

Some Effects of Fuel Autoignition Quality and Volatility in Premixed Compression Ignition Engines

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ABSTRACT

Previous work has shown that it may be advantageous to use gasoline type fuels with long ignition delays compared to today's diesel fuels in compression ignition engines. In the present work we investigate if high volatility is also needed along with low cetane (high octane) to get more premixed combustion leading to low NO_x and smoke. A single-cylinder light-duty compression ignition engine is run on four fuels in the diesel boiling range and three fuels in the gasoline boiling range. The lowest cetane diesel boiling range fuel (DCN = 22) also has very high aromatic content (75%vol) but the engine can be run on this to give very low NO_x ($< 0.4 \text{ g/kWh}$) and smoke (FSN < 0.1) e.g. at 4 bar and 10 bar IMEP at 2000 RPM like the gasoline fuels but unlike the diesel fuels with DCNs of 40 and 56. If the combustion phasing and delay are matched for any two fuels at a given operating condition, their emissions behaviour is also matched regardless of the differences in volatility and composition. However the same high aromatic, low cetane, diesel fuel can match different gasoline fuels depending on operating conditions - a result in line with previous understanding of autoignition quality from HCCI and knock studies.

1. INTRODUCTION

Conventional diesel fuels used in current diesel engines have cetane numbers (CN) greater than ~ 40 and will auto-ignite very quickly after the start of injection into the cylinder. Particularly at high loads, fuel continues to be injected after the combustion has started and this leads to high NO_x and particulates. If the mixing is accelerated or if the chemical reaction is slowed down, auto-ignition can be made to occur after the fuel and air are better mixed and soot levels can be

reduced. NO_x levels can be reduced by reducing combustion temperature by either running lean, pre-mixed or using EGR, exhaust gas recirculation. In fact, a lot of technology is used in advanced diesel engines, making them expensive and complicated, to counteract the high ignitability of the diesel fuel in order to promote premixed combustion. Thus high injection pressures and high swirl are used to increase mixing rates while high levels of cooled EGR are used to delay combustion; this in turn requires higher boost pressures to achieve the required loads. Even then, with conventional diesel fuels, low NO_x and low smoke with partially premixed CI combustion is possible only at low loads. Modern engines now require after-treatment systems to further reduce NO_x and particulates, making them even more complicated and expensive. Moreover, regulations on emissions from diesel engines are becoming more stringent in all parts of the world. Hence there is a great deal of interest in developing engine combustion systems that offer the efficiency of a diesel engine, but with low smoke and low NO_x . Several operating strategies have been employed to promote pre-mixed and low-temperature combustion in CI (Compression Ignition) engines. In the Toyota UNIBUS (Uniform Bulky Combustion System) a double-injection strategy is used [1] with some fuel injected very early in the cycle to get better mixing with air. Nissan's MK (Modulated Kinetics) design utilizes high levels of EGR in conjunction with high swirl [2] and late injection to promote fuel/air mixing by increasing the ignition delay. Reduction of NO_x could also be achieved by PREDIC (Premixed Lean Diesel Combustion) at low to part loads and its evolution MULDIC (Multiple Stage Diesel Combustion), with multiple fuel injectors and injections, at higher loads [3]. In homogeneous-charge compression-ignition (HCCI) fuel and air are fully premixed as in a spark ignition engine and compressed till the charge auto-ignites. In modern direct-

injection (DI) diesel engines with common rail injection systems, HCCI combustion can be obtained by injecting the diesel fuel very early in the cycle, giving it enough time to mix completely with air. However, with HCCI, the phasing of the heat release will be determined by the initial conditions of the mixture at the start of the compression stroke and the auto-ignition characteristics of the fuel. There is no in-cycle control of combustion phasing with HCCI unlike with the spark timing in SI engines or the injection timing in diesel engines. Hence HCCI combustion is very difficult to control and is also limited to low loads. Practical CI engines always operate in a fuel injection timing range - injection near TDC (Top Dead Centre) - where combustion phasing can be varied by varying injection timing.

