Experimental study on a small diesel genset dual fuelled with methane

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Abstract

A dual fuel engine is an internal combustion engine where the primary gaseous fuel source is ignited by a small quantity of diesel known as the ‘pilot’ that is injected towards the end of the compression stroke. The motivation to dual-fuel a CI engine is partly economic due to the lower cost of the primary fuel, and partly environmental as some emissions characteristics are improved. In this study, a direct injection four cylinder CI engine, typically used in engine-generator set or genset applications, was fuelled with methane. The performance and emissions (NOx and smoke) characteristics of various gaseous concentrations were recorded at 1500rpm (synchronous speed) and at no load, 1/4, 1/2, 3/4 and full load. In order to investigate the combustion performance under these different conditions, a three zone heat release rate analysis was applied to the data. The resulting mass burned rate, ignition delay and combustion duration are used to explain the emissions and performance characteristics of the engine. The results of the study showed that the combined effect of dual fuelling a DI engine with methane reduces both NOx and smoke emissions. This technology provided a beneficial method to manipulate the classic diesel engine NOx-smoke tradeoff.

Keywords: Dual Fuel, Genset, Methane, Diesel Engine

1. INTRODUCTION

The term “dual fuel” refers to a Compression Ignition (CI) engine where the primary fuel is a homogenous mixture of gaseous fuel and air, which is then ignited by a small quantity of diesel fuel – the pilot. The objective of this technique is to reduce problematic diesel engine emissions, particularly of NOx and smoke, but the drawback is that this reduction is often accompanied by an increase in emissions of CO and unburned hydrocarbons (UHC) [1]. In light of proposed emissions legislation for non road diesel engines, dual fuel presents a method of achieving these reductions without the need for traps or reduction catalysts, provided that the combustion process can be optimized. A second benefit to using a gaseous fuel in a diesel engine is economic as the gaseous fuel can cost much less than the liquid fuel it replaces.

Indirect Injection (IDI) dual fuel combustion process can be described as proceeding in three stages after ignition [2]. The first stage is due to the combustion of around half of the pilot fuel and a small amount of gaseous fuel entrained within it. The second stage is due to diffusive combustion of the rest of the pilot and the rapid burning of gaseous fuel in the immediate surroundings. The third stage is due to flame propagation through the remainder of the gaseous fuel-air charge. This description of the combustion processes allows some explanation of the mechanisms of formation of dual fuel exhaust emissions. For example, oxides of nitrogen (NOx) formation is known to be strongly dependent on local temperatures and so most NOx would be formed in the region around the pilot spray where high temperatures exist and the equivalence ratio is close to stoichiometric [3].

Dual fuel engines typically use either natural gas/methane or LPG/propane as the primary fuel [4]. Methane, the main constituent of natural gas (typically 94% by volume in the UK), is preferred for use in dual fuel engines as it is highly knock resistant and contains almost as much energy as other fuels, whilst fuel cost savings in using natural gas offset the cost of engine conversion. It is the simplest and most stable hydrocarbon and its gaseous nature allows the use of simple control systems leading to excellent mixing with air to provide more even charge distribution and smoother heat release rates [1]. Methane has a wide flammability range, low global toxicity (as compared to diesel) and has low photochemical reactivity. Most of the UHC emissions in natural gas fuelled engines are methane, and although it is chemically resistant and toxicologically inert, it is does have 12 to 30 times the greenhouse effect of carbon dioxide and so requires control.

The present study focuses on the effect of concentration of gaseous fuel, and quantity of diesel pilot on engine performance and emissions. In order to make direct comparisons, the operating conditions, such as injection timing, intake air pre-heat etc were not optimised.

2. THREE_ZONE MODEL

In order to optimize engine performance, an understanding of the processes occurring inside the combustion chamber is essential. Heat release analysis of in-cylinder pressure data is possibly the most widely used combustion diagnostic tool, and reveals information regarding the rate processes and combustion characteristics occurring inside the engine. In itself, heat
Heat release analysis is used here to investigate the dual fuel combustion process. The present contribution consists of three control volumes, as this is conceptually close to dual fuel combustion where diesel is injected into an unburned zone, (made of air and a gaseous fuel) and eventually a burned zone is formed. This approach was also chosen because it allows a model for fuel injection derived from actual operating conditions to be used. The assumptions made are;

- The combustion chamber consists of a diesel fuel zone, and unburned and a burned zone.
- The diesel zone refers to the diesel pilot only. Upon injection, the diesel fuel is assumed to instantly vaporize and obey the ideal gas law.
- The unburned zone into which fuel is injected is assumed to consist of air, exhaust gas residuals and a gaseous fuel (if present), in their measured proportions.
- The burned zone appears when combustion begins. Start of combustion is first determined from the point at which the first derivative of pressure with respect to time reaches a minimum value, and then confirmed by the second derivative of pressure being zero and the third being positive.
- Combustion is assumed to occur due to the entrainment of fuel and unburned gasses in stoichiometric proportion to air.
- Thermodynamic properties are assumed to vary in time, but not space. Individual species of the burned, unburned and vaporized fuel can be modelled as ideal gasses. Each zone has uniform temperature composition.
- Pressure is uniform across the combustion chamber.

