Relationship between In-cylinder Flow and Pressure and GDI Spray Propagation

By

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ABSTRACT

Fully variable valve trains allow full control of the fresh inlet charge into the engine cylinder, in terms of air mass and timing. This control can result in radical changes to the ambient conditions for direct injection sprays, even when working in a homogeneous charge configuration. This paper presents the effects on the spray morphology and penetration, for early injection into different in-cylinder conditions determined by different valve profile strategies. The engine was motored using four strategies, the standard valve profiles, late inlet valve opening, early inlet valve closing and a strategy for running the engine in controlled auto-ignition. A brief summary is given of earlier work, relevant to this present study, as regards the effects of in-cylinder pressure on spray morphology and a comparison of incylinder flow fields for two distinct valve profiles for spark and auto-ignition. The work reported in this study is aimed specifically at the in-cylinder conditions, which could be expected when a direct injection system is coupled to a fully variable valve train.

INTRODUCTION

The introduction of fully variable valve trains (FVVT) has allowed new strategies in engine control to be realised, two of the most significant being those of throttle-less combustion control and controlled auto-ignition. However, when combined with gasoline direct injection, whole new areas of engine control can be explored using conventional spark ignition, and offering some of the gains of fuel economy from lean burn operation to be obtained with homogeneous charge combustion. The FVVT systems remove the need for a throttle, as the inlet charge can be controlled by the valve events, and so minimise the throttling losses associated with a conventional throttle plate. This offers the chance to run the engine in direct injection, homogeneous mode, while still gaining the benefits of no throttle in the inlet system. The main advantage of homogeneous direct injection strategies is the avoidance of using a NOx trap and, with it, the associated extra materials costs. Additionally, there is a fuel consumption penalty with lean burn, to regenerate the NOx trap, which diminishes the theoretical benefits.

Two principal valve strategies can be employed to control the quantity of fresh charge being introduced to the cylinder, early inlet valve closing, EIVC, and late inlet valve opening, LIVO. These processes will have a direct effect on both the in-cylinder pressure, and the flow structure developed during the inlet valve opening phase. The work reported here, investigates the morphology and penetration of the spray, with early injection, 40 degrees ATDC, to gauge the effects of the in-cylinder conditions on the spray development. Four different valve timing strategies were used, standard, early inlet valve closing, EIVC, late inlet valve opening, LIVO and late inlet valve opening with early exhaust valve closing. The latter combustion mode is for controlled auto-ignition, CAI.

It is known from previous work, Kashdan et al (2002), Pitcher and Winklhofer (1998) and Wigley et al (2002), that the incylinder flow conditions play an important role in the functionality of the injector and spray development. These results will be compared to those from running the engine with the original valve profiles and a strategy known to work for controlled auto-ignition. For standard and CAI valve strategies, the full flow field has been measured using Laser Doppler Anemometry, Pitcher and Wigley (2001) and Pitcher et al (2003), and here the effects of the inlet flow on the spray development has been discussed.

INSTRUMENTATION

OPTICALLY ACCESSED ENGINE

The optical engine is a single cylinder research engine, incorporating a fused silica liner and sapphire piston crown to provide optical access, shown in Figure 1. It is based on one cylinder of a 1.8L, 4 cylinder engine and has a bore of 80.5mm and a stroke of 88.2mm. The inlet valves have a diameter of 31mm and are inclined at 68^o to the cylinder head face in the pent roof combustion chamber. The engine has been designed specifically for the application of optical diagnostic techniques. Unlike other ‘optical engines’ it has both primary and secondary balance shafts to allow for high-speed operation, up to 5000rpm. The cylinder head was based on a production design but maintained the exact combustion chamber geometry. A carbon fibre piston ring, running in the optical liner, maintained correct compression pressures. Engine timing data were provided by two optical encoders, one mounted on the crank and the other on a 2:1 drive representing the crankshaft timing.

