Greenhouse Gas and Noxious Emissions from Dual Fuel Diesel and Natural Gas Heavy Goods Vehicles

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Abstract

Dual fuel diesel and natural gas heavy goods vehicles (HGVs) operate on a combination of the two fuels simultaneously. By substituting diesel for natural gas, vehicle operators can benefit from reduced fuel costs and as natural gas has a lower CO\textsubscript{2} intensity compared to diesel, dual fuel HGVs have the potential to reduce greenhouse gas (GHG) emissions from the freight sector. In this study, energy consumption, greenhouse gas and noxious emissions for five after-market dual fuel configurations of two vehicle platforms are compared relative to their diesel-only baseline values over transient and steady state testing. Over a transient cycle, CO\textsubscript{2} emissions are reduced by up to 9\% however methane (CH\textsubscript{4}) emissions due to incomplete combustion lead to CO\textsubscript{2}e emissions that are 50-127\% higher than the equivalent diesel vehicle. Oxidation catalysts evaluated on the vehicles at steady state reduced CH\textsubscript{4} emissions by at most 15\% at exhaust gas temperatures representative of transient conditions. This study highlights that control of CH\textsubscript{4} emissions and improved control of in-cylinder CH\textsubscript{4} combustion are required to reduce total GHG emissions of dual fuel HGVs relative to diesel vehicles.

1 Introduction

1.1 Context
Globally, road freight transport is responsible for around a quarter of transport energy use\(^1\). In the UK, heavy goods vehicles (HGVs) involved in freight movements account for 22\% and 23\% of road transport energy use\(^2\) and greenhouse gas (GHG) emissions\(^3\) respectively. Long haul and regional duty cycles account for approximately 70\% of UK HGV CO\(_2\) emissions and the substitution of natural gas (NG) for diesel has the potential to reduce HGV CO\(_2\) emissions due to the lower CO\(_2\) intensity of methane compared to diesel\(^4\). In 2012, the UK Technology Strategy Board and the Office for Low Emission Vehicles provided £11.3m (\(~$17.5m\) to trials of 362 diesel-NG dual fuel trucks, using five different dual fuel systems. Four out of five of these systems are aftermarket conversions, while one is supplied by an original equipment manufacturer (OEM)\(^5\). In 2013, the European Commission provided €14.3m (\(~$15.6m\) to the trial of 100 LNG HGVs, including dual fuel vehicles\(^6\). In the US, the Environmental Protection Agency has certified one diesel-NG conversion system for new engines, which grants exemption from the Clean Air Act tampering prohibition, and a number of other dual fuel systems for intermediate age engines\(^7\).

The use of NG as a transport fuel is at least partially motivated by the potential to reduce CO\(_2\) emissions. The primary component of NG is methane (CH\(_4\)). As CH\(_4\) has a higher ratio of hydrogen to carbon atoms (4:1) than diesel (\(~2:1\)\), less CO\(_2\) is emitted per unit of chemical energy released by combustion. A dual fuel engine is a conventional compression ignition diesel engine in which a significant proportion of the energy released by combustion is derived from the combustion of a gaseous fuel, such as NG\(^8\). All of the dual fuel systems installed on vehicles residing the UK operate by injecting NG into the intake air prior to the intake valves so that a NG-air mixture is drawn into the combustion chamber\(^5,9–12\). This NG-air charge is then ignited by an injection of diesel at the end of the compression stroke, which has a shorter ignition delay than the NG\(^8\). This dual fuel concept differs from systems that employ high-pressure direct injection (HPDI) of NG into the combustion chamber\(^13\).

Dual fuel engines offer an attractive alternative to other engine technologies due to the (i) higher thermal efficiency relative to spark-ignited engines, (ii) flexible fuel capability with the option to operate solely on diesel when gaseous fuel is not available, (iii) reduced fuel costs, and (iv) reductions in emissions of CO\(_2\) and other engine exhaust components\(^4,14\). Previous studies of dual fuel engine emissions have focussed on engine dynamometer testing, rather than full vehicle emissions testing. These studies have shown significant changes to emissions when compared to conventional diesel engines\(^8,14–21\). Dual fuel engine emissions depend on a number of parameters including the engine speed, engine load, the composition
and quantity of primary gaseous fuel, the quantity of pilot diesel, the temperature of the
intake primary fuel and air mixture, the pilot injection timing and stratification of the gaseous
fuel and air mixture. In previous studies, dual fuel combustion with NG as the primary
fuel led to reduced emissions of oxides of nitrogen (NOx) but increased emissions of carbon
monoxide (CO) and unburnt hydrocarbons (HC). The majority of the unburnt
hydrocarbon emissions from a NG dual fuel engine is CH4 and the magnitude of these
emissions depended strongly on the concentration of the gaseous fuel in the cylinder.