The total mass in the combustion chamber consists of the mass of the trapped air, which is air and residual exhaust gasses, and in the dual fuel case, a gaseous fuel. The charge air and gaseous fuel proportions can be estimated from measured mass flow rates, and the residual gas fraction is assigned an arbitrary value (as the gas exchange process is not simulated). Residual gasses were assumed to have the composition. After the start of fuel injection, the mass of the cylinder also includes the mass of the fuel injected. Therefore the conservation of mass in the cylinder at any instant can be expressed as,

$$ m = m_u + m_b + m_f $$

where subscripts f, u and b denote the diesel fuel zone, the unburned zone and the burned zone, respectively.

The rate at which the fuel flows from the fuel zone to the burned zone can be determined by the difference between the mass of fuel injected at any instant and the current mass in the fuel zone. For the dual fuel case, there is the added complexity that the mass of the burned zone will also be a function of the mass of gaseous fuel that has been burned during each time step. In order to express this, it is assumed that combustion occurs at a stoichiometric air fuel ratio (AFRs). The AFRs have two hydrocarbon fuel components with molecular formulas of $C_{d}H_{d}$ and $C_{g}H_{g}$ where $d$ and $g$ denote diesel and gaseous fuel respectively; and the mass ratio of the two fuels is also known. The dual fuel stoichiometric fuel to air ratio is calculated as,

$$ AFR = \frac{\alpha(x_d + \frac{y_d}{4}) + \beta(x_g + \frac{y_g}{4})}{\alpha(x_dM_c + \frac{y_d}{4}M_H) + \beta(x_gM_c + \frac{y_g}{4}M_H)} $$

where $\alpha = \frac{m_d}{m_d + m_g}$ and $\beta = \frac{m_g}{m_d + m_g}$, $M$ is the molecule mass.

Conservation of mass, ideal gas law and first law of thermodynamics are applied to each zone so that at any instant, there are twelve unknowns to be solved; masses $m_u, m_f, m_b$, volumes $(V_u, V_f, V_b)$, temperatures $(T_u, T_f, T_b)$, and internal energies $(u_u, u_f, u_b)$ of each zones. However, the system can be reduced to two ordinary differential equations and three algebraic equations with five unknowns. The differential equations are,

$$ \frac{dT_u}{d\theta} = \frac{dp}{d\theta} \frac{R_u T_u}{m_u c_p} + \frac{dQ_u}{d\theta} $$

$$ \frac{dT_f}{d\theta} = \frac{dp}{d\theta} \frac{R_f T_f}{m_f c_p} + \frac{dQ_f}{d\theta} $$

The unknowns are $dT_u$ and $dT_f$ are solved by 4th Order Runge-Kutta method, and $m_u, m_f, T_b$ are found from three algebraic equations that are solved by Newton-Rhapson technique:

$$ f_1(m_u, m_f, T_b) = m_u + m_j + (m_{fj} - m_f)(1 + AFR_{j, tot}) $$

$$ f_2(m_u, m_f, T_b) = m_u R_u T_u + $$

$$ (m_{gj} - m_g)(1 + AFR_{g, tot}) $$

$$ f_3(m_u, m_f, T_b) = (m_{u} + m_{u} + m_{g} + R_{u} u_{b}) $$

where the rate at which the fuel flows from the fuel zone to the burned zone can be calculated by the difference between the mass of fuel injected at any instant $(m_{fj})$ and the current mass in the fuel zone $(m_f)$.

The main inputs to the model are a record of the cylinder pressure against crank angle, and data for diesel
fuel mass flow rate, needle lift, and fuel line pressure to determine the mass flow rate and injection velocity of the fuel. Other inputs required are the inlet temperatures and mass flow rates of the gaseous fuel and air, from which initial conditions at inlet valve closure and the mass fractions of gaseous fuel and air can be calculated.

The engine speed is also required. The mixture properties in the unburned zone are calculated from inlet valve closure to start of injection, using the initial conditions, pressure data and the ideal gas law. At start of injection the fuel zone comes into existence, and during the short ignition delay period, the reference conditions are added to the fuel and unburned zone, increasing the values of temperature, heat transfer (\(dQ_f\) and \(dQ_t\)) and internal energy.

The burned zone then appears at the start of combustion, and equations 5 to 9 are solved. In the case where a small pilot quantity is injected, the pilot may be completely consumed before combustion of the gaseous fuel is complete. A constraint is added here that if the mass of diesel becomes zero, the gaseous fuel will continue to burn in dual fuel mode, whereas in diesel mode, the combustion is complete and the gasses are expanded from that point. In this way, the turbulent flame propagation through the gaseous fuel zone can be implicitly included in the heat release analysis, if it is present. In order to implement this method, a record of the burned zone composition is preserved and used to calculate the new thermodynamic properties.

Selected key data for the fuels are presented in table 1.

![Figure 3: Schematic description of three zone model](Image)

Table 1: Selected Properties of the gaseous fuels, Properties of diesel from ESSO Ultra Low Sulphur Diesel from Esso Marketing Technical Bulletin (ExxonMobil, 2001), Properties of methane from manufacturers data sheets

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Methane</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical Formula</td>
<td>CH₄</td>
<td>C₁₂H₂₆</td>
</tr>
<tr>
<td>Molecular Weight</td>
<td>16.0</td>
<td>170.0</td>
</tr>
<tr>
<td>LHV (MJ/kg)</td>
<td>0.647</td>
<td>~840</td>
</tr>
<tr>
<td>Stoich Air/Fuel</td>
<td>17.2</td>
<td>14.5</td>
</tr>
<tr>
<td>Cetane Number</td>
<td>~0.0</td>
<td>40-55</td>
</tr>
<tr>
<td>Flammability U Limits</td>
<td>15.0/5.0</td>
<td>7.5/0.6</td>
</tr>
</tbody>
</table>

3. EXPERIMENTAL APPARATUS

The details of the engine used in this study are given in Table 2. This is typical of engines used in small diesel genset applications between 20 and 60kVA.