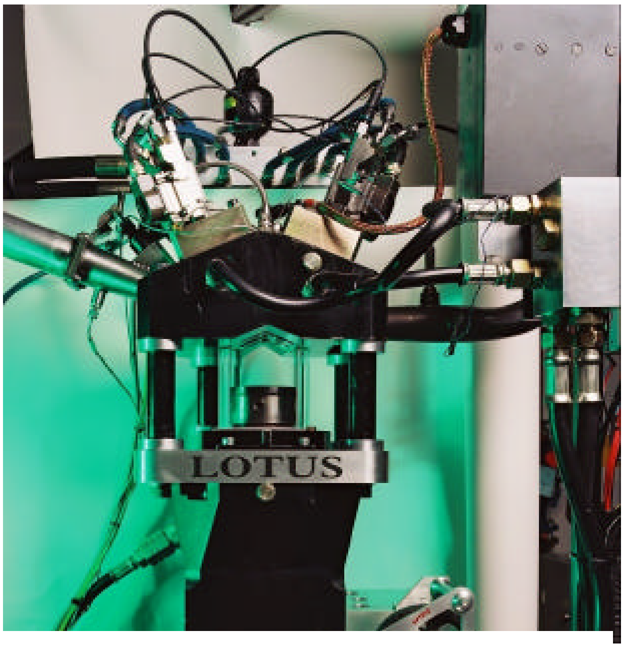


Figure 1. Optically accessed engine and AVT system

The engine was fitted with a fully variable valve system, the Lotus Active Valve Train, AVT, allowing full control of the valve profiles. The AVT consisted of a hydraulic piston attached to each engine poppet valve. Movement of this piston, and thus the engine poppet valve, was controlled by the flow of hydraulic fluid either, above or, below the piston, see Figures 2 and 3. The hydraulic flow was controlled by a high-speed servo valve. Operating at approximately 400Hz, the servo valve allows controlled valve velocities up to engine speeds of 4000rpm maintaining ‘soft-touchdown’ capabilities.

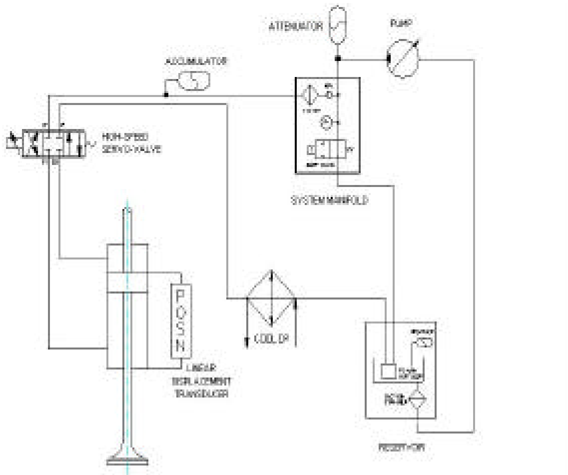


Figure 2. Schematic of hydraulic AVT system.

Positional feed-back from the piston and engine poppet valve assembly, was continually provided by fast linear displacement transducers. These allowed valve profiles to be continually monitored and corrected, from cycle-to-cycle, to ensure accuracy and repeatability. It is worth noting that non-linearities due to speed variations requires the manual input of pressure and differential gains in order to ‘fine tune’ the desired valve lift profile.

The electro-hydraulic valve actuation system has full and flexible control over valve timing, lift and velocity including precision valve closing, deemed necessary for complete control of the engine valves. The system controls each individual valve separately and can operate different lift profiles on different valves. The system is also capable of opening and closing valves more than once per engine cycle, and is limited only by hydraulic fluid delivery in terms of valve velocity and hence, operating strategy.

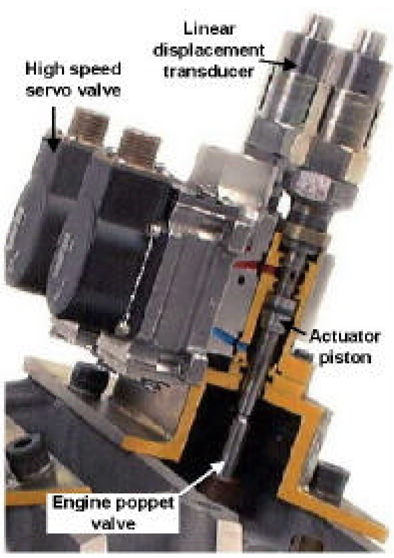


Figure 3. Sectioned model of AVT system.