Using a global warming potential (GWP) of 25 for CH4, Besch et al. showed that while CO2
emissions were reduced by 3-8% for three dual fuel engines relative to diesel operation over
the heavy duty engine Federal Test Procedure, emissions of CH4 led to 18-129% increases in
CO2e emissions.

Previous smoke opacity measurements have shown that diesel and NG dual fuel combustion
has the potential to reduce smoke emissions. Graves et al. characterised the
morphology and volatility of particles emitted by an HPDI dual fuel engine, however this is a
distinctly different dual fuel concept from that employed by the systems currently in use and
evaluated in this study.

1.2 Overview

This paper presents an evaluation of transient and steady state energy use, greenhouse gas,
nitrous oxides, carbon monoxide and particulate matter (PM) emissions of five vehicle
configurations of two dual fuel aftermarket conversion systems that account for the majority
(~60%) of the 362 trialled dual fuel vehicles in the UK. In contrast to previous studies that
have tested engines installed on engine dynamometers, this study tests in-use vehicles with
emissions control devices on a chassis dynamometer to quantify dual fuel HGV emissions
factors for subsequent use in environmental impact assessments and provide emissions
metrics that are relevant to policy makers and logistics fleet operators. Furthermore, this
paper highlights areas for technology development. Due to commercial sensitivities, the
vehicle owners and the dual fuel conversion suppliers have not been disclosed.

2 Materials and methods

2.1 Outline of study

The vehicle configurations and specifications of the original vehicle platforms are
summarised in Table 1. This study’s authors did not modify or attempt to control the
performance of the dual fuel systems or fuelling strategies. On both vehicles where methane
oxidation catalysts are present, these were installed upstream of the OEM selective catalytic reduction (SCR) unit by the dual fuel conversion suppliers. The dosing of aqueous urea solution in the SCR systems on both vehicles was not altered and neither vehicle was fitted with a diesel particulate filter. Configurations A0 (4×2 tractor) and B0 (6×2 tractor) are taken as baseline tests, while A1 and B1 are the most common diesel-NG dual fuel vehicle configurations for suppliers A and B respectively. The configuration in A2 and A4 is unique for this study and is not a standard option offered by supplier A; it was tested to evaluate the impact of an oxidation catalyst on performance and emissions. This oxidation catalyst was a prototype design that was installed immediately before these emissions tests were conducted. In contrast, the oxidation catalyst on vehicle B had been installed on the vehicle for approximately 500,000 km and is therefore representative of catalysts in operation. Transient cycle and steady state vehicle emissions testing was conducted on a 1.22 m (42 in.) single roll chassis dynamometer capable of simulating 20 tonnes of inertia (Millbrook Proving Ground Ltd, Bedford, MK45 2JQ, UK). Further experimental details including fuel properties and uncertainty analyses are shown in the Supporting Information.
Table 1: Test vehicle specifications and list of transient test parameters.

<table>
<thead>
<tr>
<th>Base vehicle</th>
<th>Ref.</th>
<th>After-treatment</th>
<th>Fuel</th>
<th>Hot/cold start</th>
</tr>
</thead>
<tbody>
<tr>
<td>EURO V 4×2 tractor</td>
<td>A0</td>
<td>SCR</td>
<td>Diesel</td>
<td>Hot</td>
</tr>
<tr>
<td>Max power: 228 kW</td>
<td>A1</td>
<td>SCR</td>
<td>Dual fuel</td>
<td>Hot</td>
</tr>
<tr>
<td>Max torque: 1275 Nm</td>
<td>A2</td>
<td>SCR &amp; Oxi cat</td>
<td>Dual fuel</td>
<td>Hot</td>
</tr>
<tr>
<td>Engine displacement: 9.2 L</td>
<td>A3</td>
<td>SCR</td>
<td>Dual fuel</td>
<td>Cold</td>
</tr>
<tr>
<td>Compression ratio: 17.4</td>
<td>A4</td>
<td>SCR &amp; Oxi cat</td>
<td>Dual fuel</td>
<td>Cold</td>
</tr>
<tr>
<td>Bore/ Stroke: 118/140 mm</td>
<td>Wheelbase: 3.6 m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>EURO V 6×2 tractor</td>
<td>B0</td>
<td>SCR &amp; Oxi cat</td>
<td>Diesel</td>
<td>Hot</td>
</tr>
<tr>
<td>Max power: 295 kW</td>
<td>B1</td>
<td>SCR &amp; Oxi cat</td>
<td>Dual fuel</td>
<td>Hot</td>
</tr>
<tr>
<td>Max torque: 2000 Nm</td>
<td>B2</td>
<td>SCR &amp; Oxi cat</td>
<td>Dual fuel</td>
<td>Cold</td>
</tr>
<tr>
<td>Engine displacement: 11.97 L</td>
<td>Compression ratio: 18.5</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bore/ Stroke: 128/155 mm</td>
<td>Wheelbase: 3.9 m</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