Kalghatgi and co-workers have demonstrated both in large [4, 5] and small [6, 7] single-cylinder research engines that if a gasoline-like fuel is injected near, but before, TDC, it ignites much later than diesel fuel and combustion occurs when fuel and air have had more chance to mix. If the same amount of gasoline is injected early (e.g. before 50 CAD BTDC) at the same conditions i.e. with fully premixed, HCCI conditions, ignition might not occur at all. Thus the inhomogeneity is essential for combustion to occur but the high ignition delay makes combustion happen when fuel and air are better mixed - fuel and air are "premixed enough" but must not be fully premixed. Under such conditions there will be some part of the mixture which will ignite which in turn will get the rest of the mixture to burn. This mixture stratification, which enables control of the phasing of combustion through injection timing, is easily achieved simply by relatively late injection compared to HCCI. Much higher loads can be achieved at very low smoke and NO_x levels with gasoline compared to diesel while retaining control over combustion. More recently groups at Lund [8,9,10] and Wisconsin [11, 12] universities have demonstrated that heavy-duty diesel engines can be run on gasoline as well as ethanol [8,9,10] with very high efficiency and very low NO_x and smoke. Weall and Collings have also demonstrated the benefits of low reactivity fuels by studying mixtures of gasoline and diesel [13] and have also addressed the issues arising from running a diesel engine on gasoline at low load and low speed [14].

Quite clearly CI engines can be run on fuels in the gasoline auto-ignition range if the correct injection timings and strategies are used. Moreover, the higher resistance to autoignition associated with such fuels allows more time for fuel to mix with air before combustion and greatly facilitates low NO_x and low smoke operation. Some of the problems associated with premixed CI combustion such as high pressure rise rates caused by high heat release rates can be alleviated by using multiple injections [4, 6, 8,9,10]. Gasoline also might not need the small injector holes and the high injection pressures required by diesel fuels to increase mixing rates. Since smoke and NO_x can be controlled at much higher

loads at a given speed, there might be opportunities to optimize the transmission system by "downspeeding", using gasoline in CI engines. Combustion of this type also leads to higher CO and HC emissions. Use of gasoline-like fuels in CI engines will also present problems associated with starting the engine. Thus there are many areas in which this approach needs further development and optimization [15].

The autoignition quality of practical fuels has been discussed in [16]. Conventional diesel fuels are significantly less volatile compared to conventional gasoline fuels. The autoignition quality of diesel fuels is described by the Cetane Number (CN) or the Derived Cetane Number (DCN) based on ignition delay measured in the standard IQT (Ignition Quality Test). Practical diesel fuels have CN/DCN greater than 40. The autoignition quality of gasoline fuels is described by their Research and Motor Octane Numbers - RON and MON. RON and MON and other traditional autoignition measures like ignition delays in shock tubes and rapid compression machines, relevant to gasoline autoignition quality, are measured in systems where fuel and air are fully premixed. In contrast, CN and DCN are also affected by the fuel / air mixing process. There is an inverse relationship between CN and the Research Octane Number (RON) where these can be measured for the same fuel [16] but usually this is not possible for many practical diesel fuels.

In all the pre-mixed CI studies described above, the fuels in the gasoline autoignition range are also in the gasoline boiling range and thus are far more volatile than the diesel fuels. It is not clear if the high volatility of the gasolines, as well as the high ignition delays, plays a role in promoting premixed combustion and reducing NO_x and smoke. In this study fuels in the diesel volatility range but with high ignition delay are compared with gasoline fuels. It is found that if the combustion phasing and delay are matched for two fuels at a given operating condition, their emissions and other pressure based parameters such as Maximum Pressure Rise Rate (MPRR) will also be matched regardless of volatility and chemical composition. The diesel fuel with the lowest cetane number tested (DCN = 22.5), unlike the diesel fuels with DCNs of 56 and 40, enabled the engine to be run with very low NO_x ($< 0.3 \text{ g/kWh}$) and smoke ($< 0.05 \text{ FSN}$) at 4 bar and 10 bar IMEP at 2000 RPM even though it had very high aromatics ($\sim 75\% \text{ vol}$).