Four valve timing strategies profiles were generated, with polynomial profiles, and stored in an array. This allowed dynamic changing between any stored profile with the demanded profile change occurring at TDC firing of the next cycle. The engine was motored at 1500rpm throughout for the measurements reported here. The cylinder block and head were pre-heated by circulating engine oil from a reservoir at 70^oC. The outer wall of the optical liner temperature started at 30^oC and rose to 70^oC by the end of each experiment. Essential mechanical details of the engine can be found in Tables 1 and 2.

Table 1: Single cylinder engine specification

Engine Details			
Bore	80.5mm	Capacity	0.45 l
Stroke	88.2mm	Number of valves	4
Compression Ratio	10.5:1	Maximum speed	5000 rpm
Peak compression pressure	27 bar	Maximum pressure	50 bar
Operational Engine Details – 1500 rpm – Compression Pressure			
Peak Pressure (STD)	13.5 bar	Peak Pressure (LIVO)	15 bar
Peak Pressure (CAI)	15 bar	Peak Pressure (EIVC)	9.5 bar

Table 2. Valve lift profiles and injection timing

Conventional Cam Timing		Modified Valve Timing (LIVO)	
Inlet Valve Opening	705 degrees	Inlet Valve Opening	79 degrees
Inlet Valve Closing	235 degrees	Inlet Valve Closing	211 degrees
Exhaust Valve Opening	504 degrees	Exhaust Valve Opening	504 degrees
Exhaust Valve Closing	28 degrees	Exhaust Valve Closing	28 degrees
Maximum Valve Lift	8.5 mm (Inlet)	Maximum Valve Lift	3.6 mm
Modified Valve Timing (EIVC)		Controlled Auto-ignition Valve timing	
Inlet Valve Opening	705 degrees	Inlet Valve Opening	79 degrees
Inlet Valve Closing	117 degrees	Inlet Valve Closing	211 degrees
Exhaust Valve Opening	504 degrees	Exhaust Valve Opening	509 degrees
Exhaust Valve Closing	28 degrees	Exhaust Valve Closing	641 degrees
Maximum Valve Lift	3.6 mm	Maximum Valve Lift	3.6 mm
Injection Timing	40 degrees	Injection Duration	1 ms (13 degrees)

LASER ANEMOMETER SYSTEM

The two component laser Doppler anemometer (LDA) system was described in detail by Wigley et al (1998) The only differences being the receiver optics configured for back scatter light collection and the inclusion of a two dimensional traverse system for computer controlled scanning of the measurement volume inside the glass liner. As all the measurements were made through the curved liner wall, alignment for coincidence of the two orthogonal measurement volumes would have been time consuming to achieve over the whole of the measurement mesh. Therefore, it was used as a single component system with the two velocity components being measured sequentially.

The inlet manifold was seeded with a mist of silicone oil generated by a medical nebuliser to act as light scattering centres for the LDA system. The mean droplet size was 3 to 5 microns. The LDA data were processed with a Dantec Enhanced PDA processor operated in ‘velocity only’ mode. For two component measurements, this processor demands near perfect coincidence for the two signals and, as that could not be guaranteed without considerable effort, was the main reason for performing sequential single component measurements.

The number of single component, velocity samples acquired at each point in a measurement plane varied from 30000 close to the head, down to 15000 close to the cylinder bottom. Seeding levels were kept relatively sparse, to minimise window soiling and to ensure that data were collected over a sufficient number of engine cycles. Therefore, data arrival rates were generally low and did not allow a cycle resolved analysis of the flow data. However, the flow structures were well ordered and exhibited small cycle to cycle variations, particularly with the CAI cam profiles, Pitcher et al (2003).