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Lister-Petter 4x90, DI, 4-stroke, naturally aspirated diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Configuration</td>
<td>Vertical in-line 4-cylinder</td>
</tr>
<tr>
<td>Cylinder Bore x Stroke</td>
<td>90 x 90 mm</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>138 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>18.5:1</td>
</tr>
<tr>
<td>Total displacement</td>
<td>2.29 liters</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>1800 rpm</td>
</tr>
<tr>
<td>Rated Power</td>
<td>37.5 kW at 2100 rpm</td>
</tr>
</tbody>
</table>

Table 2: Engine Specifications
Figure 1 shows the schematic diagram of the engine and test equipments. The engine was coupled to a Heenan-Dynamatic MkII 220kW eddy-current dynamometer which controlled and measured torque and speed, with a maximum error in speed of +/- 1 rpm and +/-2 Nm in torque. Intake airflow was measured using a laminar viscous flow air meter with a type 5 Cussons manometer. Inlet air depression was measured by a Druck type general purpose pressure transducer coupled to a digital readout. Various temperatures around the engine were measured via 'K type' thermocouples and diesel fuel consumption was recorded using a volumetric fuel measurement system.

![Fig.1: Schematic diagram of the engine and equipment](image)

High-speed data, comprising of cylinder pressure, fuel line pressure and crank angle were acquired using a National Instruments PCIO-MX16-E PC-BNC rack interface coupled with a BNC 2090 capture board. Cylinder pressure was measured using a Kistler type 6053B60 piezocapacitive transducer connected to a Type 5011 charge amplifier. Fuel line pressure was obtained using a Kistler 4065A piezoresistive sensor and 4617A amplifier. This data was recorded at a resolution of 0.5 degrees crank angle on the falling edge of the crank degree marker signal from an AVL optical encoder, mounted directly on the engine crankshaft. Emissions measurements were obtained using an AVL 415 Variable Sampling Smoke Meter for smoke and a Horiba MEXA-7100 HEGR exhaust gas analyzer system for NOx (using a chemiluminescent method).

A central point mixing system is an inexpensive and straight forward method of admitting a gaseous fuel to the dual fuel engine, (although the penalty will be high emissions of UHC’s) [5]. Therefore; a simple venturi type gas mixer valve was installed at a distance of ten pipe diameters upstream of the inlet manifold to ensure complete mixing of air and methane were achieved before being inducted to the combustion chamber, as illustrated in Figure 2. Methane flow rate was controlled by a needle valve located immediately upstream of an Omega FMA 1610 mass flow meter, which also recorded line pressure and gas temperature. The only other modification made to the engine was the replacement of standard with reduced flow injectors in order to improve injection performance.

![Fig.2: Schematic of gas installation](image)

The typical duty cycle of this type of engine is to operate for 90% of the time between 25% and 75% load [6]. Therefore, engine performance and emissions data were obtained under steady state operating conditions at 0, ¼, ½, ¾ and full load conditions at 1500 rpm (synchronous speed), for various concentrations of methane.

4. RESULTS AND DISCUSSION

4.1 Combustion Phasing

Mass burning rates cannot be analysed in isolation, as factors such as delayed Start of Combustion (SOC) will be shown to be primarily a function of injection timing before they are affected by fuelling levels. Therefore the analysis begins with a comparison of combustion phasing parameters.

The combustion phasing data that were obtained from the three-zone model are presented in Table 3. Data was sampled at a resolution of 0.5 degree crank angles, which is too coarse to allow for a detailed analysis, but does allow gross trends to be compared. Combustion duration was established as the time in crank angle degrees from SOC to 90% mass fraction burned. Where Start of Injection (SOI) and SOC are constant, combustion duration increases with increase gaseous fuel concentration. As the gaseous fuel equivalence ratio increases, the reaction zone surrounding pilot fuel also increases, and combustion takes longer. The only exceptions to this trend that are observed at low equivalence ratios are where there is a large degree of variation in SOI resulting in an earlier or later pilot injection (such as at $\Phi = 0.09$ at no load). SOC data correlates more closely with SOI than methane concentrations, which would imply that SOI has the primary influence on SOC.
<table>
<thead>
<tr>
<th>Methane (Φ)</th>
<th>Pmax (bar)</th>
<th>Pmax Location</th>
<th>Pmax Sdev</th>
<th>Power (Nm)</th>
<th>SOC (CA deg)</th>
<th>Duration (CA)</th>
<th>SOI (CA deg)</th>
<th>Ignition Delay</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Load</td>
<td>0.00</td>
<td>45.40</td>
<td>4.00</td>
<td>0.48</td>
<td>2.7</td>
<td>-3.0</td>
<td>27.5</td>
<td>-9.5</td>
</tr>
<tr>
<td></td>
<td>0.01</td>
<td>49.03</td>
<td>4.00</td>
<td>0.37</td>
<td>2.8</td>
<td>-4.0</td>
<td>28.0</td>
<td>-11.5</td>
</tr>
<tr>
<td></td>
<td>0.05</td>
<td>44.48</td>
<td>4.00</td>
<td>0.47</td>
<td>2.7</td>
<td>-2.5</td>
<td>28.0</td>
<td>-9.5</td>
</tr>
<tr>
<td></td>
<td>0.09</td>
<td>45.20</td>
<td>4.50</td>
<td>0.49</td>
<td>2.7</td>
<td>-2.0</td>
<td>27.5</td>
<td>-7.5</td>
</tr>
<tr>
<td></td>
<td>0.13</td>
<td>45.61</td>
<td>5.00</td>
<td>0.64</td>
<td>2.6</td>
<td>-2.5</td>
<td>28.5</td>
<td>-8.0</td>
</tr>
<tr>
<td></td>
<td>0.14</td>
<td>45.42</td>
<td>5.00</td>
<td>0.88</td>
<td>2.7</td>
<td>-2.0</td>
<td>30.0</td>
<td>-9.5</td>
</tr>
<tr>
<td></td>
<td>0.15</td>
<td>44.11</td>
<td>5.00</td>
<td>0.66</td>
<td>2.4</td>
<td>-1.5</td>
<td>28.0</td>
<td>-9.0</td>
</tr>
</tbody>
</table>