2. EXPERIMENTAL SETUP

The experiments presented in this paper were all performed on a 4-valve single-cylinder research engine with dimensions as presented in Table 1. All experiments were done with coolant and oil temperatures at 90°C and the inlet air temperature was kept at 60°C . Fuel was injected via a Bosch 7 hole injector, with injector cone angle of 153° and hole diameters of 0.13 mm, fed by an independent fuel supply rig. An external air compressor was used to simulate boosted

Table 2. Fuel properties

	D1	D2	D3	D4	G3	G4	G5
Density [g/cm ³]	0.833	0.852	0.86	0.865	0.715	0.736	0.738
DCN	56.2	39.6	28.6	22.5	~ 24'	~ 19'	~ 17'
Aromatics [v%]	25.2	42.5	62.6	75.3	19	26.5	29.8
Heat of combustion [MJ/kg]	42.9	42.6	42.2	42.1	43.2	43.2	43.1
Sulphur [ppm]	7	10	7	<10	<10	<10	<10
Lubricity ² [μm]	285	332	245		275	241	254
RON					73	84	90.7
MON					68.5	78	81.8
H/C	1.97	1.78	1.62	1.59	1.997	1.874	1.84

1--estimated from Eq. 6 in [16] and 2--the European EN590 specification for lubricity requires the scar size to be below 460 μm

conditions. When EGR was introduced, the exhaust backpressure was set 0.2 bar higher than the inlet manifold air pressure and the recirculated gases were cooled using an external cooling circuit to the same temperature as the inlet air, i.e. 60°C. In-cylinder pressure was measured with a water-cooled pressure transducer (Kistler 6041A). Emissions and inlet CO₂ level were measured using a Horiba MEXA-9500H system and soot was measured using an AVL 415 smoke meter. After a stabilisation period, the emissions were logged once per second for 60 seconds and the average of those 60 recordings are what is presented in this paper. At the same time, the in-cylinder pressure was recorded for 250 cycles.

Four fuels in the diesel volatility range (D1-D4) and three fuels in the gasoline volatility / autoignition range (G3-G5) are considered in this paper. The volatility characteristics of these fuels are shown in [Figure 1](#) where the volume percent recovered at a given temperature in the ASTM volatility test is plotted against the temperature. Other fuel properties are listed in [Table 2](#). D1 is a commercial low-sulphur European diesel fuel with a DCN of 56 while D2, with a DCN of 39.6, is like a U.S. diesel. D3 and D4, with DCNs of 28.6 and 22.5, are mixtures of D2 and an aromatic solvent, Shellsol A100 with a boiling range between 162°C and 185°C. Hence D3 and D4 have very high aromatic content. We can define [16] the gasoline autoignition quality range as RON > ~ 60 or DCN < ~ 30. By this criterion, D3 and D4 are in the gasoline autoignition range and their autoignition properties should be really specified by RON and MON. Unfortunately, RON and MON cannot be measured in the CFR engine for these fuels because of their low volatility. If this had been possible, these fuels could be expected to have high sensitivity, $S = \text{RON} - \text{MON}$, because of the high aromatic content. The DCNs of the three gasoline fuels, G3, G4 and G5 are estimated using Equation 6 in [16]. The autoignition behaviour of D4 might be expected to lie between that of G3 and G4 but as the results in this work show, this is not necessarily the case. Of these seven fuels, all except D3 and D4 have been studied in

this engine previously [6, 7, 17]. All the fuels had a sufficient amount of lubricity additive (300 ppm of Paradyne R655 from Infineum) to ensure that the lubricity scar size was well within the European specification. It can also be seen from [Table 2](#) that all the fuels have similar gravimetric heat of combustion.

In the present case fuel consumption for the high volatility fuels was calculated from exhaust emissions because there were problems of stability with the fuel flow meter at the low flow rates encountered; this is less desirable than direct measurements of fuel consumption.

The aim of the current experiments is to study some fuel effects on low-NO_x / low smoke diesel operation at some fixed conditions rather than to develop a practical fuel/engine solution to meet fixed emissions and fuel economy targets. At fixed operating conditions we have tested fuels with different properties. The operating conditions such as injection pressures or intake temperatures and loads chosen have no special significance other than that they are well within normal diesel operating conditions.

Table 1. Engine dimensions

Compression ratio	16:1
Displacement	0.537 l
Bore	88 mm
Stroke	88.3 mm
Connection rod length	149 mm
I/O	362 CAD
I/V	595 CAD
E/O	143 CAD
E/V	385 CAD

<table 2 here>

3. RESULTS AND DISCUSSION

All the experiments described below were done with a single injection.

