MIXTURE PREPARATION AND COMBUSTION CHARACTERISTICS IN A COMPRESSION-IGNITION GASOLINE ENGINE

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Abstract The recent interest in controlled auto-ignition (CAI) in gasoline engines has been focused on the potential for higher efficiency, lower fuel consumption and emissions of carbon dioxide (CO₂) and the oxides of nitrogen (NOₓ). Strategies for spark-ignited (SI), controlled auto-ignition (CAI) and spark assisted, controlled auto-ignition (SA-CAI) were studied experimentally in an optical, direct injection, poppet-valve, gasoline engine. The air motion, fuel distribution and combustion performance were characterised by the application of PIV, PLIF, flame imaging, high speed photography and in-cylinder pressure data sampling. The range of engine operating conditions was determined by a parallel, firing engine study. Several strategies were employed; firstly, conventional full lift camshafts were replaced with low lift, shorter opening duration, camshafts. Secondly, negative valve overlap (NVO) timings were used to increase the retention of residual gases (EGR) and augment the in-cylinder charge temperature.

In the SA-CAI case, with NVO, the CoV in NIMEP was lower than in the SI case for the same load conditions. PIV measurements of the air motion showed lower bulk velocities and turbulence in the global flow structures in the SA-CAI case. The magnitude of the mean and RMS velocity fluctuations in the SI case at 330 °CA ATDC NF was more than 2 times greater than that of the SA-CAI case. Measurements of the homogeneity in the fuel distribution with PLIF showed that the in-cycle and cycle-to-cycle variation were significantly lower than in the SI case for the same engine load. The ignition delay angle was comparable between the two cases, however, the duration of the main burn was significantly slower in the SA-CAI case. The duration of the main burn was approximately constant at 45 °CA for all ignition timings. Furthermore, in the SA-CAI case, the flame propagation was unevenly distributed in the chamber and individual ignition sites (flame spots) were observed in the main combustion flame and in the period following early exhaust valve closure. The high level of EGR contributed to the low fluorescence intensity observed. The SA-CAI combustion was observed to follow three distinct phases: the initial spark and flame kernel development, a delay period with low flame luminosity and flame propagation with a triple flame structure and multiple ignition sites within the advancing flame front.

1. Introduction

If gasoline engines are to be used in the power trains of future passenger cars then they must become cleaner (reduce emissions of oxides of Nitrogen, NOₓ) and more efficient (reduce emissions of carbon dioxide, CO₂). The use of exhaust gases to dilute the air fuel mixture is one potential solution. A variety of systems for exhaust gas recirculation (EGR) have been developed, using external circuits [e.g. Cairns and Blaxill 2005] and phasing of the valve opening periods, such as the negative valve overlap approach, [Koopmans et al. 2003, Wilson et al. 2006]. At low engine speeds and load the level of EGR approaches 30% (mass concentration). This can have a significant effect on combustion stability in a spark ignition (SI) engine. Other factors influencing this include the location of the spark plug and low charge velocity and turbulence in its vicinity. Combustion stability can be improved by altering the phasing of the valve opening periods, timing of the fuel injection and by increasing the strength of the dominant air motions that exist in the cylinder [e.g. Lake et al. 1995]. The aim is to maintain a stoichiometric air-fuel mixture in the vicinity of the spark plug at ignition, whilst the overall air to fuel ratio is lean. However, it also compromises the volumetric efficiency of the engine at high-speeds and loads [Deschamps and Baritaud, 1996, Lake
et al. 1996 and Meyer and Heywood 1999]. Generally, such lean-burn engines are predominantly direct injection.