| ¼ Load      | 0.00       | 50.24         | 4.00      | 0.50      | 4.6          | -4.5          | 28.5         | -11.5         | 7.0           |
|             | 0.02       | 49.98         | 4.00      | 0.47      | 4.6          | -4.5          | 28.5         | -11.5         | 7.0           |
|             | 0.06       | 49.32         | 5.00      | 0.51      | 4.5          | -4.5          | 29.0         | -11.0         | 6.5           |
|             | 0.12       | 46.99         | 4.50      | 0.64      | 4.6          | -3.0          | 28.5         | -10.0         | 7.0           |
|             | 0.17       | 48.16         | 5.00      | 0.83      | 4.5          | -2.5          | 29.5         | -10.5         | 8.0           |
|             | 0.20       | 45.99         | 5.00      | 0.90      | 4.5          | -2.0          | 29.0         | -10.0         | 8.0           |
|             | 0.29       | 47.35         | 5.00      | 0.51      | 4.6          | -3.5          | 30.5         | -11.5         | 7.5           |

| ½ Load      | 0.00       | 49.74         | 7.00      | 0.40      | 9.5          | -1.5          | 30.5         | -8.0          | 6.5           |
|             | 0.03       | 49.40         | 0.50      | 0.43      | 9.4          | -1.5          | 30.5         | -8.0          | 6.5           |
|             | 0.09       | 48.27         | 6.00      | 0.47      | 9.6          | -1.5          | 29.0         | -8.0          | 6.5           |
|             | 0.18       | 52.76         | 5.00      | 0.59      | 9.3          | -4.0          | 30.5         | -11.5         | 7.5           |
|             | 0.27       | 47.14         | 6.00      | 0.75      | 9.4          | -3.0          | 31.5         | -10.5         | 7.5           |
|             | 0.29       | 49.49         | 6.50      | 2.90      | 9.3          | -3.0          | 32.0         | -10.5         | 7.5           |
|             | 0.31       | 50.52         | 6.50      | 1.15      | 9.2          | -3.0          | 31.0         | -10.5         | 7.5           |
|             | 0.33       | 51.40         | 7.00      | 1.33      | 9.2          | -3.0          | 32.0         | -10.5         | 7.5           |

| ¾ Load      | 0.00       | 49.01         | 8.50      | 0.37      | 14.1         | -0.5          | 31.0         | -6.5          | 6.0           |
|             | 0.05       | 49.63         | 8.00      | 0.47      | 14.0         | -0.5          | 31.0         | -6.5          | 6.0           |
|             | 0.13       | 51.10         | 8.00      | 0.52      | 14.1         | -1.0          | 32.0         | -7.5          | 6.5           |
|             | 0.27       | 53.60         | 7.00      | 0.79      | 13.8         | -2.5          | 33.0         | -9.5          | 7.0           |
|             | 0.41       | 51.37         | 8.50      | 1.24      | 14.2         | -2.5          | 33.5         | -9.5          | 7.0           |
|             | 0.44       | 51.68         | 9.00      | 2.10      | 14.1         | -1.5          | 33.0         | -8.5          | 7.0           |
|             | 0.47       | 48.59         | 10.00     | 1.48      | 14.6         | -1.0          | 34.5         | -8.0          | 7.0           |
|             | 0.70       | 50.32         | 11.50     | 1.24      | 15.9         | -1.5          | 32.0         | -7.5          | 6.0           |

| Full Load   | 0.00       | 52.57         | 9.00      | 0.54      | 18.5         | -1.0          | 34.0         | -7.0          | 6.0           |
|             | 0.06       | 52.09         | 9.00      | 0.45      | 18.5         | -0.5          | 32.5         | -6.5          | 6.0           |
|             | 0.17       | 52.41         | 9.00      | 0.47      | 18.5         | -0.5          | 33.5         | -6.5          | 6.0           |
|             | 0.34       | 56.38         | 8.50      | 0.66      | 18.6         | -1.5          | 33.0         | -8.0          | 6.5           |
|             | 0.51       | 61.54         | 9.00      | 2.89      | 18.4         | -3.5          | 29.0         | -10.5         | 6.5           |
|             | 0.55       | 53.55         | 12.00     | 2.58      | 18.4         | 0.0           | 32.0         | -6.5          | 6.5           |

Table 3 – Comparison of Combustion Phasing Parameters at 1500 rpm

The data from ¼, ½ and full load support the observation that ignition delay increases with gaseous fuel addition. Where there is a departure from this trend (no load and ¾ load), SOI has occurred earlier or later. However, data for ignition delay is subject to the largest degree of error as ignition delay is found from measured SOC and SOI: both parameters could be subject to up to 0.49 degrees crank angle.