3.1. RESULTS AT 1200 RPM, 4 BAR IMEP, NO EGR

We will first consider the low load case at 1200 rpm. The inlet pressure was set to 1.1 bar absolute, the engine speed was fixed and a single injection at 650 bar injection pressure was used without any EGR. For each fuel, the fuelling rate was fixed to get an IMEP of around 4 bar at a CA50 (the crank angle degree at which 50% of total heat release has taken place) of 10 CAD ATDC (Crank Angle Degree After Top Dead Centre) for each fuel and then the start of injection (SOI) was varied, keeping the injection pulse width constant. The normalized global air/fuel, λ , is around 2.6. Over the SOI range considered, IMEP varies little for the high cetane diesel fuels and by up to 10% for the low cetane fuels. The emissions results are shown as indicated specific values to account for these changes in IMEP. The smoke level for all the fuels was low, < 0.08 FSN for D1 which had the highest level compared to the other fuels.

Figure 2a shows CA50 vs. the start of injection, SOI. Figure 2b shows the Combustion Delay, $CD = CA50 - SOI$, plotted against CA50. Of course, there are many different ways of defining the delay between combustion/ignition and the injection event. Previous work [5,6,7, 15] has shown that CD is a reasonably good parameter to help understand fuel effects in this type of partially premixed CI combustion. The larger the value of CD, the more premixed fuel and air are at the time of the main combustion. Figures 2c,2d,2e,2f show Indicated Specific NO_x (IS NO_x), Maximum Pressure Rise Rate (MPRR), Indicated Specific CO (ISCO) and Indicated Specific HC (ISHC) respectively plotted against CA50.

It can be seen from Figure 2b that there is only a small difference in CD for a given CA50 between D1 (56 DCN) and D2 (~ 40 DCN) and the DCN needs to be reduced below 28 to see a significant increase in CD at this condition. Thus the autoignition resistance of the fuel needs to be sufficiently high, i.e. DCN needs to be sufficiently low before a significant increase of CD can be seen [4, 6, 7] at a given condition. If autoignition becomes more difficult, e.g. if the compression ratio (CR) is decreased or if at a given condition the EGR level is increased, the difference in CD between fuels increases. Ignition / combustion delay parameters, such as CD, change non-linearly with conditions determining autoignition. Thus with a CR of 11.4:1 there was a noticeable difference in CD between two fuels of 54 DCN and 39.5 DCN [18] with consequent differences in emissions but this difference between the same fuels was very much smaller in the same engine but with a CR of 14:1 [4].

In Figure 2b, D4 is similar to G4, the 84 RON gasoline, while D3 behaves very similarly to G3, the 73 RON gasoline. The three fuels with the highest values of CD viz D4, G4 and G5, show significantly lower NO_x at the same CA50 compared to the other fuels. However, as discussed in [7] for gasoline-like fuels, if CD is increased moderately compared to the conventional diesel fuel - D2, D3 and G3 compared to D1 in Figure 2b - NO_x increases at a given CA50 (Figure 2c). Such behaviour was explained in [7] the following way. At the condition studied, the injection event is over before combustion starts for all the fuels. Thus when the first ignition occurs, there are packets of varying mixture strengths in the combustion chamber. The true variation of NO_x with mixture strength is likely to be qualitatively as indicated by the dashed line on the rich side and the solid line on the lean side, in the sketch in Figure 3. With the low ignition delay of the 56 DCN diesel fuel (D1), the mixture packets will mostly be rich when combustion starts. Autoignition will take place in rich mixture packets and produce higher heat release rates. With increasing combustion delays, we move to the left along the curve in Figure 3. With the combustion delay slightly higher, as for D2, D3 and G3, combustion will start with the mixture packets still rich but nearer the stoichiometric mixture strength, on average, compared to the diesel fuel. Hence these fuels will give more NO_x compared to D1. With higher combustion delays and more mixing of fuel and air, as for D4, G4 and G5, heat release will occur in mixture packets which will be much nearer the global mixture strength which is very lean. This results in very much lower NO_x . For the same reason, MPRR is lower for D4, G4 and G5 (Figure 2d). As noted before [4,5,6,7], when the overall mixture strength is lean, more premixed combustion leads to higher CO and HC so that fuels like D4, G4 and G5 show much higher HC and CO compared to fuels like D1 and D2 (Figures 2e and 2f). Further discussion, including plots of NO_x vs. combustion delay, can be found in [7] for many of the fuels considered here.