IMAGING SYSTEM

A diffuse back-lit scheme is normally the preferred lighting method for characterising spray morphology. However, lighting through the optical liner of the engine can introduce severe reflections on to the spray images. The solution was to illuminate up through the sapphire window in the piston crown. An EG&G MVS 7020 Xenon flash unit was coupled to a Fostec fibre optic cable to provide a diffuse light intensity distribution over the incylinder volume. The single-shot images were digitally recorded with a PCO Sensicam Fast Shutter CCD camera. It provided an image size of 100 mm by 80 mm represented by 1280 by 1024 pixels with an intensity level resolution of 12 bits. The flash duration was approximately 8 µs but the camera shutter timing was set to peak flash intensity with a shutter speed of 1 µs. An AVL 4210 Instrument Controller, with inputs from the optical encoders, provided drive signals for the injector solenoid relative to piston TDC and an electronic trigger for initiating the imaging system delay timer unit. This provided a variable delay trigger, to control both the flash and image capture time, that was programmed to increment the delay time sequentially while the engine was running. The result was a high resolution sequence of 77 images covering the time history from 0 to 1.6 ms of the spray development during injection over consecutive engine cycles. As the maximum frame rate of the camera was 8 Hz at full pixel resolution the camera and spray was activated every other cycle.

RESULTS AND DISCUSSION

IMAGING STUDY

The images shown in Figures 4 and 5, have been taken 1.4 ms after the injector trigger signal at 40 degrees crankangle. The injection duration was set to 1 ms; when combined with the delay of 0.4 ms between the trigger and the first appearance of fuel at the injector orifice, then the image time corresponds to the fuel spray just prior to needle closing. The images therefore represent the full development of the spray cone. The crankangle position corresponding to this image time is 53 degrees. There is dark horizontal line across the spray cone which represents the axial location of the top of the piston ring at TDC. It is incylinder debris, collected by the ring on the upward stroke and deposited there during the downward stroke. The distance from the injector orifice to the ring at TDC and the piston crown is 21 and 37 mm respectively.

The spray images for the standard and EIVC valve lift profiles, Figures 4(a) and 4(b) respectively, show:- (1) there is a build up of fuel spray above the center of the piston crown, due to the impingement of the initial pre-swirl part of the spray, (2) the spray cones are about to impinge on the piston crown, (3) large drops are clearly visible in the leading edge of the spray cone and (4) that both display the same nominal cone angle of 65 degrees. This cone angle equals that measured under atmospheric conditions in the near nozzle region, Wigley et al 2002/3.

Apart from some variation in intensity levels no discernable differences can be detected in the spray development between these two images, even though the inlet valve lift for EIVC is much reduced compared with that for the standard lift seen in Figure 4(a). However, at this engine speed of 1500 rpm, this low lift is adequate for air inlet and therefore the pressure conditions are the same. The difference in valve lift strategy will be reflected in reduced peak in-cylinder pressures and flow velocities. These variations in incylinder conditions do not appear to have a major impact on the fuel spray, during the main phase of the injection period.

With the CAI valve lift profile the start of injection occurs before the inlet valve opens and, since the exhaust valve was closed, early the spray is injected into a compressed charge. The in-cylinder pressure at the time of the injection trigger is 2.5 bar, dropping to 1.5 bar at the end of the injection period. As there is a 0.4 ms delay between the trigger and fuel appearing at the nozzle, it is estimated that the spray is injected into a back pressure of 2.0 bar. The effect of this increased pressure can be seen in the reduced penetration of the fuel spray, with a slight reduction in cone angle to 61.5 degrees, Figure 5c. Previous work, Pitcher and Winklhofer (1998) and Wigley et al (2002), has shown that the cone angle would decrease significantly with greater in-cylinder pressures.