More recently, researchers have sought to use the high temperatures within the EGR gases to initiate compression ignition of a homogeneous mixture of air, fuel and residual gases. This has been referred to as controlled auto-ignition (CAI), or Homogeneous Charge Compression Ignition, (HCCI). This mode of operation enables the engine to be operated with a wide open throttle thus reducing pumping losses. Combustion occurs spontaneously at multiple ignition sites within the combustion chamber and is governed by the local chemical reaction rates (fuel concentration, fuel properties and temperature) and not by the propagation of a flame front from a fixed ignition point [Koopmans et al. 2003a]. This mode of combustion is less dependent upon in-cylinder turbulence to augment the rate of combustion. The principle role of turbulence is to produce a homogeneous mixture conditions. The rate of heat release is generally much greater than that of SI combustion. This can lead to engine knock and must therefore be controlled by dilution of the mixture [Shibata et al. 2004 and Thirouard et al. 2005]. The rate of combustion can also be reduced by thermal stratification [Kakuho et al. 2006]. It is generally agreed that for CAI, the stratification of temperature and air-fuel mixture with the residuals in the combustion chamber at the beginning of the compression stroke has the greatest influence upon the combustion phasing (auto ignition timing), rate of heat release and combustion stability [Koopmans et al. 2002]. The influence of air motion has less of an effect on combustion phasing than that generally reported for lean-burn spark ignition engines [Arcoumanis et al. 1997].

The range of engine speeds and loads when CAI operation can be used have been extended by using a spark to initiate combustion. In this spark-assisted controlled auto-ignition (SA-CAI) regime the main heat release timing has been shown to be insensitive to spark timing [Koopmans et al. 2003b, Santos et al. 2005 and Daw et al. 2007.]. Using a similar combustion system to this study SA-CAI was observed by Osborne et al. 2003 for 4-stroke operation. SA-CAI was demonstrated for engine speeds between 1000 and 2500 rpm and loads of 2.5 to 5.5 bar net indicated mean effective pressure (NIMEP). EGR levels of the order of 55% were used, which is far higher than that tolerated by the engine for SI operation. At a typical speed load condition (2000 rpm and 3 bar) the duration of the 10-90% mass fraction burned (MFB) was found to be 22 °CA for SA-CAI and 35 °CA for conventional SI. Li et al. 2001 and Koopmans 2002 both provide comprehensive reviews of CAI in gasoline engines.

This paper presents the results of an experimental study of CAI combustion using an optical engine. Suitable test conditions were obtained from parallel studies using a similar metal engine [Osborne 2003, 2008]. High-speed photography was used to capture the evolution of the combustion event. Planar Laser-induced fluorescence (PLIF) images of the in-cylinder fuel distribution were used to quantify homogeneity in the mixture and were combined with simultaneous flame imaging. The in-cylinder air motion during the intake and compression strokes was characterised using Particle Image Velocimetry (PIV) under motored operation. Combustion stability was calculated from in-cylinder pressure data. In section 2, the experimental approach and apparatus are described. In section 3, the results of the air motion, flame photography and simultaneous LIF measurements are presented and discussed.

2. Experimental Approach

2.1 Apparatus

A single cylinder Ricardo Hydra research engine with optical access to the combustion chamber was used for the present work. The engine was fitted with the Ricardo Flagship, top-entry, direct injection (G-DI), single cylinder head. The four valve cylinder head design has a pent-roof
combustion chamber and twin, upright, intake runners that promoted a reverse tumble air motion. The spark plug was located centrally and the fuel injector was positioned between the intake valves at an angle of 46° to the gas face. The design of the combustion system is shown in Fig. 1.

Optical access to the combustion chamber was achieved using a Bowditch-type glass piston and a quartz annulus fixed between the cylinder head gas face and the piston liner. The annulus was 20 mm in height. Two interchangeable camshafts enabled both high and low valve lift configurations to be investigated. A summary of the engine specification is given in Table 1.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>74 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>75.5 mm</td>
</tr>
<tr>
<td>Swept volume</td>
<td>324.7 cm³</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>9.2</td>
</tr>
<tr>
<td>Piston type</td>
<td>flat</td>
</tr>
<tr>
<td>Fuel injector</td>
<td>single-hole, swirl</td>
</tr>
<tr>
<td>Cone angle at 1bar</td>
<td>70°</td>
</tr>
<tr>
<td>Fuel pressure</td>
<td>100 bar</td>
</tr>
</tbody>
</table>

