Effects of intake swirl and coolant temperature on spray structure of a high pressure multi-hole injector in a direct-injection gasoline engine

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Abstract. The spray characteristics of a 6-hole injector were examined in a single cylinder optical direct injection spark ignition engine. The effects of injection timing, in-cylinder charge motion, fuel injection pressure and coolant temperature were investigated using the 2-dimensional Mie scattering technique. It was confirmed that the in-cylinder charge motion played a major role in the fuel spray distribution during the induction stroke while injection timing had to be carefully considered at high injection pressures during the compression stroke to prevent spray impingement on the piston. A new approach has been applied to the processing of Mie images to analyse the effect of coolant temperature on the spray liquid phase structure. These results quantified the fraction of the liquid phase that was vaporised as a result of the increase in coolant temperature.

1. Introduction

Direct-injection spark ignition (DISI) engines offer the best promise for simultaneous reduction of fuel consumption and exhaust emissions in gasoline engines. Several DISI engine models have emerged into the international market and have highlighted the potential benefits on both fuel economy and pollutant reduction. Most of these engines are based on the wall-guided combustion design concept [1-2]. These “first generation” injection systems, with swirl pressure atomisers have proved to offer lower fuel consumption of up to 20% in the case of stratified, overall-lean part-load operation, but not significant improvements in HC and NOx emissions [3]. The key success in DISI engines is in preparing the right amount of stratified fuel mixture under part-load operation when the fuel is injected late during the compression stroke; the goal is to quickly transport the fuel/air mixture towards the spark plug with no impingement on surfaces and to achieve complete evaporation of the droplets in the short time available between the end of injection and the start of ignition.

Most recent researchers have focused on an alternative strategy to the wall- and air-guided mode of mixture preparation for producing stratified fuel mixtures, the so-called spray-guided using a new generation fuel injection system with either central or side fuel injection [4]. The major advantage of this configuration is that it makes use of the injection process to ensure that a stable combustible mixture reaches the spark plug at the time of ignition which, in turn, depends strongly on the spray characteristics and, in particular, its cycle-to-cycle stability which otherwise may even cause a misfire. Thus, to utilize the full benefit of DISI technology, knowledge of the temporal evolution of the spray structure, its tip penetration and the distribution of the droplet velocities and diameters as a function of nozzle design, injection and chamber pressures, is a prerequisite. It should be mentioned that the swirl pressure atomiser (first generation) was found to be unsuitable for the new concept of mixture preparation due to the demonstrated spray cone angle instability with back pressure, leading to a complete collapse of the spray structure when fuel was injected during the compression stroke [5-6].

Recently, a number of injector manufacturers have designed new high-pressure multi-hole injectors and outwards opening piezo injectors, referred to as ‘second-generation’ systems, on the expectation that they produce stable fuel sprays with fine fuel droplets independent of the time of fuel injection [4]. Multi-hole injectors have been studied due to their potential of achieving good fuel stratification, thus being able to extend the lean limit further [7]. They also offer the highest possible flexibility in adapting the spray pattern layout to a
particular combustion chamber design. The investigations of [8-11] on multi-hole injectors for gasoline engines confirmed the improved stability of the spray at elevated chamber pressures relative to that of swirl injectors. Also, enhanced air entrainment has been observed due to the separated spray jets producing larger surface area, with enhanced flexibility to direct the sprays towards the proximity of the spark plug and improved matching between the injector, the generated spray and the combustion chamber design. Recently some detailed experimental investigations have been carried out and reported in [10-12] on the gasoline spray characteristics and mixture distribution in engines equipped with high-pressure multi-hole injectors at injection pressures up to 200 bar and back pressures up to 12 bar. These studies confirmed that the overall spray angle relative to the axis of the injector is independent of injection and chamber pressure. The effects of injection and chamber pressure on droplet velocities and diameter were also quantified. For late fuel injection during the compression stroke aiming at stratified overall lean mixtures, the elevated in-cylinder gas pressure/density reduces spray penetration and produces a more compact spray that can more easily be directed towards the spark plug. In addition, the investigations of [13-14] identified the complex nature of the in-nozzle flow and, in particular, the development of different types of cavitation that can influence the stability of the emerging jet sprays.

Mixture preparation in direct injection engines is one of the most important processes in ensuring a successful DISI combustion system [15]. Preparing the desired mixture inside the combustion chamber over the full range of engine operating conditions is quite difficult, as the fuel/air mixing process is influenced by many time dependant variables. In this study, the spray characteristics generated by a high pressure multi-hole injector have been examined as a function of injection timing, in-cylinder air charge motion, coolant temperature and injection pressure using the Mie scattering technique. The engine configuration and experimental techniques for the present experiments are described in the following section, the results are presented and discussed in section 3, and the paper ends with a summary of the most important findings.