2.2 Test cycles

Transient cycle emissions were obtained over the vehicle version of the European Transient Cycle, developed by the FIGE Institute. The FIGE cycle has three distinct phases, representing urban (U), rural (R) and motorway (M) driving. Hot start tests followed a consistent engine warm up procedure, while cold start tests were conducted at 0°C after the vehicle had been soaked overnight. Steady state emissions testing on the chassis dynamometer was conducted to develop a map of engine emissions as a function of engine torque and speed (revolutions per minute, rpm) that would also be of further use in vehicle emissions models. Engine speed and torque were systematically varied in ~200 rpm and ~200 Nm steps respectively. Each test point was held for at least two minutes, consistent with regulatory engine test cycles.

2.3 Instrumentation

The net flow rate of diesel supplied to the engines was measured using an FMS MK10 fuel flowmeter (JPS Engineering, UK), which recorded flow in 5 mL increments. The mass flow rate of NG delivered to the engines was measured by a Rotamass RCCS34 Coriolis flowmeter (Yokogawa Electric Corporation, Japan) placed in line between the low pressure regulator and gas injectors of the dual fuel systems. This Coriolis flow meter was sized to minimise interference with dual fuel system by minimising pressure drop while maintaining accuracy; at a nominal flow rate of 22.5 kg/hour at 5 bar and 20°C, the pressure drop and accuracy were calculated to be 17 mbar and 1.1% respectively. The energy substitution ratio (ESR) is defined as the proportion of total energy supplied to the engine in the form of NG,
where $\dot{m}_{NG, \text{in}}$ and $\dot{m}_{\text{diesel, in}}$ are the mass flow rates of NG and diesel supplied to the engine respectively, and LCV is the lower calorific value of the fuel.

Engine emissions and exhaust gas temperatures were measured at two locations in the exhaust; (i) post-turbo (PT), equivalent to an engine-out measurement, and (ii) at the tailpipe. A complete list of the emissions analysers and emissions species is included in the SI. Reported emissions factors have been calculated based on the modal 1 Hz data for transient and steady state testing. For CO$_2$, NO$_x$ and CO, these emissions factors were within 1%, 1% and 7% of emissions factors calculated based on averaged bag measurements over the entire transient test cycle.

CH$_4$ emissions were measured at the post-turbo and tailpipe locations using Fourier Transform Infra-Red (FTIR) spectrometers (Multigas 2030 and 2030 HS respectively, MKS Instruments, MA, USA). CH$_4$ slip is reported as the ratio of the mass flow rate of exhausted CH$_4$, $\dot{m}_{\text{CH}_4, \text{tailpipe}}$, to the mass flow rate of CH$_4$ supplied to the engine in the NG, $\dot{m}_{\text{CH}_4, \text{in}}$.

$$\text{CH}_4 \text{ slip} \%= \frac{\dot{m}_{\text{CH}_4, \text{tailpipe}}}{\dot{m}_{\text{CH}_4, \text{in}}} \times 100\%.$$  

The effectiveness of oxidation catalysts in the vehicles’ exhaust after-treatment systems was evaluated by comparing CH$_4$ concentrations at the post-turbo and tailpipe sampling points measured by the two FTIR instruments.

A DMS500 (Cambustion Ltd, UK) measured real-time particle size distributions at the tailpipe. PM and black carbon (BC) mass emissions were estimated using the integrated particle size distribution (IPSD) method$^{30,31}$. The aggregate effective density, $\rho_{\text{eff}}$, as a function of mobility diameter, $d_p$, was calculated as

$$\rho_{\text{eff}} = \rho_0 \left( \frac{d_p}{d_{pp}} \right)^{D_m},$$  

where $\rho_0$ is the material density, $d_{pp}$ is the primary particle diameter and $D_m$ is the mass-mobility exponent with assumed values of 2 g/cm$^3$, 16 nm and 2.35 respectively$^{32}$. For $d_p$ less than 30 nm, an effective density of 1.46 g/cm$^3$ is assumed$^{32}$. Total PM mass was
calculated by integrating over the entire size distribution (5-1000 nm), while for BC, a lognormal distribution was fitted to the accumulation mode. Reported particle number (PN) emissions factors are calculated by summing the product of the particle size distribution and a detection efficiency curve fitted to data for condensation particle counters with detection efficiencies of 50% at 23 nm. This enables comparison to measurements following the solid particle number measurement protocol developed through the Particle Measurement Programme (PMP).

Total GHG emissions, as CO\textsubscript{2}e were calculated by summing CO\textsubscript{2}, CH\textsubscript{4}, N\textsubscript{2}O, NO\textsubscript{x}, CO and BC emissions multiplied by their GWPs for a time horizon of 100 years. GWPs for CH\textsubscript{4} (34) and N\textsubscript{2}O (298) are taken from the IPCC’s Fifth Assessment Report (AR5) and include climate-carbon feedbacks. All GWPs are included in the SI and we note that choosing a shorter time horizon (e.g. 20 years) would increase the importance of short-lived species (e.g. CH\textsubscript{4}).