Table 1: Four-stroke engine specification

2.2 Test bed and installation

The engine was installed on a test bed at the Sir Harry Ricardo Laboratories in Brighton. The facility is equipped with a Lawrence Scott dynamometer and electrically driven pumps for oil, coolant and fuel supplies. The oil and coolant temperatures were maintained at 80 °C and 90 °C ± 2 °C respectively. Oil, coolant, fuel, intake air and exhaust gas temperatures were recorded using type K thermocouples. Intake manifold pressure was recorded using two, 2 bar absolute pressure transducers (Kistler 4045A2, Druck DPI 201). Exhaust manifold pressure was recorded using 2, water-cooled, 2 bar absolute pressure transducers of the same type. The intake manifold gas pressure and temperature were varied using a closed-loop feedback system that controlled the flow from a 10 bar reservoir fed by two screw-compressors, through a 3-phase heater. In-cylinder pressure was recorded using a gauge pressure transducer (Kistler 6125). Air to fuel ratio (AFR) was measured close to the exhaust port using a wide-range lambda sensor (Bosch ETAS LA3). The sensor was calibrated for the mixtures of reference fuel and fluorescent tracer used.
The rotational speed of the engine was measured using an optical encoder (Leine and Linde) with a resolution of 720ppr directly coupled to the crankshaft. The engine speed was maintained to an accuracy of ± 5 rpm. The throttle valve was driven by a geared stepper motor. The position was controlled with a multi-turn potentiometer.

The fuel rig comprised of production Bosch automotive components (tank, pressure regulator, fuel rail, and low and high-pressure pumps) integrated within a standalone unit with provision for fuel cooling and flushing. The high-pressure pump was driven by an electrical motor. The fuel injection pressure was maintained at 100 bar.

A MTS Baseline CAS data acquisition system was used to record data for combustion analysis over 500 consecutive cycles. The in-cylinder gauge pressure was pegged to the intake manifold absolute pressure conditions at 180 CA after top dead centre, non-firing (ATDC NF). Before each test run, a hot, motored TDC determination was performed to determine the thermodynamic loss angle.

2.3 Description of Optical Techniques

The Laser used in both the PLIF and PIV studies was a Spectra-Physics, double-pulse, Quanta-Ray PIV 400. The Laser was fitted with two harmonic generators that produced a 10mm diameter beam at 266 nm (PLIF) and 532 nm (PIV). The beam was formed into a 1mm thick and 40 mm wide sheet using a set of expanding and collimating lenses. The repetition rate of the Laser was adjusted to 10 Hz. The-shot-to-shot energy of the Laser was recorded. Two La Vision CCD cameras were used to capture images; a FlowMaster 3s and an Imager Intense. The pixel resolutions were 1280x1024 and 1376x1040 respectively. An La Vision image intensifier (IRO) was used to increase the PLIF image gain by a factor of 10. The gating and delay of the IRO were additionally controlled. A 105 mm UV-Nikkor lens was used for the PLIF study and a standard 50 mm Nikon lens for the PIV and flame study. Two Schott filters (one long pass and one band pass) were chosen to remove unwanted wavelengths outside of the fluorescence band (330 nm to 550 nm for 3-pentanone). A Vision Research Phantom v5.0 high-speed CMOS camera was used for the photographic study. The camera has a resolution of 1024x1024 pixels at a frame rate of 1000 fps. The synchronisation of the Lasers, cameras, image intensifier and the engine was achieved using a programmable timing unit and DaVIS 6.2 software. The timing signals were generated by a bespoke test-cell control system.