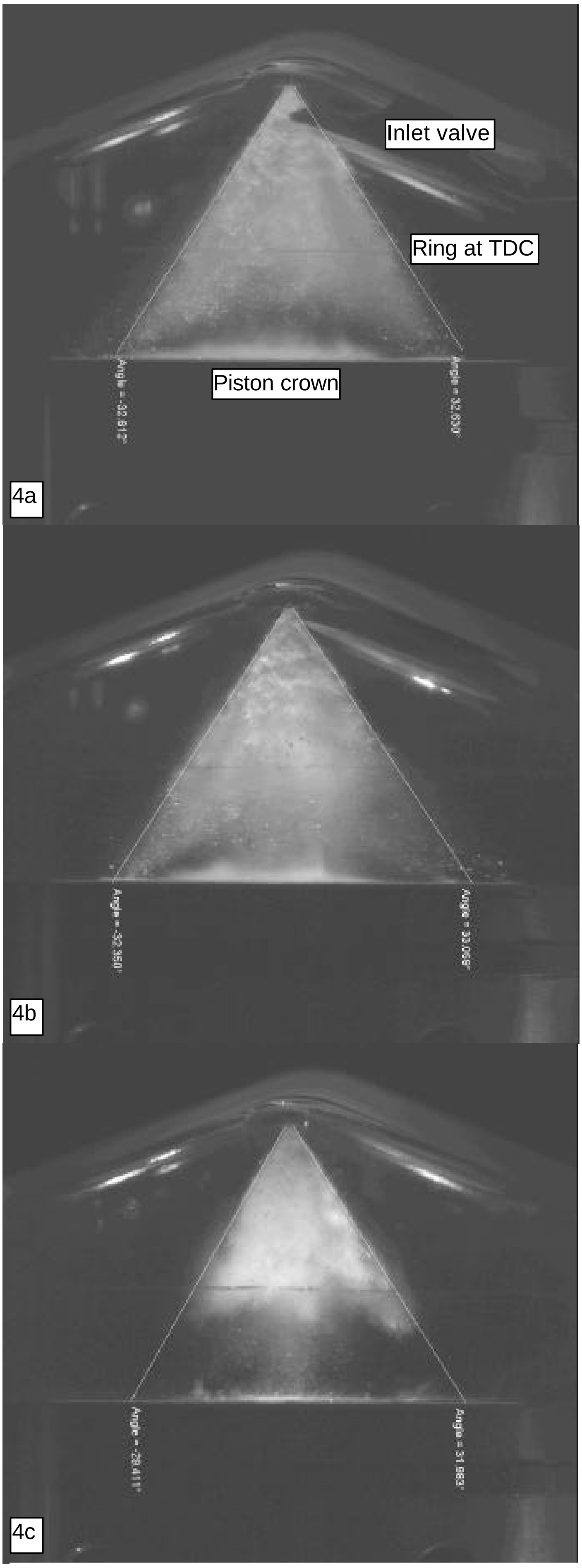


Figure 4 Grey scale spray images for (a) standard (b) EIVC and (c) CAI valve timings at 53 degrees crankangle

Spray images obtained for the LIVO injection strategy had a totally different character to those for the other three strategies. The spray appeared as a mist; the spray cone boundary was indistinct, the fuel was better dispersed and no large droplets were to be seen. Since the inlet valve did not open during injection the fuel was injected into a partial vacuum with the incylinder pressure at approximately 0.1 bar. This leads to flash evaporation and greatly enhances fuel spray penetration and atomisation.

It is difficult to estimate spray characteristics from the grey scale image, Figure 5(a), although a cone angle of 65 degrees has been superimposed on the image for comparison with the spray images in Figure 4. Step intensity contour levels of 15% have been employed to highlight the less dense parts of the spray structure, Figure 5(b). There is evidence to suggest that the spray cone has impacted the piston crown and starts to generate a recirculation zone at the cylinder wall – piston interface. Of note is the asymmetry of the spray with the spray boundary extending up the cylinder wall on the right towards the inlet valve. With the cooling effect of the spray there may well be some condensation present from the water vapour contained in the inlet air flow. Step intensity contour levels of 15% have also been calculated for the standard valve timing profile at 74 degrees ATDC. This highlights the spray density after injection has finished, it will be discussed later in relationship to the incylinder flow fields.