3 Results

3.1 Transient cycle energy consumption and emissions

Distance specific energy consumption and emissions factors over different phases of the FIGE cycle (U = urban, R = rural, M = motorway, C = combined) for the different vehicle configurations are shown in Table 2 and Table 3. This data is shown graphically in the SI. Distance specific energy consumption is generally highest during the urban phase, followed by the motorway and rural phases for all vehicle configurations. In general, dual fuel operation reduces energy efficiency and results in higher total energy consumption compared to the baseline diesel only tests (A0 and B0). For A1 and B1, combined energy consumption is 12% and 10% higher than the equivalent diesel tests. Previous studies have also shown that energy efficiency of dual fuel engines are reduced compared to diesel at low and intermediate engine loads, which is attributed to lower combustion efficiency of the lean NG-air charge mixture and higher rates of heat loss during combustion. Over the four dual fuel tests with vehicle A, higher ESR during A1 and A3 compared to A2 and A4, indicate that the dual fuel system supplier may have updated their control software. Comparing A1 to A2, higher ESR is correlated with higher energy consumption yet lower CO\textsubscript{2} emissions. For vehicles A and B, the highest ESR is observed during the motorway phase and during this phase, the CO\textsubscript{2} emissions factor is lowest and the relative reduction compared to diesel is greatest at up to 11% and 13% for A1 and B1 respectively. For context, the fuel properties (shown in the SI)
indicate that complete combustion of NG would emit approximately 21% less CO\textsubscript{2} per unit of energy than diesel used in this study, assuming no change in energy efficiency. For the cold start tests, (A3, A4 and B2), distance specific energy consumption and emissions of CO\textsubscript{2} are generally greater than the equivalent hot start test. Both dual fuel conversion suppliers account for engine temperature in their control of the ESR; during the urban phase, when engine temperatures are cold, the ESR is significantly reduced relative to the same phase in hot start tests.

Table 2: Energy consumption, energy substitution ratio, CO\textsubscript{2} and CO\textsubscript{2e} emissions factors from transient testing. U = urban, R = rural, M = motorway, C = combined phases of the FIGE cycle.

<table>
<thead>
<tr>
<th></th>
<th>Energy (MJ/km)</th>
<th>ESR (%)</th>
<th>CO\textsubscript{2} (g/km)</th>
<th>CO\textsubscript{2e} (g/km)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>U</td>
<td>R</td>
<td>M</td>
<td>C</td>
</tr>
<tr>
<td>A0</td>
<td>12.8</td>
<td>10.2</td>
<td>10.5</td>
<td>10.7</td>
</tr>
<tr>
<td>A1</td>
<td>13.7</td>
<td>11.1</td>
<td>12.3</td>
<td>12.0</td>
</tr>
<tr>
<td>A2</td>
<td>12.5</td>
<td>10.4</td>
<td>11.6</td>
<td>11.3</td>
</tr>
<tr>
<td>A3</td>
<td>16.7</td>
<td>11.5</td>
<td>12.3</td>
<td>12.6</td>
</tr>
<tr>
<td>A4</td>
<td>13.1</td>
<td>12.1</td>
<td>12.6</td>
<td>12.5</td>
</tr>
<tr>
<td>B0</td>
<td>14.6</td>
<td>10.1</td>
<td>11.0</td>
<td>11.2</td>
</tr>
<tr>
<td>B1</td>
<td>20.4</td>
<td>9.9</td>
<td>11.9</td>
<td>12.2</td>
</tr>
<tr>
<td>B2</td>
<td>15.7</td>
<td>13.3</td>
<td>14.3</td>
<td>14.1</td>
</tr>
</tbody>
</table>