2.3.1. Combined PLIF and Flame Photography

Combined LIF/Flame imaging was conducted through the optically-accessed piston with a 45° mirror mounted below the piston window. A second dichroic mirror was used to split the image between the flame camera (La Vision Imager Intense) and the PLIF camera (La Vision FlowMaster 3S with IRO). The set-up of the Laser and cameras on the engine for combined PLIF and flame imaging is shown in Fig. 2. The location of the valves, spark plug, fuel injector and pressure transducer as viewed from above are included. A 40mm wide, 1mm thick Laser sheet entered the engine on the intake side, 2mm below the gas face and passed directly through the centre, allowing visualisation in the horizontal plane. The choice of a suitable fluorescent tracer and its volume fraction in the reference fuel used in these studies is described in detail by deSercey 2005, deSercey et al. 2002 and Parmenter 2008. The results of in-situ testing concluded that a volume ratio of 10% 3-pentanone in 90% iso-octane yielded the best compromise for fluorescence signal, calorific value and octane rating for the test conditions. The results presented in this investigation are qualitative only. The high-speed flame photography was also performed through the optical piston. The Phantom camera was used to acquire images from 30°CA BTDC (NF) to 330°CA ATDC (F) in
1°CA intervals. The resolution of the camera for a frame rate of 6 kHz was 608x600 pixels. The exposure time was 150µs.

In the PLIF study, data was acquired in the crank angle ranges of 180 °CA to 330 °CA in 10 °CA steps, with further measurements between 335 °CA and 440 °CA being taken in 5 °CA steps. Four background images were taken (2 prior to the test and 2 following the test) at each engine crank angle. These were averaged and then subtracted from the acquired PLIF and flame images. 30 individual images were averaged at each test point, for each crank angle step and for each engine condition. The first 5 were discarded due to lower Laser energy levels at the start of each acquisition scan. The image intensifier was set with a delay of 2.28µs and a width of 200ns. The optimal gain of the image intensifier was found to be approximately 8.0.

2.3.2 PIV

The PIV measurements of the in-cylinder air motion were conducted through the quartz annulus. A vertical Laser sheet was passed through the annulus in the mid-cylinder tumble plane. The PIV camera was positioned perpendicularly to the Laser sheet. The optical distortion of the annulus was corrected in the final images. The in-cylinder air flow was seeded using corn oil particles with a diameter of 1 to 2 µm generated by a series of 10 Laskin nozzles in a Scitek LS-10 seeder. The density of the seeding particles was optimised to the PIV interrogation grid size (32x32 pixels) and overlap area (50%), pulse separation (15 µs) and Laser energy over the range of crank angles investigated. PIV images were acquired in 10°CA steps throughout the entire cycle, from 30°CA ATDC (NF) to 330°CA ATDC (NF). Again, 30 image pairs were averaged with the first 5 discarded from the analysis.

2.2 Test Conditions

A range of test points for SI and SA-CAI combustion, shown in Table 2, were taken from a parallel study [Osborne et al. 2003] using a non-optical engine. The two engines used the same cylinder head, but the non-optical engine used an offset piston bowl geometry whereas the optical piston crown was flat topped. This resulted in a lower compression ratio in the optical engine (9.2 compared to 11.7). Differences in the heat transfer and piston ring leakage that affect the evolution
of the in-cylinder gas pressure and temperature make direct comparisons difficult. Higher intake manifold pressures and temperatures were used to compensate. Stoichiometric air-fuel mixtures were used to ensure the heat release and in-cylinder pressure were maintained within the safe limits of the quartz components and to reduce the affect of window fouling. With the quartz annulus replaced by a metal one, speeds of up to 2250 rpm in two stroke operation were achieved. The increased reciprocating mass of the extended piston limited fired operation to approximately 10 minutes. As a result the lower speed, spark-supported cases were selected as better suited to the optical engine since this enabled the thermal equilibrium required for CAI to be achieved. Examples of CAI at 1250 rpm for four-stroke operation are shown in Fig. 3a, b at 353 and 354 °CA ATDC NF for a mixture of 50% n-heptane and iso-octane.