2. Experimental set-up
2.1 Engine design
The single cylinder research engine used in this study, was designed for optical measurements and, as such, it offers good optical access, with its 4-valve modern pentroof cylinder head designed to allow both wall-guided and spray-guided operations.

The optical engine set up is shown in Figure 1 (a-d), and the engine configuration details are summarized in Table 1. As shown in Figure 1a, downstream of the throttle valve there is a second valve installed at the inlet of one of the ports, named Swirl Control Valve (SCV). This valve, when closed, blocks the air flow through this port redirecting the whole intake air flow to the second intake valve. In this way, in-cylinder swirl is generated. The position of this valve can be varied manually from fully open to close using an external gauge controller. Without the swirl control valve, this cylinder head was designed to generate high tumble flows; and the TVRo (steady flow tumbling vortex) values measured were 1.38 and 1.7 for 1000 and 1500 rpm respectively, in a steady flow rig test [8]. From this point onwards, “SCV open” means in-cylinder air motion is mainly tumble while “SCV closed” means swirl is the dominant fluid motion.

![Figure 1](image-url)
Optical access to the combustion chamber was provided from the side (vertical images) through a fused silica cylinder liner (Figure 1b and 1c). As can be seen in Figure 1b, there are also two quartz windows fixed on both sides of the cylinder head to provide access to the pentroof area. Between the cylinder head and the optical liner there is a cylinder gasket, which is sometimes seen in the images as a dark strip. The piston crown has a flat design so that an optical window can be fitted to obtain horizontal images. The orientation of the injector and the spark plug is in the longitudinal arrangement shown in Figure 1d; the line of the spark plug and the injector is in the middle of the pentroof between the intake and the exhaust valves. In such an arrangement, swirl motion of the air will have a direct impact on the fuel spray injected during the induction stroke. Due to the emphasis on mixture preparation, all tests were carried out without combustion where the engine was motored by a regenerative 21 kW DC dynamometer.

### Table 1: Engine specifications

| Specification                  | Details |
|-------------------------------|---------|
| Combustion chamber            | 4-valve, pent-roof |
| Cylinder head ports           | Angle between valves, 45° |
| Bore x Stroke (mm)            | 83 x 92 |
| Displacement (cc)             | 498     |
| Compression ratio             | 10.5    |
| Intake valve open/close       | 6° BTDC/50° ABDC |
| Exhaust valve open/close      | 50° BBDC/6° ATDC |

Identification of the engine cycle and crank angle position was achieved by an optical pick up sensor mounted on the exhaust camshaft and a crankshaft encoder (Muirhead Vactric) which produced 1440 pulses per revolution, thus resulting in a resolution of 0.25 crank angle degrees (CA). Engine control was achieved by using an advanced timer card (NI PCI-6602) with an in-house software (Labview), which controlled injection and ignition as well as provided two additional general purpose triggers. The timer card also synchronised a data acquisition card (NI PCI-6024E) for engine diagnostics.

The injector used in the present experiments is a high pressure six hole injector having the 6 holes symmetrically arranged on the periphery of an imaginary circle. It produces 6 jet sprays and operates with injection pressures up to 200 bar. The fuel flow circuit comprises a fuel tank, low fuel pressure electrical pump (12 volts), a high fuel pressure pump, a common rail system equipped with a pressure regulator and the injector. High injection pressure is essential for stratified spray guided DISI engines because more effective spray atomisation is required to deliver the required air-fuel mixture within the smaller time/space scales relative to those in wall guided DISI engines.

#### 2.2 Mie scattering system

The optical set-up depicted in Figure 2 was used for capturing Mie scattering images. The illumination of the spray was achieved by means of a xenon flash light directed via a couple of optical fibres to the area of interest. Qualitative and quantitative information of the spray was extracted from high-resolution forward illuminated images recorded with a non-intensified 12bit CCD PCO SensiCam camera, offering a resolution of 1024x1240 pixels and low readout noise, in conjunction with a Nikkor telescopic zoom lens (75-300mm 1/4.5-5.6). Imaging timing is controlled by the engine control system which has equipped with two general purpose triggers. An activating cycle frequency of the image capturing and fuel injection was set in such a way to allow sufficient time (15s) for the xenon flash light to recharge fully. Both the camera and the xenon light were triggered by the same TTL pulse from the PC timer card at specified time after start of injection (ASOI) which is referenced from the rising edge of the injection signal to the injector driver during activating cycle.