It is also worth noting from Figures 2c,2d,2e,2f that NO_x , MPRR, CO, and HC at a given CA50 are comparable for D4 and G4 and D3 and G3 which also behaved very similarly to each other in Figures 2a and 2b. Thus if the combustion phasing and combustion delay, which is a measure of the degree of mixing at the time of combustion, are matched for two fuels at a given operating condition, their combustion and emissions behaviour will also be similar regardless of volatility and composition.

Fuel D4 (or D3) is significantly heavier, less volatile and has very much higher aromatic content compared to G4 (or G3) but D4 and G4 (or D3 and G3) are very similar to each other in Figures 2a,2b,2c,2d,2e,2f. This is similar to the observation in [18] where n-heptane behaved very similarly, in terms of emissions as well as combustion parameters, to two European diesel fuels of comparable ignition delay at a given CA50 in

this type of combustion where the injection event is well separated from heat release.

3.2. EGR SWEEP AT 2000 RPM, 4 BAR IMEP, 2.0 BAR INTAKE PRESSURE, 11 CAD CA50

In this section we consider the effect of EGR on each of the fuels at 2000 RPM, 4 bar IMEP, 2.0 bar abs. intake pressure and the injection pressure set to 900 bar. This simply represents another operating condition - there is no rigid logic behind the choice of the precise operating conditions mentioned above. Qualitatively, at a higher engine speed, less time is available for autoignition and a higher intake pressure will make autoignition easier. The higher injection pressure compensates for the higher engine speed by not increasing the injection duration excessively in terms of crank angles.

EGR has to be used to control NO_x at these conditions. For each fuel, CA50 is fixed at 11 CAD ATDC, the fuelling rate is fixed to get a nominal IMEP of 4 bar without EGR, and then with fuelling rate fixed, EGR is varied. The injection timing changes as EGR is varied in order to keep the CA50 fixed. Figure 4a shows CD plotted against EGR, defined as the measured inlet CO_2 concentration expressed as a percentage of the measured exhaust CO_2 concentration. Compared to Fig. 2b, CD is much lower at low EGR levels for a given fuel at these conditions - the difference would be even more marked if CD were expressed in terms of time rather than crank angles. This is primarily because of the higher intake pressure. However, as expected, CD increases as EGR level increases. Figure 4b shows that, as expected, the normalised global air/fuel ratio, λ , decreases as EGR increases. The coefficient of variation (CoV) of IMEP is plotted against EGR in Figure 4c - the gasoline fuels show higher CoV of IMEP compared to the diesel fuels though, even amongst the diesel fuels, as DCN decreases, cycle-to-cycle variation increases. In a practical engine, injection strategies and injector design options to ensure that CoV of IMEP is not too high need to be investigated while using unconventional fuels.

From Figure 4a, it is obvious that Fuel D4 behaves like fuel G5, a 91 RON gasoline, at this condition rather than G4, 84 RON gasoline, as was seen in Figures 2a, 2b, 2c, 2d, 2e, 2f. Fuel D3 shows higher values of CD compared to Fuel G3 which it nearly matched in Figures 2a, 2b, 2c, 2d, 2e, 2f. In fact, at higher EGR D3 matches G4. Thus both D4 and D3 - high aromatic fuels - are showing greater resistance to autoignition at this condition compared to the condition discussed in Section 3.1. This is in line with previous observations about the autoignition behaviour of non-paraffinic fuels as summarised in [16].