Cyclic variability within the data was assessed by recording the peak pressure in each of the successive 50 engine cycles logged for each test point, and then taking the standard deviation of the maximum pressure and the...
location of the maximum pressure. A more straightforward trend is shown here (which figure?). As the gaseous fuel concentration is increased, so are the cycle to cycle variations until the point where a typical dual fuel combustion pattern is observed such as at the maximum gas substitution levels at ¾ and full load. At these points, the in cylinder conditions allow for a stable combustion process causing a decrease in variability. The same trends are repeated in location of peak cylinder pressure.

4.2 Mass Burned Analysis

The data presented in Figure 3 is on the basis of the methane equivalence ratio that was calculated from experimental data. The features of particular interest are the peak rate of mass burned of the premixed engine charge, the following diffusion burning phase and finally the late burning phase [7]. Additionally, the initial mass burning rates (up to 3 degrees crank angle after SOC) also provide a useful comparison.

![Graph of mass burned at different loads](image)

4.2.1 Results at No Load

Figure 3a shows the rate of mass burned with increasing methane equivalence ratio at 1500 rpm with no load. It shows that as methane concentration is increased (with the exception of $\Phi = 0.01$), start of combustion is delayed and consequently position of peak cylinder pressures occur later, and maximum cylinder pressures are lower. This trend correlates with the delayed SOI observed as the pilot quantity is reduced.

At $\Phi = 0.01$ there is an earlier SOI, resulting in higher peak pressures and a greater initial mass burning rate. However, combustion duration is increased compared with the diesel and $\Phi = 0.04$ cases, while there is almost no change to the diffusion burning period compared with the higher methane substitution levels. The similar diffusion burning period and extended combustion duration are most likely caused by near complete combustion of gaseous fuels in the crevice volumes that are engulfed by the wide pilot reaction zone, itself a function of injection timing.

Increased addition of methane results in increased rates of both premixed and diffusion burning, until the highest substitution level. The data at $\Phi = 0.09$ and $\Phi = 0.14$ have the same SOC and data at $\Phi = 0.05$ and $\Phi = 0.14$ have the same SOI. For these four test conditions the observed effects can be related to the fuelling strategy: initial mass burning rates are slowed, peak cylinder pressures are reduced and delayed and combustion duration increases. At $\Phi = 0.05$, the initial mass burning rates are the highest of the four test conditions, but peak
cylinder pressure is lowest. This suggests that while the initial mass burning rates are a function of the pilot, the magnitude and position of the premixed peak are a function of methane equivalence ratio (i.e. the reaction zone surrounding ignition centres).

The highest equivalence ratio \((\Phi = 0.15)\) exhibits a different trend. Injection timing is delayed by the greatest extent for this data set due to the small pilot quantity. However, combustion duration decreases slightly, while diffusion burning rates increase. This suggests that the reaction zone extending from ignition centres has increased with methane concentration, but when the pilot is consumed, combustion ends. This is an example of a transition combustion pattern that can occur between the classic diesel and dual fuel type combustion mechanisms as methane concentration is increased.

As the pilot quantity is reduced, the fuel spray becomes increasingly dispersed amongst the cylinder with a deleterious effect on atomisation and penetration characteristics. This leads to the increasingly sluggish initial mass burning rates and when the pilot characteristics degrade, methane consumption is also reduced. Whilst the pilot quantity is sufficiently high to maintain a positive ignition source, a larger proportion of the gaseous fuel is consumed during the premixed combustion phase.

4.2.2 Results at ½ Load

Figure 3b shows the rate of mass burned with increasing methane equivalence ratio at 1500 rpm with ¼ load. For the diesel case and \(\Phi = 0.02\) and 0.06, injection and combustion timing remained constant. Increased methane concentrations have a deleterious effect on the initial mass burning rates, but mass burning rates then increase during premixed burning.

SOC is delayed at the methane equivalence ratio of 0.2 with the result that peak cylinder pressures are reduced and delayed. When compared against the 0.12 and 0.17 cases, it can be seen that the diffusion burning rates increase with methane concentration. The effect of increased methane fuelling levels is that initial mass burning rates are slowed, and peak burning rates and peak pressures are reduced through the reduction of the pilot. The rates of mass burning during the diffusion phase are significantly higher due to the higher methane equivalence ratio. The late burning phase is much increased compared to the other dual fuel cases, and combustion duration is extended.

4.2.2 Results at ¼ Load

The effect of increasing methane concentration is shown in Figure 3c where much higher substitution levels were achieved. The main effect of increasing load is to increase the in cylinder temperature, which is known to widen the flammability limits of the gaseous fuel leading to more stable operation [3]. The first three data sets (diesel, \(\Phi = 0.03\) and 0.09) have the same injection timing and the effect of the addition of very small quantities of methane seems to be to promote combustion; the reduction in pilot quantity is so small that it is unlikely to affect penetration and atomisation characteristics, while the addition of methane widens the reaction zone surrounding each ignition centre.

At the four highest methane concentrations, injection timing was constant. Comparing these results it can be seen that increased primary fuelling levels increases the premixed peak and cylinder pressures. The increased methane levels and maximum pressures result in higher diffusion burning rates, and a slightly higher late burning phase. The increased methane levels extended ignition delay resulting in a later SOC and longer combustion durations. The initial mass burning rates and peak cylinder pressures are lower at higher primary fuelling levels but the diffusion burning rates are significantly higher.