Figure 2 Mie scattering set-up for the in-cylinder fuel spray visualisation.
To explore the spray pattern of a high pressure 6-hole injector, a variety of different operating modes and conditions were tested as shown in Table 2. The injection duration was kept at 1 ms for all the test conditions. Spray imaging was repeated 3 times for each time step of each test case. The processing of the acquired images was performed by a Matlab program which automatically identified the spray cone angle and penetration.

The spray cone angle and penetration that have been obtained from the Mie images regarding the multi-hole injector are defined and provided in Figure 3; images taken in A-A plane view was used to obtain the spray cone angle. For the investigation of in-cylinder spray characteristics, the injected sprays of 6-hole injector were visualized at engine speed of 1000 rpm. There is a 0.2 ms delay between the injection trigger signal (the electronic SOI) and the first appearance of spray out of the injector (the optical SOI). The injection delay time due to the needle opening was similar to that of 0.18ms found by [10]. In the present investigation, spray evolution images were captured from 0.3ms ASOI to 1.1ms ASOI by the time interval of 0.1ms.

| Table 2 Experimental cases |
|-----------------------------|
| Engine speed            | 1000 rpm |
| Throttle valve          | Wide open |
| SCV                     | Open/Closed |
| Coolant temperature    | 40ºC/90ºC |
| Int. air temperature   | ~20ºC |
| Injector               | 6 hole multi-hole |
| Injection pressure     | 70bar/120bar |
| Injection strategy     | Single |
| Fuel                   | 100% isoctane |
| Oper. mode             | Homogeneous/Stratified |

3 Results and discussion

3.1 Early injection for homogeneous stoichiometric operation

Multi-hole injectors are known to have stable spray structure under various operating conditions [10-18]. Overall spray cone angle remains close to the nominal design value with increasing chamber pressure; thus early and late injection during an engine’s cycle appear to have almost identical spray shape, affecting only spray’s penetration in the combustion chamber. Therefore, injection timing mainly controls wall impingement. Known advantages of stoichiometric homogeneous operation of DI engines over their conventional PFI counterparts, such as increased volumetric efficiency and reduction of charge temperature, create the need of exploring the potential of multi-hole injectors in generating a well homogenised, stoichiometric mixture. Homogeneous operation dictates early injection of the fuel during the induction stroke. A selection of early injection timing includes injection of fuel at 60° CA, 90° CA and 120° CA after induction TDC.

From the previous LDV measurement of in-cylinder flow under ‘SCV open’ or tumble flow condition [16], it was realised that high velocities were generated during the intake process, rising to a maximum between 60ºCA and 120ºCA and then decreasing in response to the piston motion. During this period, the incoming high velocity annular air-jet flows were directed axially towards the down-going piston and radially towards the exhaust. The results also showed that the generated swirl flow was not strong and not well defined with respect to cylinder axis. At an engine speed of 1000 rpm, the axial and the swirl mean velocities were measured to be ~ 6Vp and -0.9−1.5Vp, respectively, in the centre of the cylinder at axial distances of 10.25mm and 12.25mm.
below the TDC. And corresponding RMS values were ~3Vp and 2.7~2.2Vp. From these results of in-cylinder flow measurement and those reported by [19], it can be expected that the injected spray pattern during the intake stroke with ‘SCV open’ can be strongly affected by the tumble motion and its variation will result from the turbulence of the swirl motion.

Evolution of spray pattern at different injection timing of 60°CA, 90°CA and 120°CA after induction TDC, with the SCV fully closed (maximum swirl), are displayed in Figure 4. As can be seen, there are two distinct features in the spray structures, one is that the multiple spray plumes (jets) from the multi-hole nozzle can not be discriminated and the second is a clear tilt of the overall spray towards the exhaust side and down the same as that of the incoming annular air jet trajectory. The merging or smearing of the spray plumes is taking place as soon as the fuel plumes are generated from the nozzle. This is because the plumes are subjected to a strong intake flow with high tumble and swirl velocities, and high turbulence. As a result the smaller and slower droplets will be dispersed rapidly under highly turbulent and swirling flow causing the separated injected fuel plumes to smear together. It is also clear from the images that the tilt of the overall spray is in the direction of the intake cross-flow and in accordance with the results of [19]. These effects are more evident when the elapsed time goes over 0.7ms ASOI, the whole spray is now inclined towards the downstream and further more the fuel droplets of tip edge start to be separated from the main plumes jet towards the cross-flow direction; the latter effect can be due to high swirling and turbulence. The extent of the separation increase according to the elapsed time after start of injection and those of 90°CA and 120°CA SOI are more pronounced than that of 60°CA. at 0.9-1.1ms ASOI, the spray tilt is even more recognisable with downstream injected fuel droplets are largely distributed in the cross-flow direction. This phenomenon represents the promotion of the injected fuel distribution through the combustion chamber. The swirl flow activates the spatial advantage of multi-hole nozzle to accommodate the homogeneous charge mixture. At the elapsed time of 1.1ms, small portion of the separated fuel droplets reaches to the cylinder wall which is undesirable. But the period that the piston returns to the same position of the compression stroke will be over 30ms and it would be enough to vaporise.