FLOW FIELDS

Cycle averaged time history mean and RMS velocity profiles for the axial and radial velocity components were made in three axial planes in the cylinder throughout the inlet and compression stroke. The measurement planes were on:-
(1) the plane of symmetry on the diameter between the inlet valves, (2) the orthogonal diameter parallel to the pent roof of the combustion chamber and (3) on a chord through the centre of one inlet and exhaust, Pitcher et al (2003).

For each plane the measurements started at 10mm below the head face and 5mm from the cylinder inner wall. The axial scan was in 10mm increments and the horizontal scan in 5mm increments. The crank angle resolved measurements were obtained between top dead centre valve overlap, and top dead centre compression. The optical encoder, 3600 pulses per revolution, attached to the cam gave a resolution of 0.2 crank degrees for the arrival time history of the discrete velocity data samples. The time varying mean velocity data were calculated using 4 degree crankangle time bins.

Only the flow fields for the symmetry plane and the plane under a valve pair for 74 degrees crankangle are presented in Figure 7. The flow fields for Figure 7(a) and 7(b) are for the standard valve timing while those in Figure 7(c) were obtained for the CAI valve timing. For the 74 degrees crankangle the maximum tumble ratio for the standard valve timing was obtained.

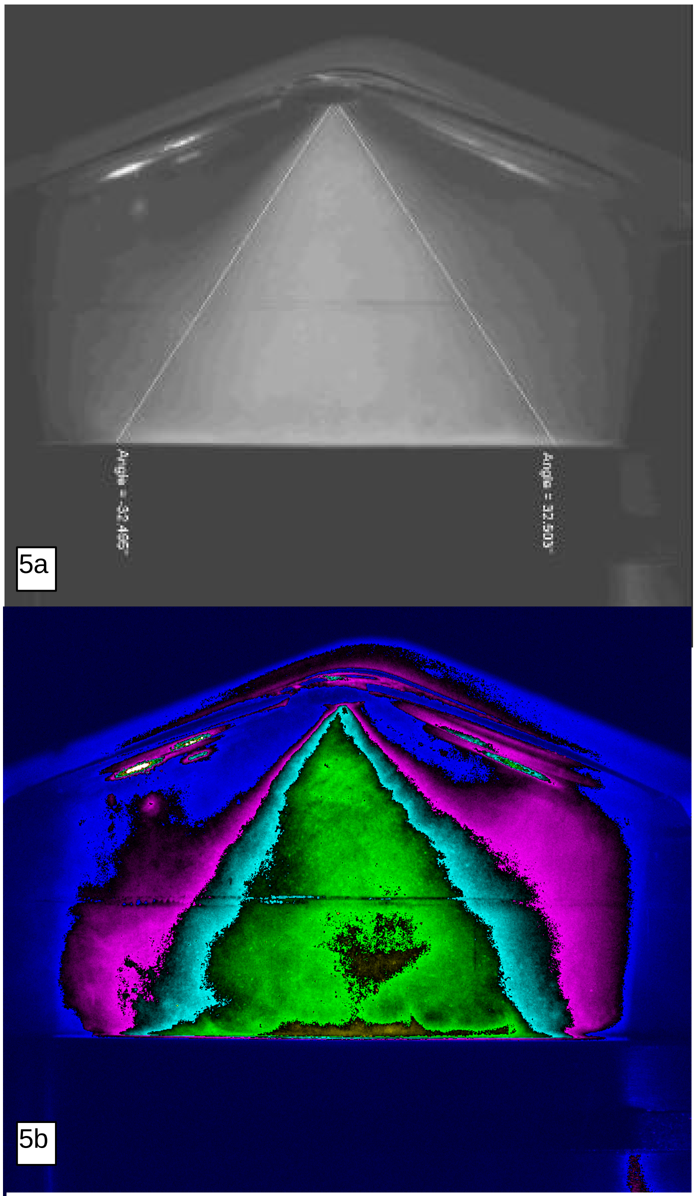


Figure 5 Spray Images (a) grey scale and (b) intensity contours for LIVO valve timing at 53 degrees crankangle