<table>
<thead>
<tr>
<th>Combustion mode</th>
<th>SI</th>
<th>SI</th>
<th>SI</th>
<th>SI</th>
<th>SA-CAI</th>
<th>SA-CAI</th>
<th>SA-CAI</th>
<th>SA-CAI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Key point (KP) number</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>7</td>
<td>8</td>
</tr>
<tr>
<td>Net mean effective pressure bar</td>
<td>2.00</td>
<td>3.00</td>
<td>4.00</td>
<td>5.00</td>
<td>3.07</td>
<td>2.83</td>
<td>3.54</td>
<td>1.22</td>
</tr>
<tr>
<td>Injection duration main ms</td>
<td>1.20</td>
<td>1.54</td>
<td>1.78</td>
<td>2.02</td>
<td>1.75</td>
<td>1.75</td>
<td>1.75</td>
<td>1.75</td>
</tr>
<tr>
<td>Angle of ignition ATDC (NF) °CA</td>
<td>-23.50</td>
<td>-22.00</td>
<td>-22.00</td>
<td>-20.00</td>
<td>-20</td>
<td>-25</td>
<td>-30</td>
<td>-35</td>
</tr>
<tr>
<td>Inlet manifold absolute pressure mbar</td>
<td>405</td>
<td>527</td>
<td>627</td>
<td>732</td>
<td>1280</td>
<td>1000</td>
<td>1000</td>
<td>1000</td>
</tr>
<tr>
<td>Peak valve lift mm</td>
<td>8.3</td>
<td>8.3</td>
<td>8.3</td>
<td>8.3</td>
<td>2.3</td>
<td>2.3</td>
<td>2.3</td>
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</tr>
</tbody>
</table>

Table 2: Key test points for a stoichiometric AFR at 1000 rpm with start of fuel injection at 60 °CA ATDC (NF) from Osborne et al. 2003.

In the first instance, SI operation was performed to compare the performance of the optical engine with the firing engine of Osborne et al. 2003. The fuel and ignition timings of the engine, for the speed and load range investigated were matched. A stoichiometric air to fuel ratio (AFR) was maintained for the 3-pentanone and iso-octane mixture. A conventional camshaft was used with a peak valve lift of 8.3 mm and typical opening and closing timings. The engine was throttled. In the second phase, the well-established negative overlap (NVO) approach was adopted to increase the fraction of trapped residuals required to raise the in-cylinder temperature to that required for auto-ignition. The peak valve lift on the NVO, short lift duration camshaft was 2.3 mm. The engine was run at wide open throttle. The intake manifold pressure was increased due to the reduction in compression ratio of the optical engine. The exhaust back pressure was varied to alter the fraction of trapped residuals. The predicted level of EGR by this method was estimated to be approximately 45% at 1000 rpm.

3. Results and Discussion

3.1 In-cylinder air motion

The in-cylinder mean flow patterns during the intake and compression strokes were compared for the SI and SA-CAI configurations. As would be expected, the change in valve timings and valve lifts resulted in significantly different air motion characteristics. The results of the mean velocity measurements at approximately peak intake valve lift (IVP) are compared in Fig. 4a, b (it should be noted that the scales have been adjusted for clarity). In the SI case, the predominant reverse-tumble motion, characteristic of the combustion system, was established from approximately 60 °CA ATDC in the intake stroke. During the intake stroke, the reverse tumble motion filled the cylinder, completing a loop at approximately 190 °CA ATDC. During the end of the compression stroke, a weak reverse-tumble flow was present on the exhaust side, driven upwards by the piston as shown in Fig. 5a. Towards the end of the compressions stroke, the dominant flow was predominantly piston driven as shown in Fig. 6a. The lowest RMS velocity fluctuation was observed during this period.
In the SA-CAI case, with NVO, a secondary, counter-rotating vortex-like structure started to form on the exhaust side of the chamber from 130 °CA ATDC (Fig. 4a). The magnitude of the peak velocities at IVP were comparable to the SI case. The counter-rotating structure was seen to persist up to 180 °CA ATDC. During the compression stroke, the flow pattern differed greatly from that of the SI case. In Fig. 5b, two counter-rotating structures meet in the centre of the cylinder, opposing the motion of the piston. At 310 °CA ATDC, in Fig. 6b, the flow pattern was reversed and weakly counter-rotating structures rotated in the opposite sense. A low velocity region in the centre of the cylinder suggested that out of plane motion was occurring, indicating the breakdown of the bulk motions. Despite this effect, the magnitude of the mean velocity and RMS velocity fluctuation in the flow field for SI case at 330 °CA ATDC was over twice that of the SA-CAI case.
### 3.2 Combustion Performance of the Optical Engine