4.2.3 Results at ¾ Load

At ¾ load as shown in Figure 3d, the pairs of results at \(\Phi = 0.13\) and 0.7; and \(\Phi = 0.27\) and 0.44 were subject to the same injection timing. As the methane concentration is increased, the same delayed initial mass burning rates and reduced peak cylinder pressures and premixed peak are observed whilst diffusion burning rates are elevated. When the cases at \(\Phi = 0.27\) and 0.44 are compared, the lower equivalence ratio has a faster mass burning rates during the premixed phase, but lower diffusion burning rates. The increased mass burning rate during the premixed phase compared with the diesel baseline case could be explained on the basis of an extended ignition delay, but this also demonstrates a continued dependence on the pilot throughout the combustion process. This pattern is characterised by the diffusion burning rates making a much more significant contribution to heat release, and the combustion process will complete quickly if the pilot is still large enough to experience good penetration and atomisation processes. However; when the pilot ceases to ignite, combustion will end.

At ¾ load, the combustion process was not limited by misfire and higher substitution levels were obtained than at ½ load. The highest methane substitution levels show a further, different combustion pattern that much more closely mirrors the ‘classic’ dual fuel profile [1]. Increasing methane concentrations lead to a greater contribution to mass burning rates from the diffusion phase. The premixed and diffusion burning phases remain distinct at the higher fuelling levels, but previous analysis have suggested that this pattern shows a transition from diesel combustion to a flame propagation processes. This is an uncomfortable explanation as it is based on the fact that the much reduced pilot quantity, which is known to suffer from poor atomisation and penetration, is still consumed first and quickly before the flame propagation process begins. These results have consistently shown that the reduction of pilot quantities slow initial mass burning rates, therefore the whole of the pilot is unlikely to be consumed during the initial combustion phases.
4.2.4 Results at Full Load

At full load, as shown in Figure 3e, the engine was more difficult to control with the greatly reduced pilot quantities, and power output was ultimately limited by knock. The results for $\Phi = 0.06$, 0.17 and 0.55 experienced the same ignition timing, with the higher methane levels showing a later SOC and extended ignition delay. It can be seen in figure 5 that the premixed and diffusion burning peaks remain distinct; however the diffusion burning period (and hence primary fuel) makes a greater contribution to overall heat release rates.

It is known that there is a need to optimise injection timing according to primary fuelling levels, and the much earlier SOI recorded for the shows this effect. The earlier SOC results in higher initial rates of heat release (even though the pilot quantity is reduced), and elevated peak cylinder pressures. The mass burning rate is much closer to those observed by previous researchers where the premixed and diffusion burning processes become indistinct. However; this data set does not support the notion that there is a different combustion regime. This is because a flame propagation process is slower than diffusion combustion, particularly in lean fuel air mixtures, and this data set is showing a much faster combustion process than the corresponding diesel or dual fuel cases with higher methane equivalence ratios. It is more likely that the pilot is mixing and becoming prepared for combustion at an increasingly fast rate, thus more of the combustion process is premixed. There is still slight evidence of a diffusion burning phase, but it is less distinct because as more fuel has been consumed earlier, less is available towards the end of the cycle.

The data set of figure 5 shows the clearest difference and changes in combustion patterns of all the preceding figures, and what is striking is that at no point is there any evidence of separate stages in the initial rates of combustion. Initial burning rates are primarily a function of SOI, but where this is held constant, initial burning decreases as pilot quantity is reduced.

4.2.5 Effect of increasing load

The effect of increasing load is to increase the temperatures inside the cylinder, and as a result, the flammability limits of the primary fuel are widened. This is illustrated in Figure 4 for varying equivalence ratios but where around 70% of the diesel energy was replaced by methane.

This type of comparison is more typical of previous research [8-10], where the dual fuel engine uses a near constant pilot quantity and power is controlled through varying the methane concentration. The data shows a definite transition from a CI type combustion pattern where the primary fuel concentration is lean and makes little contribution, to a dual fuel pattern where the gaseous fuel makes the major contribution to heat release rates. This type of analysis has also been the basis for the discussion of the three stages of dual fuel combustion theory [1, 2, 8]. However; none of the data here supports the argument that the pilot initially burns in two stages. In all data sets there is a smooth increase in initial mass burning rates, and no evidence of a step change from stage I (around half of the pilot and entrained methane) to stage II (remaining pilot with a wide reaction zone).

The data at full load and half load experienced the same injection timing (10.5 degrees CA BTDC), and it can be seen that the increased load decreased the ignition delay resulting in an earlier SOC. This result therefore shows the strong dependence of SOC on the pilot quantity which was larger at full load. The full and half load data sets also show that the initial mass burning rates are lower for the lower pilot and longer ignition delay at ½ load. It can be concluded that initial rates of heat release are a primarily a function of the pilot spray characteristics.

Combustion duration is increased as load is increased except at full load. This causes difficulties with the traditional argument that the dominant combustion pattern is one of flame propagation. If SI type flame propagation ever occurs it would occur at full load, and combustion duration should increase with load. When the mass burning patterns for diesel are compared, it can be seen that combustion duration increases with load, but by a greater extent than is observed for the dual fuel cases. This suggests that the dual fuel combustion processes are relatively faster. The shortened combustion duration at full load can be explained on the basis that combustion ends when the small pilot quantity is consumed.