The high aromaticity of D3 and D4 suggests that their sensitivity, $S = \text{RON} - \text{MON}$, would have been high if it had been possible to measure their RON and MON. In the diesel autoignition range, i.e. with $\text{DCN} > \sim 30$, the autoignition quality is not related to aromaticity. Even for an aromatic diesel component, the autoignition quality is determined by the long chain attached to the aromatic ring. Hence, for fuels which are very prone to autoignition, usually one measure such as the DCN is sufficient to rank them for autoignition behaviour at different engine operating conditions. Moreover, the DCN of blends of such fuels can be predicted reasonably well from the DCN of the components using linear blending rules. However, this is not the case for fuels in the gasoline autoignition range where the autoignition behaviour depends very much on the chemical structure and paraffinic fuels, for instance, behave very differently compared to aromatics as operating pressures and temperatures change. Thus if a paraffinic and a non-paraffinic fuel are matched for autoignition behaviour at one operating condition, and then the pressure is increased while holding the temperature fixed, the non-paraffinic fuel will become more resistant to autoignition compared to its paraffinic partner. The true autoignition quality is given by the Octane Index, $\text{OI} = (1-K) \cdot \text{RON} + K \cdot \text{MON} = \text{RON} - K \cdot S$, where K is a constant dependent on the pressure and temperature conditions in the engine. S is a measure of how different the autoignition behaviour of the fuel is compared to Primary Reference Fuels (PRF), mixtures of iso-octane and n-heptane, which by definition have $S = 0$. OI is the octane number of the PRF fuel which matches the test fuel at the testing condition. As the temperature for a given pressure decreases, K decreases - from a value of 1 at the MON test condition to 0, at the RON test condition and negative values "beyond" RON test conditions [16]. Thus an unleaded gasoline of 95 RON and 85 MON will match 85 PRF at the MON test condition, 95 PRF at the RON test condition and in test conditions with higher intake pressures with $K = -0.5$, will behave like 100 PRF. Compared to the conditions described in Section 3.1, with the intake pressure at 1.1 bar abs., when the intake pressure is at 2.0 bar abs., the value of K becomes more negative. Indeed, in many HCCI experiments highly negative K values e.g. -2.5 (Fig. 9 in [16]) have been measured. Hence for aromatic fuels like D3 and D4 with higher expected sensitivity, OI will increase more than for fuels with lower sensitivity i.e. such fuels will become more resistant to autoignition. In the present case, for illustration, let us assume $K = 0$ at the conditions described in section 3.1 and $K = -1.3$ at the conditions in this section. Further, let us assume that D4 has RON of 84 and MON of 70. Then, for conditions in Section 3.1, OI will be 84 for fuels D4 and G4 and they will behave like each other as observed while, with $K = -1.3$, OI will be 102.2 for D4 and 102.3 for G5 and these two fuels will match as observed in Figure 5a (OI for G4 will be 91.8).

There is also another line of evidence for such behaviour. The ignition delay, τ , measured in a premixed system, for instance

in a shock tube, at different pressures and temperatures can be described as a function of the temperature, T , and the pressure, P i.e., $\tau = f(T)P^{-n}$ [19]. It is found that the value of the exponent, n , is much larger for paraffinic fuels compared to non-paraffinic fuels (Figure 5, [19]) e.g. 1.64 for n-heptane compared to 0.83 for a toluene/ heptanes / isooctane mixture. Hence, as pressure is increased for a given temperature, the ignition delay decreases much less for non-paraffinic fuels compared to paraffinic fuels and they become relatively more resistant to autoignition.

Hence the fact that D4 (D3) matches G4 (G3) at one condition but shows more resistance to autoignition at a higher intake pressure, is in line with previous studies. These previous studies on autoignition behaviour of fuels in the gasoline autoignition range ($RON > \sim 60$, $DCN < \sim 30$) have all been done in conditions where fuel and air are fully premixed. Quite clearly similar effects regarding fuel structure effects on autoignition behaviour are also observed when the fuel and air are not completely mixed as in the current experiments. These phenomena need further study.

Figures 4d,4e,4f,4g,4h show, respectively, $ISNO_x$, smoke, ISHC, ISCO and MPRR plotted against EGR level. Without any EGR, fuels with higher CD show higher NO_x compared to Fuel D1 (Figure 4d) for reasons discussed in Section 3.1. However, at high EGR levels different fuels give similar NO_x and with around 55% EGR, NO_x can be reduced below 0.3 g/kWh. Figure 4e shows that at these higher EGR levels, smoke increases rapidly for D1 and D2 but not for D3 and D4 even though D3 and D4 have very high aromatic levels. However, the highest level of smoke measured for D1 is below 0.4 FSN at this low load condition i.e. not very high. HC (Figure 4f) and CO (Figure 4g) are significantly higher for D4 and G5, the fuels with the highest value of CD but as will be shown in the next section, lower injection pressures reduce CO and HC significantly for the volatile fuel, G5.