The average in-cylinder pressure traces, over 500 engine cycles, for the SI and SA-CAI cases are presented in Fig. 7a, b respectively. In the SA-CAI case, the angle of maximum pressure showed a greater variation than that of the SI case. The location of the pressure peak indicated a degree of dependency upon the spark timing with regards to the phasing of the main heat release stage of combustion. Pressure spikes, indicative of engine knock, were notably absent from the individual cycle pressure curves. In both cases, the coefficient of variation in NIMEP (CoV in NIMEP) was less than 8% which was considered very good for an optical gasoline engine at 1000 rpm. This suggested that combustion stability had been attained through thermal equilibrium after 10 minutes of continuous firing. At the 3 bar NIMEP cases, KP2, recorded a CoV in NIMEP of 6.5% opposed to 4.8% for KP5. At the highest SI load condition of 5 bar NIMEP (KP4), the CoV in NIMEP was 5.2%.

![Figure 7a: Average in-cylinder pressure trace for the SI at 1000rpm](image)

![Figure 7b: Average in-cylinder pressure trace for SA-CAI at 1000rpm](image)

The heat release rates were compared between the SI and SA-CAI cases at the 3 bar NIMEP load condition for points KP2 and KP5. In the SI case, KP2, the 10-90% mass fraction burned duration was 15 °CA and the angle of 50% mass fraction burned was 5 °CA. These closely matched the firing study of Osborne *et al.* 2003 when scaled with engine speed. In the SA-CAI case, the delay angle (ignition to 10% mass fraction burned) was comparable with that observed by Osborne *et al.* 2003 when scaled with engine speed. However, the 10-90% mass fraction burned duration (45 °CA) at KP5 was three times greater than at recorded at KP2. The angle of 50% mass fraction burned increased to 25 °CA. In all the SA-CAI cases, the burn rates were significantly slower. The main burn duration was approximately constant at 45 °CA across the entire ignition timing swing, irrespective of NIMEP. A small reduction of 5 °CA in the angle of 50% MFB was observed for the...
most retarded ignition timing. In the SA-CAI case, stable combustion was maintained for ignition timings of 35 °CA BTDC F at 1000 rpm. In the case of spark ignition, combustion stability decreased for ignition timings greater than 25 °CA BTDC F.

3.3 In-cylinder Fuel Distribution and Flame Structure

The simultaneous acquisition of fluorescence images of fuel concentration was used to evaluate the difference in the degree of homogeneity in the fuel/air mixing process, at the point of ignition, between the two cases. The greatest fluorescence signal was achieved in the spark ignition case for KP4 where the greatest quantity of fuel was injected. A selection of the averaged, simultaneous PLIF and flame images are shown in Fig. 8.

It can be seen from Fig. 8 that the intensity in the average LIF images increased through the compression stroke until approximately 340-345 °CA ATDC NF at which point the piston intersects the Laser sheet and the fluorescence signal was attenuated. However, a weak signal was observed after this point with a maximum recorded at approximately 10 °CA ATDC. For KP4, ignition occurred at 20 °CA BTDC firing and a propagating flame front was observed that filled the extents of the optical window at TDC. The flame continued to burn until approximately 410 °CA ATDC NF. The greatest flame luminosity was recorded between 365 and 370 °CA which corresponded to the secondary PLIF signal obtained due to flame emission wavelengths between 527 and 671 nm, transmitted by the tracer-LIF filter. At approximately 380 °CA, the piston cleared the Laser sheet and the subsequent images confirmed that the remaining fuel had been burned.

