Moreover, 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 [29–31,45,46].

2. Experimental setup

2.1. Experimental facilities

A schematic diagram of the single cylinder HD engine experimental setup is shown in Fig. 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 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.

2.2. Engine specifications

Base hardware specifications are outlined in Table 1. The combustion system consisted of a 4-valve swirl-oriented cylinder head and a stepped-lip piston bowl design with a geometric compression ratio of 16.8.

The diesel introduction was controlled via a dedicated engine control unit (ECU) with the ability to support up to three injections 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