D4 and G5 which match each other in Figure 4a also produce similar HC, CO and MPRR at the same EGR (Figures 4f,4g, 4h). Thus, again, if the combustion phasing and combustion delay, which is a measure of the degree of mixing at the time of combustion, are matched for two fuels at a given operating condition, their combustion and emissions behaviour will also be similar regardless of volatility and composition in partially premixed combustion.

3.3. INJECTION PRESSURE EFFECTS - EGR SWEEP AT 2000 RPM, 4 BAR IMEP, 2.0 BAR INTAKE PRESSURE, 11 CAD CA50

The injection pulse width will increase and the mixing rate decrease as the injection pressure is reduced at a fixed IMEP.

Effects on emission and other parameters of such changes are discussed in this section. D1, D4 and G5 are the fuels considered. All three fuels were tested with 600 bar and 900 bar injection pressure. In addition D1 and G5 were also tested with 400 bar injection pressure. Other engine operating conditions were as described in the previous section.

A useful time constant in this type of combustion is the Ignition Dwell (IDW), the difference between the end of injection (EOI) and the start of combustion (SOC) i.e. $IDW = SOC - EOI$. If IDW is negative, combustion will be the classical diesel combustion in a lifted diffusion flame. Even if it is positive but small, the part of the fuel entering the cylinder at the end of the injection will have had little chance to mix with air before burning. In both cases, at least a part of the fuel will burn at a mixture strength rich enough to produce soot. In Figures 5a and 5b, IDW and smoke levels, respectively, are plotted against EGR for these fuels at different pressures at this operating condition. For D1, with DCN of 56, IDW becomes more negative as injection pressure is reduced and smoke increases significantly. Of course, as EGR increases, the overall mixture also becomes richer as the oxygen concentration decreases (Figure 5b) and this also contributes to the increase in smoke. IDW decreases with decreasing injection pressure for both G5 and D4 but is still high enough at the low injection pressure so that even the fuel injected later has a chance to mix well with air and there is very little smoke (Figure 5b).

Figures 5c,5d,5e,5f show, respectively, ISHC, ISCO, CoV of IMEP, MPRR and $ISNO_x$ plotted against EGR. For G5, there is a significant reduction in HC, CO and CoV of IMEP as injection pressure is decreased from 900 bar to 400 bar (Figures 5c,5d,5e). Clearly with high injection pressure and high resistance to autoignition, G5 is too well mixed, combustion occurs by autoignition in mixture packets some of which are too lean to lead to complete combustion [20,21,22]. This leads to high CO, HC and also increases cycle-to-cycle variation. With a lower injection pressure, this "overleaning" is alleviated to some degree and there is a substantial reduction in CO and HC emissions and improvement in combustion stability. MPRR increases for G5 with decreasing injection pressure (Figure 5f) as autoignition occurs in richer mixtures on average. Figure 5g shows that NO_x increases as injection pressure decreases for Fuel G5 at low EGR in line with what would be expected if the mixture packets got richer and moved up the solid line on the left-hand-side in Figure 3. Similar results were discussed for Fuel G3 in [7]. Surprisingly, similar trends are not observed for the much less volatile fuel, D4 even though it matches G5 with the injection pressure at 900 bar. As the injection pressure is reduced to 600 bar (open triangles) from 900 bar (filled triangles) for Fuel D4, CoV of IMEP increases slightly (Figure 5d) and there is a slight decrease in MPRR (Figure 5e), NO_x decreases at low EGR (Figure 5g) and there is little

change in HC or CO. It could be that the stratification increases and the heat release is more spread out as injection pressure is reduced. These trends for D4 can't be explained properly at this stage and this highlights the need to model the mixing, combustion and emission processes in detail. There is little difference in CO, HC, CoV of IMEP or MPRR as injection pressure is changed for Fuel D1 which appears to operate in conventional diesel combustion, certainly at 600 bar and 400 bar injection pressures.

In summary, it is important not to “overmix” volatile fuels which also have high resistance to autoignition. With such fuels, lower injection pressures result in better combustion stability and significantly lower HC and CO without compromising NO_x or smoke. Also the injectors and the injection system were designed for diesel fuels. There might be scope for further optimisation using larger injector hole sizes and even lower injection pressures with gasoline-like fuels.