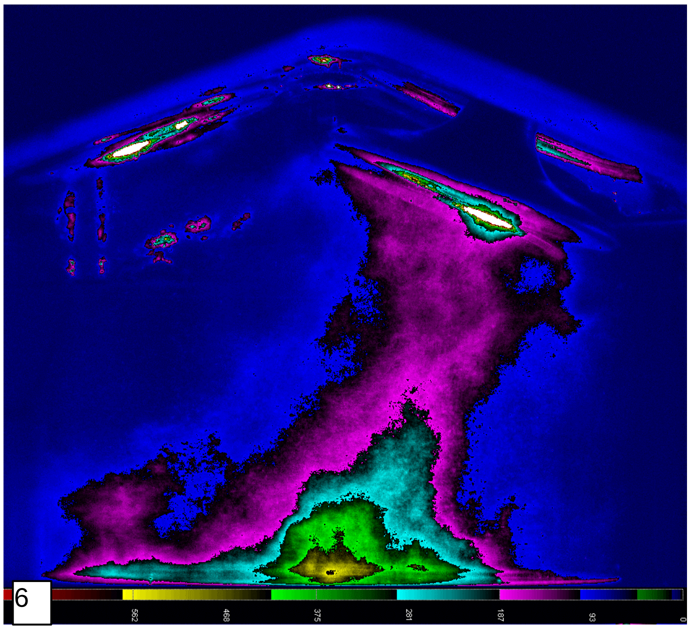
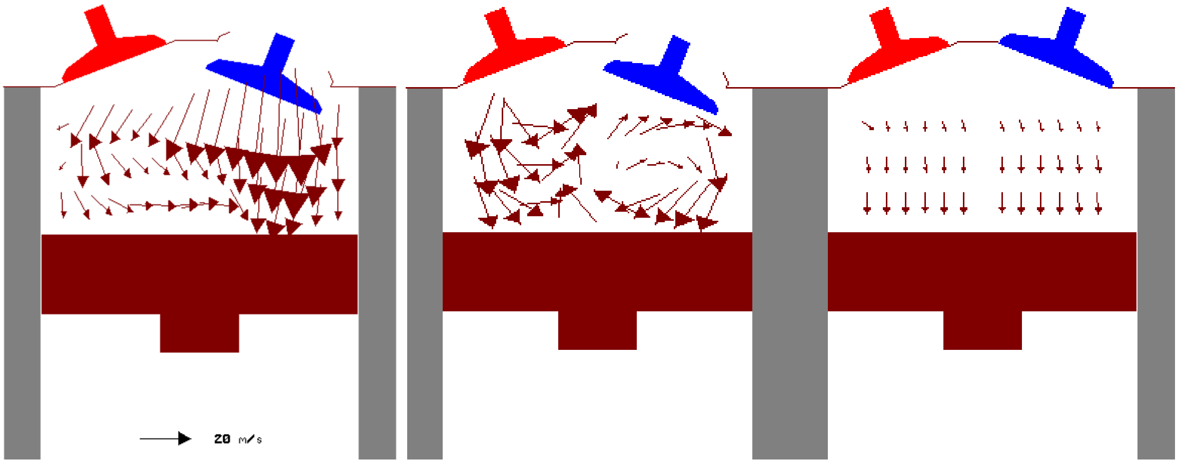


Figure 6 Spray image intensity contours for standard valve timing at 74 degrees crankangle



(a) Symmetry plane (b) Under a valve pair (c) Under a valve pair

Figure 7 Flow fields at 74 degrees crankangle for standard valve timing (a) and (b) and CAI valve timing (c)