CO\textsubscript{2} and CH\textsubscript{4} (shown in Table 3) dominated total CO\textsubscript{2e} emissions and capture the first-order climate impacts of dual fuel relative to diesel. The combined cycle relative contribution of CH\textsubscript{4} normalised to the CO\textsubscript{2} contribution was between 40% for A3 and 150% for B2. CH\textsubscript{4} emissions factors are typically highest during the motorway phase, which is also the phase with highest ESR. For all dual fuel tests on both vehicles, CH\textsubscript{4} emissions led to increased CO\textsubscript{2e} emissions relative to the diesel baseline tests. Comparing A1 to A0 and B1 to B0, dual fuel operation increased CO\textsubscript{2e} emissions factors by 32%, 24%, 42% and 123%, 86%, 160% for the urban, rural and motorway phases respectively. Therefore, it is clear that CH\textsubscript{4} emissions outweigh potential reductions in CO\textsubscript{2} emissions that result from substituting diesel for NG. For A1, 8.4%, 6.5% and 8.1%, and for B1, 30.1%, 18.4% and 21.0% of CH\textsubscript{4} that is delivered to the engine is emitted to the atmosphere during the urban, rural and motorway phases respectively. These findings are in agreement with previous engine dynamometer testing of dual fuel retrofit systems, which also measured significant increases in CO\textsubscript{2e} emissions due to CH\textsubscript{4} emissions\textsuperscript{18}. The addition of an oxidation catalyst to A2 compared to A1 did not significantly reduce CH\textsubscript{4} emissions; indeed distance specific CH\textsubscript{4} emissions are...
increased, which could be indicative of an altered fuelling strategy or that backpressure due to the oxidation catalyst may have had an adverse effect on in-cylinder combustion and therefore engine emissions performance. Oxidation catalyst effectiveness is discussed further in Section 3.3.

**Table 3:** NO$_x$, CH$_4$, PM and PN emissions from transient testing. U = urban, R = rural, M = motorway, C = combined phases of the FIGE cycle.

<table>
<thead>
<tr>
<th></th>
<th>NO$_x$ (g/km)</th>
<th>CH$_4$ (g/km)</th>
<th>PM mass (mg/km)</th>
<th>PN ($\times 10^{14}$ #/km)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ref. U R M C</td>
<td>U R M C</td>
<td>U R M C</td>
<td>U R M C</td>
</tr>
<tr>
<td>A0</td>
<td>7.8 4.4 4.0 4.6</td>
<td>0.0 0.0 0.0 0.0</td>
<td>54.5 26.8 19.5 26.8</td>
<td>1.66 0.84 0.72 0.89</td>
</tr>
<tr>
<td>A1</td>
<td>7.4 3.6 1.9 3.3</td>
<td>8.5 6.5 11.4 9.2</td>
<td>77.8 48.3 13.8 35.1</td>
<td>1.89 1.10 0.43 0.87</td>
</tr>
<tr>
<td>A2</td>
<td>10.2 4.4 2.0 4.0</td>
<td>11.5 8.3 16.5 12.8</td>
<td>89.0 54.5 24.6 44.2</td>
<td>2.31 1.47 0.79 1.24</td>
</tr>
<tr>
<td>A3</td>
<td>12.3 4.4 2.1 4.2</td>
<td>3.9 6.6 11.8 8.8</td>
<td>29.4 36.3 12.7 23.7</td>
<td>0.61 0.85 0.37 0.58</td>
</tr>
<tr>
<td>A4</td>
<td>14.3 5.3 2.6 4.8</td>
<td>2.3 10.5 19.8 14.4</td>
<td>76.2 33.6 13.8 27.7</td>
<td>2.12 0.93 0.51 0.83</td>
</tr>
<tr>
<td>B0</td>
<td>7.3 1.0 0.3 1.5</td>
<td>0.0 0.0 0.0 0.0</td>
<td>28.7 10.7 6.9 11.2</td>
<td>0.99 0.28 0.22 0.34</td>
</tr>
<tr>
<td>B1</td>
<td>2.4 0.9 2.3 1.8</td>
<td>38.9 18.9 35.6 29.7</td>
<td>40.4 13.9 8.5 14.7</td>
<td>1.38 0.31 0.25 0.42</td>
</tr>
<tr>
<td>B2</td>
<td>19.7 2.8 2.6 4.4</td>
<td>0.1 24.7 45.8 33.0</td>
<td>13.0 6.3 5.1 6.3</td>
<td>0.38 0.14 0.15 0.17</td>
</tr>
</tbody>
</table>

The combined cycle relative contributions of all species to CO$_2$ emissions are shown in the SI. For the dual fuel tests on vehicle B, the next largest contributor was N$_2$O emissions with 10% and 16% of the CO$_2$ contribution for tests B1 and B2 respectively. Measured N$_2$O emissions for these tests were significantly higher than for B0 or for any of the tests with vehicle A Tailpipe NO$_x$ emissions during the motorway phase of B1 were significantly increased compared to B0. This is despite post-turbo NO$_x$ emissions being approximately halved in B1 compared to B0 over the same phase (shown in the SI). For vehicle A dual fuel tests, post-turbo NO$_x$ emissions were reduced by ~40% and tailpipe NO$_x$ by ~50% during the motorway phase compared to the baseline A0. For A1, post-turbo and tailpipe NO$_x$ emissions were reduced by 30% compared to diesel over the combined cycle. Post-turbo NO$_2$/NO$_x$ ratios were also different for dual fuel compared to diesel; for A1 this ratio was 0.34 over the combined cycle compared to 0.06 for A0 (shown in the SI). After the SCR, NO$_2$/NO$_x$ ratios decreased to 0.15 for A1 and increased to 0.22 for A0. Indeed at the tailpipe, dual fuel NO$_2$/NO$_x$ ratios are generally lower than for the diesel tests for both vehicles A and B.