The full load data can also be compared with the ¼ load data as the two sets share the same SOC. The full load case has a shortened ignition delay, although this is more likely to be caused by the larger pilot. The initial mass burning rates are lower with the lower pilot quantity at full load, but thereafter the full load combustion process proceeds more quickly. This shows that the reaction zone surrounding each ignition site is widened, leading to higher rates of temperature and pressure rise, which enhances atomisation and mixing processes of the reduced pilot quantity. This implies that for dual fuel combustion, (even at high methane...
substitution levels), the combustion of the pilot has a dominant effect.

The combined effect of increasing load (i.e. in cylinder temperatures) and methane concentrations is that initial mass burning rates decrease and diffusion burning rates are enhanced. Combustion duration is increased, except for the full load case, which shows a different shape to diesel mass burning patterns; there is no distinction between the two premixed and diffusion burning. The merging of these two phases supports the idea that the pilot acts only as an ignition source, however the shortened combustion duration supports the argument that this is not a flame propagation process: when the pilot is consumed, combustion ends.

A second means by which the data may be compared is for (approximately) the same equivalence ratio at increasing load, as shown in Figure 5. This analysis allows the effects of increasing methane concentration to be separated (to some extent) from temperature. This fuelling strategy is not, however, typical of practical dual fuel conversions.

When the data is compared for (approximately) the same equivalence ratio, the combustion pattern remains the same, except that more fuel is burned at higher loads in both the premixed and diffusion burning phases. The transition from diesel to dual fuel patterns does not seem to occur as a function of temperature alone at these relatively low equivalence ratios. However, as both the premixed and diffusion burning phases increase by approximately the same amount with load, it suggests that the same overall combustion process occurs as temperature increases, and any change in combustion mechanism is primarily a function of fueling levels, with temperature having a secondary effect.

5. EMISSIONS RESULTS AND DISCUSSION

5.1 BsNOx

The formation of NOx is complicated in CI engines where fuel and air mix at a range of equivalence ratios and fuel is burned due to spontaneous ignition and flame propagation from the ignition sites. This leads to temperature variations due to fuel and air mixing, in addition to temperature variations during combustion and expansion. Most CI NOx is formed when burned gas temperatures are at a maximum (between SOC and just after peak pressure). The burned gasses then expand and mix with cooler unburned charge, quickly freezing NOx chemistry.

The dependence of NOx formation on equivalence ratio is explained on the basis of oxygen availability and combustion temperatures. Under fuel rich conditions, oxygen is limited, while in fuel lean mixtures, NOx chemistry freezes early in the expansion stroke due to mixing with cooler unburned charge. The result of both scenarios is low NOx emissions.

Another important factor is combustion timing. When SOC is advanced, NOx emissions are higher as peak pressures and hence temperatures are higher where more fuel is burned before TDC. Similarly, NOx formation is reduced due to lower peak temperatures when SOC is retarded.

As the speed and load tended to fluctuate at each test point, the emissions of NOx are normalised against power to simplify the analysis. The results are presented in Figure 6 are at 1500 rpm for each load at increasing gaseous fuel concentrations. (give a definition of BsNOx here before using it)

![Fig.6: BsNOx at 1500 rpm](image)

At no load (NL) and ¼ load (QL), power was constant within experimental tolerances and emissions of BsNOx decrease with increasing methane equivalence ratio. This correlates closely with an increasingly delayed SOC and the lower initial mass burning rates as well as the elevated diffusion burning rates. The higher diffusion burning rates imply a widening to the pilot reaction zone, which consumes more methane at low equivalence ratios and hence lower temperatures. The decreasing BsNOx trend also follows from a decreasing pilot quantity in terms of extended ignition delay and lower initial mass burning rates. This is consistent with the explanation of NOx formation for CI engines, and confirms that the mass burning analysis did show a strong reliance on the pilot combustion process even at the highest methane substitution levels.

At ½ (HL), ¾ (3Q) and full load (FL) BsNOx at first increases then eventually decreases with increasing equivalence ratios. Again, this result primarily correlates with SOC (table 3), but where this was held the same BsNOx is a strong function of initial then diffusion burning rates: low initial mass burning and high diffusion burning lead to reduced BsNOx. Where
the combustion pattern was identified as a classic dual fuel process, BsNOx formation rates fall and this is due to a smaller pilot and a greater quantity of fuel being consumed late in the expansion process.

The transition between the diesel and dual fuel combustion mechanisms is more difficult to describe, as BsNOx increases even though the pilot decreases. This is most likely to be due to an increasing, but still lean methane air fuel ratio which widens the reaction zone of a pilot that was sufficiently large so that injection performance was not degraded. It is also worth noting that NOx chemistry is not ‘frozen’ as quickly as when the engine is fuelled with diesel alone due to the widened reaction zone surrounding the pilot droplets. In all data sets, the initial mass burning rates and pilot quantity are better predictors of BsNOx than gaseous fuel concentration and hence the diffusion burning phase.

As load increases, the BsNOx emissions decrease, which is would be an expected trend in data where emissions are normalised against power. The higher load burned gases contain increasing amounts of H2O and CO2, which increase the specific heat capacity of the charge, thus lowering flame temperatures. Once again, the fundamental reliance on a heterogeneous combustion process is demonstrated.