![Figure 8: Simultaneous PLIF and Flame Images for KP4 for °CA ATDC NF](image)

For the SA-CAI case, the PLIF images showed the same increasing trend as KP1-4 during the compression stroke. The magnitude of the fluorescence intensity was lower throughout the compression stroke than that observed during the SI cases. This was indicative of the high EGR levels present and consistent with the findings of Steeper and Han 2002 who reported that the fluorescence intensity of 3-pentanone dropped by 10 to 15% per 100 K increase in temperature. The PLIF images for KP6 are shown in Fig. 9. Flame images were not available in this case.
The degree of homogeneity in the PLIF images was predicted, after background image subtraction, by calculating the coefficient of variation (CoV) in the image mean pixel intensity for a given crank angle set, within a given cycle and from cycle to cycle. A mask was used to remove unwanted reflections. The results for KP1 and KP6 are shown in Fig. 10a, b.

The mean pixel intensity was greatest in the SI case and increased to a peak value at the point where the piston intersected the Laser sheet. The cycle to cycle CoV fluctuated about 40% throughout. For KP6, the mean intensity remained approximately constant throughout the compression stroke. The
The delay in the phasing of the main combustion event was seen in the series of images. In KP2, the visible flame filled the viewing window at 15 °CA ATDC F whereas in KP5, this occurred as late as 80 °CA ATDC. The propagation of the flame front in the SI cases was approximately equal in all directions. In the SA-CAI case, the flame front propagated as a series of uneven strands dependent upon the local mixture concentration. The observation was consistent with a triple flame structure. Within the flame front, random, individual ignition sites (flame spots) were clearly observed. These ignition sites were additionally observed following early exhaust valve closure (EVC), in the period where the piston re-compressed the hot gas mixture from 260 to 360 °CA ATDC. This indicated that compression ignition combustion occurred in the overlap period as the increase in flame luminosity was evident up to the point of inlet valve opening.

4. Conclusions

An experimental study of the air motion and combustion characteristics of an optical engine operating with both spark and compression ignition has been undertaken. The test conditions for spark ignition (SI) and spark-assisted compression ignition (SA-CAI) were determined from a parallel study, conducted using a firing, metal engine. In-cylinder pressure based analysis has been used in conjunction with PIV measurements of the air motion, PLIF of the fuel concentration and combustion photography, to compare the characteristics of each mode of operation. In the SA-CAI case, with NVO, the predicted EGR level of 45% would have resulted in flame extinction and incomplete combustion for a SI engine. However, the CoV in NIMEP has been shown to be lower than in the SI case for the same load conditions. PIV measurements of the air motion have shown lower bulk velocities and turbulence in the global flow structures in the SA-CAI case which, in conjunction with high levels of dilution, have led to a reduction in flame burning speed. The
magnitude of the mean and RMS velocity fluctuations in the SI case at 330 °CA ATDC NF was more than 2 times greater than that of the SA-CAI case. Measurements of the homogeneity in the fuel distribution with PLIF have shown that the in-cycle and cycle-to-cycle variation were significantly lower than the SI case for the same engine load. The cycle-to-cycle CoV in the PLIF image was 40% for the SI case and 5% for the SA-CAI case. The ignition delay angle has been shown to be comparable between the two cases. The duration of the main burn was significantly slower in the SA-CAI case as shown by the pressure-based analysis and the high-speed photography of the flame luminescence. The greatest variation in the angle of peak in-cylinder pressure with ignition timing has been observed in SA-CAI case. The duration of the main burn (10-90% MFB) was approximately constant at 45 °CA for all ignition timings. Stable combustion has been achieved for an ignition timing of 35 °CA BTDF at 1000 rpm. In the SA-CAI case, the flame propagation was unevenly distributed in the chamber and individual ignition sites have been observed in the main combustion flame and in the period following early exhaust valve closure. The high level of EGR contributed to the low fluorescence intensity observed for the mixture of 3-pentanone and iso-octane. The SA-CAI combustion has been observed to follow three distinct phases: the initial spark and flame kernel development, a delay period with low flame luminosity and flame propagation with a triple flame structure and the occurrence of multiple ignition sites within the advancing flame front. The main combustion phasing for SA-CAI has shown some dependence upon ignition timing, highlighting its potential as a transitional mode in a multi-mode combustion concept.

5. Acknowledgements

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