Performance of the SCR systems over the combined cycle reduced from 43% to 34-41% comparing A0 to A1 and A2, and 83% to 67% comparing B0 to B1. These results indicate that dual fuel operation affects the efficacy of the OEM SCR systems.

Particle mass and particle number emissions in warm start dual fuel tests were higher than the baseline diesel tests. The distance averaged particle size distributions (i.e. weighted by...
exhaust volumetric flow and divided by cycle distance) are shown in Figure 1. Bimodal
distributions are present for all tests. Comparing A1 to A0, the PM mass emissions factor is
increased by ~30% due to a shift in the accumulation mode peak diameter from ~65 nm to
~85 nm with total particle number emissions approximately equivalent. The peak of the
accumulation mode for vehicle B is not shifted significantly between B0 and B1, however the
PN emissions are increased by ~20% leading to a ~30% increase in total PM mass. Cold start
tests led to a more significant nucleation mode peaks and reduced accumulation mode peaks
relative to equivalent hot start tests. The contribution of BC mass to CO₂ emissions as a
percentage of the CO₂ contribution is between 1.8-4.3% for vehicle A and 0.5-1.3% for
vehicle B relative to CO₂. BC mass contributed between 60-80% of total PM mass, across all
transient tests, as shown in the SI along with uncertainties on the particle size distributions.
Steady state PM mass emissions for vehicle A are discussed further in the following section.

Engine power specific emissions factors for vehicle A over the combined FIGE/ETC cycle
are shown in the SI and compared to EURO V regulatory values for engine dynamometer
tests over the equivalent cycle. The baseline A0 configuration is compliant with CO and
PM mass standards, however NOx emissions are ~2.5 times higher than the standard (2.0
g/kWh). Indeed, NOx emissions were higher than the EURO V standard for all vehicle A
configurations. Dual fuel operation led to increased CO emissions for A1, however the
oxidation catalyst on configuration A2 meant CO emissions were equal to those of A0. CH₄
emissions, which are only regulated for NG engines at type-approval, were 8.9 times higher
than the regulatory limit value (1.1 g/kWh) for the A1 configuration of this aftermarket dual
fuel system.
Figure 1: Distance specific particle size distributions over the combined FIGE cycle.

3.2 Steady state emissions

Steady state maps of ESR and CH$_4$ slip for A1, and the ratio of dual fuel to diesel (A1:A0) CO$_2$ and CO$_2$e emissions as a function of engine speed and torque are shown in Figure 2. For A1, the ESR reaches ~70% for engine speeds between 1200 and 1600 rpm and relatively low engine torque around ~300 Nm. At higher engine torque output and engine speeds, the ESR is reduced (as controlled by the system supplier’s own proprietary software); at 600 Nm, the ESR is between 50-60% for the 1200-1600 rpm range. This can be cross referenced to the average ESR over the motorway phase of the transient cycle measured to be 58% for A1, during which the average engine torque and speed were approximately 600 Nm and 1400 rpm respectively. CH$_4$ slip is greatest at higher engine speeds, indicating that incomplete combustion of CH$_4$ is most significant when the in-cylinder residence time is lowest. The ratio of CO$_2$ emissions for A1 versus A0 indicates that the greatest reduction in CO$_2$ emissions is around 15% and that this occurs for engine speeds between 1000-1600 rpm and engine torque between 300-500 Nm. This corresponds to areas of high ESR (50-70%) and lower engine speeds. However, as shown for the transient cycle emissions, CH$_4$ emissions for A1 lead to higher total CO$_2$e emissions over almost the entire map compared to A0; this ratio is highest at low engine torques and higher engine speeds.
Figure 2: (a) Energy substitution ratio (ESR) of A1, (b) CH$_4$ slip of A1, (c) ratio of CO$_2$ emissions for A1:A0, and (d) ratio of CO$_2$e emissions for A1:A0 all as a function of engine speed and torque as measured during steady state testing of vehicle A.
Further evidence of incomplete combustion of \( \text{CH}_4 \) at higher engine speeds is provided in Figure 3, which shows the dual fuel to diesel (A1:A0) ratio of post-turbo NO\(_x\) and CO emissions and exhaust temperature. Dual fuel NO\(_x\) emissions are lower than diesel over most of the map, however the greatest reductions are observed for engine speeds greater than \(~1800\) rpm indicating lower average in-cylinder temperatures. This is further supported by the map of post-turbo exhaust temperatures which are generally lower for A1 than A0 for engine speeds greater than \(~1800\) rpm and engine torque greater than \(~400\) Nm. Post-turbo CO emissions, a product of incomplete combustion, are greater across the entire map for A1 compared to A0, and at high engine speeds they are increased by an order of magnitude.