In terms of NOx formation, the most consistent explanation at all loads and methane concentrations is on the basis of initial mass burning rates (and hence initial temperatures), and the dependence on these initial rates is more important than on any other parameter except SOC. As these initial rates are a function of the ignition process, and this is a function of the pilot, it would suggest that the combustion mechanism is consistently dominated by the pilot processes, and that dual fuel NOx tends to be formed in the initial stages of combustion. This supports the idea that even at high loads, high speed and high methane concentrations, the pilot continues to be the main controlling parameter.

5.2 Smoke

Soot forms primarily from the carbon in the fuel due to incomplete combustion where there is insufficient air. Smoke emissions are a significant issue in CI engines where combustion occurs at stoichiometric conditions and mixing with air is a function of the injection process.

Soot first forms nuclei from hydrocarbon oxidation products and or pyrolysis. Then the particles then ‘grow’ through the attachment of gas phase species to the surface of the particles, and then by coagulation when the particles collide. These particles can then also form clusters or combine in chains with the result that there is a large distribution in particle size within exhaust emissions of soot. There is also a large spatial distribution in soot particles within the combustion chamber with the highest concentrations occurring in the fuel rich centre of the fuel spray, then reducing with increasing distance from this point. At any time during the formation process, the soot can be oxidised to form CO and CO2, and 90% of the soot formed is then consumed.

In the present research smoke opacity measurements were taken at each operating condition. Visible smoke opacity gives an indication of the mass levels of particulate matter or soot, but no conclusions regarding ultra-fine particulate matter can be drawn.

Figure 7 illustrates that increased methane fuelling levels reduce emissions of visible smoke. This has been attributed by previous researchers to the fact that methane contains no heavy hydrocarbons, and therefore less soot is formed [8]. It is also logical that levels would be lower as the pilot spray is increasingly reduced, but the core will always be very fuel rich and this suggests that soot should still be formed in significant quantities in the engine. There is a close correlation between soot levels and the length of the diffusion burning process (where significant quantities of smoke are oxidised). The longer diffusion burning processes that occur at high methane substitution levels result in reduced smoke emissions.

As load is increased, smoke emissions increase. But increasing load results in a longer combustion duration, which includes an extended diffusion burning period. If dual fuel smoke emissions could be completely explained on the basis of diffusion burning periods, then they should decrease with increased load. Further inspection of Figure 7 shows that at the highest methane substitution levels, and classic dual fuel combustion patterns, smoke emissions start to increase again. This can partly be explained by an increasingly rich overall fuel air ratio, but the high levels of methane should always reduce smoke emissions, and this is not the case here. The only consistent correlation is provided by the power fluctuations, which result in increased pilot quantities. Whenever pilot quantities increase, so does visible smoke.

5. CONCLUSIONS

Irrespective of the methane-air equivalence ratio, the combustion process remains dependent on the pilot: when the pilot is consumed, combustion ends. Reducing the pilot quantity has a deleterious effect on spray atomisation and penetration characteristics. This leads to the increasingly sluggish mass burning rates. When the methane concentration is increased, but remains too fuel-lean to sustain a wide reaction zone, diffusion burning rates remain low and a function of the pilot ignition...
processes. Whilst the pilot quantity is sufficiently high to maintain a positive ignition source, a larger proportion of the gaseous fuel is consumed during the premixed combustion phase. As soon as the pilot characteristics degrade, methane consumption is also reduced.

The main effect of increasing load is to increase the in-cylinder temperature, which is known to widen the flammability limits of the gaseous fuel leading to more stable operation. The increased methane levels and maximum pressures result in higher diffusion burning rates, and a slightly higher late burning phase. The initial mass burning rates and peak cylinder pressures reduce with reduced pilot quantity.

There is a noticeable change in the combustion pattern with increasing load, from a pattern that mirrors diesel, to one that is can be identified as ‘dual fuel’. In all data sets there is a smooth increase in initial mass burning rates, and no evidence of a step change from stage I (50% pilot and entrained methane) to stage II (remaining pilot with some flame propagation). The combined effect of increasing load (i.e. in cylinder temperatures) and methane concentrations is that initial mass burning rates decrease and diffusion burning rates are enhanced.

The full load case shows a different shape to diesel mass burning patterns; there is no distinction between the two premixed and diffusion burning. The merging of these two phases supports the idea that the pilot acts only as an ignition source, however the shortened combustion duration supports the argument that this is not a flame propagation process: when the pilot is consumed, combustion ends. It is likely that the classic flame propagation explanation of the dual fuel combustion mechanism applies only to IDI diesel engines.

In all data sets, the initial mass burning rates and pilot quantity showed the strongest correlations to BsNOx. At low load, the decreasing BsNOx trend also follows from a decreasing pilot quantity in terms of extended ignition delay and lower initial mass burning rates. This confirms that the mass burning analysis did show a strong reliance on the pilot combustion process even at the highest methane substitution levels.

At high load, BsNOx remains a strong function of initial mass burning rates, then diffusion burning rates: low initial mass burning and high diffusion burning lead to reduced BsNOx. Where the combustion pattern was identified as a classic dual fuel process BsNOx formation rates fall and this also follows from reduced fuel consumption in the early parts of the combustion process.

There is a close correlation between soot levels and the length of the diffusion burning process (where significant quantities of smoke are oxidised). The longer diffusion burning processes that occur at high methane substitution levels result in reduced smoke emissions. The only consistent correlation is provided by the power fluctuations, which change pilot quantities. Whenever pilot quantities increase, so does visible smoke.

The combined effect of dual fuelling a DI engine with methane is to reduce both NOx and smoke emissions. This technology may provide a beneficial method to manipulate the classic diesel engine NOx-smoke tradeoff.

REFERENCES