These flow structures originate as early as 50 degrees crankangle, the flow structures at 74 degrees crankangle are geometrically similar and appears to be simply stretched by the piston travel. The symmetry plane is the image plane for the camera so that the spray cone angle should be most sharply focussed. It is seen that the flow field in this plane, Figure 7(a), is dominated by a strong axial flow, of up to 60 m/s, between the inlet valves. It does not follow the inlet port direction as the flow on the other side of the liner is directed down and then across the top of the piston to force the axial inlet jet nearly vertically downwards. However, this flow structure is in stark contrast to the flow in the vertical plane under an inlet – exhaust valve pair, Figure 7(b), which is 18 mm offset from the symmetry plane. There is a low pressure zone under the inlet valve which gives rise to the clockwise recirculation zone and an anti-clockwise recirculation zone under the exhaust valves.

The incylinder flow field structures shown in Figures 7(a) and (b) are geometrically similar to those existing for the spray shown in Figures 4(a) and (b) respectively. Considering that the spray images appear symmetrical about the cylinder axis then the conclusion is that the spray cone development, for early injection, can be considered independent of the flow field structure. However, this is not the case for times after the end of injection where the flow structure plays a definite role in determining the spray distribution within the engine cylinder. This is highlighted by considering the spray image in Figure 6 and the flow pattern in Figure 7(b). The intensity level contours show a plume of fuel spray reaching up from the piston crown to the inlet valve. The recirculation zone generated under the open inlet valve has caused the fuel spray to be held there. Of course, with this simple diffuse illumination method the spray images are an integration across the depth of the spray. Knowing the flow field structure justifies the above interpretation but planar light imaging will be needed to confirm this.

The flow field produced by the CAI valve timing is totally piston dominated, producing a uniform downward flow through out the engine cylinder, Figure 7(c). The flow field for the LIVO valve timing has not yet been measured, but, as the inlet valve has still not opened, it is expected to have the same piston dominated flow structure as for the CAI case. The main difference between the two timings though is the incylinder pressure, 2 bar and 0.1 bar for the CAI and LIVO cases respectively. It is the incylinder pressure that appears to have the major influence on the early spray development. The cone angle shows little variation with the different valve timing strategies whereas the spray penetration shows consistent variations for both the pre-swirl and main cone penetration. The penetration time for the pre-swirl and the main cone to reach the ring representing the piston ring position at TDC, i.e. 21 mm below the injector orifice, are listed in Table 3 with the time given in milliseconds and degrees crankangle.

Table 3 Spray penetration time between injector orifice and piston ring at TDC

	Time of Flight ms - degrees crankangle			
Valve Timing	Standard	EIVC	CAI	LIVO
Pre Swirl	0.27 - 2.5	0.26 - 2.4	0.28 - 2.6	0.25 - 2.3
2nd cone	0.59 - 5.4	0.62 - 5.7	0.75 - 6.9	0.55 - 5.0

The lowest incylinder pressure case is with the LIVO valve lift profile and, as expected, produces the lowest transit times. The converse is true for the CAI valve timing. To all intents and purposes the spray penetration times for the standard and EIVC valve lift profiles are equal.

CONCLUSIONS

A unique automotive combustion research facility which combines a fully optically accessed single cylinder engine with a combustion chamber geometry and engine speeds representative of a production engine with a fully variable valve train system has been briefly described. The emphasis of the work planned is aimed at evaluating future combustion strategies for homogeneous gasoline direct injection, under both full and part load operating conditions, and controlled auto-ignition. The work presented here is an initial assessment of the incylinder air flow structures, the fuel spray dynamics and the subsequent air-fuel mixing processes. Simple imaging analysis has been performed for four inlet valve profiles representing, standard, early inlet valve closing, late inlet valve opening and controlled auto-ignition. The flow fields have been measured for the standard and CAI valve profiles but those for EIVC and LIVO are expected to be geometrically similar. Initial analysis of the spray images shows that the spray dynamics during injection is not sensitive to flow structure but is certainly a controlling factor for spray distribution after injection. The incylinder pressure is seen to have the most important effect on early spray propogation. For the LIVO valve timing profiles flash evaporation of the fuel plays a significant role in determining spray shape and penetration even during the injection period.

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