Steady state PM mass emissions are shown in Figure 3 as a ratio of those measured for configurations A1 and A0. In the previous section, we showed that PM mass emissions increased for the dual fuel transient cycle tests compared to diesel. In Figure 3, it is evident that PM mass emissions are up to 50% lower for A1 compared to A0 at engine loads below 600 Nm and engine speeds below 2000 rpm. However at engine loads greater than 600 Nm and engine speeds less than 1500 rpm, PM mass emissions are increased for A1 compared to A0 by up to a factor of 3.
Figure 3: Ratios of (a) tailpipe PM mass emissions, (b) post-turbo (PT) NO\textsubscript{X} emissions, (c) PT CO emissions, and (d) PT exhaust temperature as a function of engine speed and torque measured during steady state testing of vehicles A1 and A0.
3.3 Oxidation catalyst effectiveness

The effectiveness of the oxidation catalysts fitted to vehicles A2 and B1 during steady state testing is shown in Figure 4. The catalysed oxidation of CH$_4$ is a function of both residence time within the catalyst and exhaust gas temperature$^{37}$. The measured catalyst effectiveness was greatest at high temperatures and residence times, i.e. low exhaust flow rates. For A2, the highest observed catalyst efficiency was 27% for a post-turbo exhaust temperature of 470°C and exhaust volumetric flow rate of 0.07 m$^3$/s. For B1, the highest observed catalyst efficiency was 30% for a post-turbo exhaust temperature of 505°C and exhaust volumetric flow rate of 0.10 m$^3$/s. For context, the average post-turbo exhaust temperatures and exhaust gas flow rates during the motorway phase of the transient cycle test were 396°C and 0.14 m$^3$/s for A2 and 370°C and 0.16 m$^3$/s for B1. These flow rates correspond to gas hourly space velocities of approximately 190,000 h$^{-1}$ and 76,000 h$^{-1}$ for the oxidation catalysts on A and B respectively. Below 400°C, the maximum performance of these catalysts were 15% and 10% for A2 and B1 respectively and therefore these results support the transient emissions results presented above that highlighted the significant contribution of tailpipe CH$_4$ to CO$_2$e emissions.

A benefit of the oxidation catalysts is to oxidise increased CO emissions during dual fuel operation to CO$_2$. Post-turbo and tailpipe CO emissions factors are shown in the SI. Even without the CH$_4$ oxidation catalyst, CO emissions are reduced by 72%, 81% and 76% by the SCR system for vehicles A0, A1 and A3 respectively. However, with the CH$_4$ oxidation catalyst, CO emissions are reduced by 94% and 92% for A2 and A4 respectively. Similarly, CO emissions were reduced by 95%, 93% and 97% for tests B0, B1 and B2 respectively.
This study evaluated the emissions performance of two vehicle platforms with five aftermarket dual fuel system configurations via chassis dynamometer testing of in-use vehicles that are part of trials of low-carbon trucks in the UK. All dual fuel systems evaluated in this study increased tailpipe total GHG (CO$_2$e) emissions compared to their equivalent diesel vehicles by 50% and 127% over the combined FIGE cycle for configurations A1 and B1 respectively. This is despite CO$_2$ emissions being reduced by up to ~9% and is primarily a result of incomplete combustion of CH$_4$ and subsequent CH$_4$ emissions, termed CH$_4$ slip. The three main mechanisms of CH$_4$ slip are (i) valve overlap which causes a proportion of the NG-air charge to be directly exhausted, (ii) incomplete combustion due to crevices and flame...
quenching at the walls of the cylinder, and (iii) incomplete combustion due to lean NG-air
mixtures and in-cylinder temperatures which prevent the flame from propagating throughout
the charge.\textsuperscript{21,38} Evidence for incomplete combustion and lower in-cylinder temperatures was
provided by post-turbo emissions measurements showing higher CO and lower NO\textsubscript{x}
respectively at steady state conditions, especially at higher engine speeds when in-cylinder
residence times are reduced. The results of this in-use vehicle emissions study are consistent
with and supplement a large number of engine emissions studies showing that NO\textsubscript{x} emissions
decrease and CO emissions increase with dual fuel combustion relative to diesel. Thus,
refinement of dual fuel systems to reduce CH\textsubscript{4} slip by addressing these three mechanisms and
to improve CH\textsubscript{4} combustion efficiency are required. Our results suggests that reducing ESR
at high engine speeds could have a beneficial effect on total CO\textsubscript{2e} emissions and the
magnitude of this effect over a transient drive cycle should be the topic of further modelling
or experimental studies. Another example is to increase the diesel pilot quantity and advance
the pilot injection timing\textsuperscript{23}, however trade-offs are increased NO\textsubscript{x} emissions and lower ESR
and therefore a reduced CO\textsubscript{2} benefit compared to diesel combustion. Direct CH\textsubscript{4} injection and
stratification of the NG within the cylinder may also have the potential to reduce dual fuel
CH\textsubscript{4} emissions.