3.4. EGR SWEEP AT 2000 RPM, 10 BAR IMEP, 2.0 BAR INTAKE PRESSURE, 11 CAD CA50

In this section we consider the effect of EGR on each of the fuels at 2000 RPM, 10 bar IMEP, 2.0 bar abs. intake pressure and the injection pressure set at 900 bar with CA50 fixed at 11 CAD ATDC.

Figures 6a, 6b and 6c show CD, IDW and global λ plotted against EGR at this condition. In Figure 6a, fuels D4 and G5 have comparable CD at this fixed CA50 while D3 lies between G3 and G4 as in Figure 4a. The only difference from the condition considered in Section 3.2 is that more fuel is injected and hence the injection duration is longer. So the overall mixture strength is richer (Figure 6c vs. Figure 4b) and IDW is smaller; IDW is negative for all conditions tested for Fuels D1 and D2.

Figures 6d,6e,6f,6g,6h,6i show respectively, ISNO_x, smoke, ISCO, ISHC, MPRR and CoV of IMEP plotted against EGR respectively. Fuels D4 and G5 which match each other for combustion delay at a given combustion phasing also match each other for all the parameters shown in these figures. Even though Fuel D4 (and to a lesser extent, Fuel D3) has very high aromatics and high density and low volatility, it can be run like the gasoline fuels G4 and G5 (to a lesser extent G3), with sufficient EGR to give ISNO_x below 0.3 g/kWh while keeping the smoke level very low (FSN < 0.1) - Figures 6d and 6e. In contrast, fuels D1 and D2, made up of components with similar boiling range as D3 and D4, result in very high levels of smoke at such EGR levels - Figure 6e. CO, HC and MPRR are higher than for D1 and D2 but at the high EGR levels required to control NO_x, not excessively so. Cyclic variations are also acceptable though it is not clear why G3

and G4 show higher CoV of IMEP compared to other fuels (Fig. 6i).

3.5. EGR SWEEP AT 3000 RPM, 10 BAR IMEP, 2.0 BAR INTAKE PRESSURE, 11 CAD CA50

In this section we consider the effect of EGR on each of the fuels at 3000 RPM, 10 bar IMEP, 2.0 bar abs. intake pressure and the injection pressure set at 1100 bar with CA50 fixed at 11 CAD ATDC.

At this higher speed, the time available for autoignition is reduced. Hence fuels which are more resistant to autoignition will have more difficulty igniting at a given pressure and temperature. Such fuels also become less tolerant to EGR which slows down the chemical reactions. Fuel D4 could not be run with more than about 25% EGR at this condition while this limit for G5 was slightly higher at 38%. Thus D4 is behaving like a fuel which is slightly more resistant to autoignition than G5 - in line with the observation in Figure 7a that at comparable EGR level, CD is higher for D4 than for G5. This difficulty with ignition can be easily overcome by increasing the temperature or the intake pressure [6]. Figures 7b and 7c show ISNO_x and smoke plotted against EGR. It can be seen that with this intake pressure, Fuel G4 can be run at high enough EGR to get ISNO_x below 0.3 g/kWh while keeping smoke very low while it is impossible to do this with D1 and D2. Again, HC, CO and MPRR are higher for the fuels with higher CD, e.g. G4 but these are not shown since the story is essentially the same as discussed in the previous section.

3.6. FUEL CONSUMPTION AND THERMAL EFFICIENCY

As mentioned in Section 2, fuel consumption is inferred from air consumption and emissions measurements. In Figures 8a, 8b,8c,8d, ISFC (Indicated Specific Fuel Consumption) and in Figures 9a,9b,9c,9d, the corresponding indicated thermal efficiency (fuel conversion efficiency/combustion efficiency), has been plotted for the four different operating conditions considered in Sections 3.1, 3.2, 3.4 and 3.5.

Increasing EGR leads to lower ISFC (Figures 8b,8c,8d) and higher indicated thermal efficiency (Figures 9b,9c,9d) for a given fuel. CA50 is fixed at 11 CAD for all these cases. Such reduction in ISFC with EGR was also noted in [11] and was attributed to reduced heat losses because of lower wall temperatures resulting from lower combustion temperatures. The difference between different fuels in Figures 8a and 9a - SOI sweep with no EGR - is small. The two lowest efficiency (highest ISFC in Figure 8a) points for Fuel G5 in Figure 9a are from very early injection which resulted in lower IMEP and higher cyclic variation. In the EGR sweep experiments,