CH\textsubscript{4} slip can also be controlled by exhaust after-treatment, however the oxidation catalysts
tested in this study reduced CH\textsubscript{4} emissions by at most 15\% at exhaust gas temperatures
representative of transient conditions (~400°C). Thus, the commercial development of
effective catalysts (e.g. Cargnello et al.\textsuperscript{39}) that successfully oxidise CH\textsubscript{4} below 400°C are
critical to the exploitation of NG as a transport fuel to reduce GHG emissions.

Both dual fuel platforms had higher PM mass emissions than their equivalent diesel
configurations in transient testing. Steady state testing revealed that PM mass emissions were
up to three times higher for dual fuel compared to diesel at high engine loads. While the
majority of previous studies report decreases in PM mass emissions for dual fuel compared to
diesel combustion\textsuperscript{21}, Papagiannakis et al.\textsuperscript{40} reported an increase in soot opacity at high engine
loads (80\%) and ESR in the range of 30-50\%. It is possible that this phenomena occurs due to
the greater likelihood of rich combustion as a result of high diesel and NG flow rates during
high engine load conditions. PM mass emissions were up to 50\% lower at low engine loads,
which is more consistent with existing literature and is attributable to the lower sooting
tendency of premixed combustion of CH\textsubscript{4} compared to diffusion mode combustion of
diesel\textsuperscript{21}. 

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Results for tailpipe NO\textsubscript{x} emissions suggest that the change in exhaust gas composition in dual fuel operation could interfere with the efficacy of the OEM SCR system, which is optimised for diesel operation. Post-turbo and tailpipe exhaust gas temperatures (shown in the SI) are generally higher in the transient dual fuel tests compared to diesel, which may result from heat release via combustion of unburned hydrocarbons over the oxidation catalyst. At higher exhaust gas temperatures and higher NO\textsubscript{2}/NO\textsubscript{x} ratios the conditions in the SCR may be significantly different when the trucks are in dual fuel mode compared to diesel, thus affecting its performance\textsuperscript{41}. Higher N\textsubscript{2}O emissions for the dual fuel tests compared to diesel tests could be due to (i) higher rates of oxidation of ammonia to N\textsubscript{2}O by NO\textsubscript{2} potentially caused by higher NO\textsubscript{2}/NO\textsubscript{x} ratios at temperatures up to 350°C\textsuperscript{42}, (ii) direct oxidation of ammonia by oxygen to N\textsubscript{2}O at temperatures above 350°C\textsuperscript{43} or (iii) by thermal decomposition of ammonium nitrate\textsuperscript{44,45}.

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6 Supporting information

Additional details describing experimental methods, supplemental results and an uncertainty analysis are included in the Supporting Information (SI). This information is available free of charge via the Internet at http://pubs.acs.org.

7 Nomenclature

\begin{align*}
\rho_0 & \quad \text{Material density (g/cm}^3) \\
\rho_{\text{eff}} & \quad \text{Effective density (g/cm}^3) \\
\text{AR5} & \quad \text{IPCC’s Fourth Assessment Report} \\
\text{AR5} & \quad \text{IPCC’s Fifth Assessment Report} \\
\text{BC} & \quad \text{Black carbon}
\end{align*}
C Combined phases of FIGE drive-cycle

CO₂e Carbon dioxide equivalent

$D_m$ Mass-mobility exponent

$d_p$ Particle mobility diameter (nm)

$d_{pp}$ Primary particle diameter (nm)

ESR Energy substitution ratio (%)

ETC European transient cycle developed by the FIGE institute

FTIR Fourier transform infrared spectroscopy

GHG Greenhouse gas

GWP Global warming potential

HC Unburned hydrocarbons

HGV Heavy goods vehicle

HPDI High-pressure direct injection

IPSD Integrated particle size distribution

LCV Lower (net) calorific value

LNG Liquefied natural gas

M Motorway (highway) phase of FIGE drive-cycle

$m_{CH_4,in}$ Mass flow rate of CH₄ supplied to the engine (kg/s)

$m_{CH_4,tailpipe}$ Mass flow rate of CH₄ exhausted at the tailpipe (kg/s)

$m_{diesel,in}$ Mass flow rate of diesel supplied to the engine (kg/s)

$m_{NG,in}$ Mass flow rate of natural gas supplied to the engine (kg/s)
NG  Natural gas
OEM  Original equipment manufacturer
PM   Particulate matter
PMP  Particle Measurement Programme
PN   Particle number
PT   Post-turbo (emissions sampling point)
U    Urban phase of FIGE drive-cycle
R    Rural phase of FIGE drive-cycle
rpm  Revolutions per minute
SCR  Selective catalytic reduction
T    Temperature (°C)

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