| SOI (°CA) | 0.5ms ASOI (Δθ=3°CA) | 0.7ms ASOI (Δθ=4.2°CA) | 0.9ms ASOI (Δθ=5.4°CA) | 1.1ms ASOI (Δθ=6.6°CA) |
|----------|----------------------|----------------------|----------------------|----------------------|
| 60       | ![](image1)          | ![](image2)          | ![](image3)          | ![](image4)          |
| 90       | ![](image5)          | ![](image6)          | ![](image7)          | ![](image8)          |
| 120      | ![](image9)          | ![](image10)         | ![](image11)         | ![](image12)         |

**Figure 4** Instantaneous Mie images during the induction stroke with swirl flow (B-B plane view); SCV is fully closed.

The spray evolution when the SCV fully open (tumble, i.e. little or no swirl) at start of injection of 60°CA, 90°CA and 120°CA after induction TDC are displayed in Figure 5. Like the spray pattern under the swirl flow, the multiple spray plumes can not be distinguished for the same reasons. The injected fuel spray plumes can’t avoid the strong influence of the incoming air cross-flow during the intake valve open due to the injector position in the cylinder head. Generally the tumble flow does not deflect the spray pattern as strong as that with swirl flow and that there is not any fuel droplets separation phenomenon; the latter suggests no impingement on the liner. The larger spray deflection and the droplets separation with swirl flow, as seen in Figure 4, clearly suggest the presence of centrifugal force acting on fuel droplets away from the centre of cylinder. Overall comparison with the spray patterns under the swirl flow of Figure 4 indicates that the spatial distribution of the
injected fuel spray under the tumble flow is apparently less than that of swirl flow especially from the elapsed time of 0.7ms. Therefore for a well distributed, homogenised and stoichiometric mixture, it is confirmed that the swirl flow is more important to be generated in cylinder than the tumble flow. Comparison with other spray patterns at different start of injection shows that the spray pattern at 90°CA after induction TDC is more affected by the tumble cross-flow.

3.2 Late injection for stratified lean operation

The concept of stratification needs to be clarified according to engine design. At the time of ignition, an ignitable mixture cloud should be around the vicinity of the spark plug. This mixture cloud could be slightly rich in fuel locally, while the remaining volume of the combustion chamber is occupied by air. As the result the overall AFR is lean in fuel (typical lean AFR values are ~30-50). The size of the mixture cloud increases with increasing engine load and the load is controlled quantitatively by the amount of fuel injection.

The most common technique to achieve mixture stratification is by injecting the fuel during the compression stroke and after the closure of the inlet valve. In this study, three injection timings during the compression stroke have been selected 270°CA, 285°CA and 300°CA which were defined as medium and late injection timings [12]. The previous study [16] showed that during 260°CA and 300°CA, the axial and the swirl mean velocities were measured 2.3~1.7Vp and -0.5~0.2Vp respectively at the centre of 10.25mm axial plane and 12.25mm axial plane below TDC at 1000 rpm by LDV system. And corresponding RMS values were measured 1~0.7Vp and 1.2~0.7Vp. During this period, tumble motion was still existed but the swirl flow was decayed and at 300°CA, the turbulence intensity increased linearly across the cylinder while the weak main flow has moved towards the exhaust valve area. These tumbling/swirl velocity values are much smaller than those of early induction which may suggest that the injected spray pattern during the compression stroke can be less affected by the tumble motion.

### Table 1

| SOI (°CA) | 0.5ms ASOI (Δθ=3°CA) | 0.7ms ASOI (Δθ=4.2°CA) | 0.9ms ASOI (Δθ=5.4°CA) | 1.1ms ASOI (Δθ=6.6°CA) |
|-----------|----------------------|------------------------|------------------------|------------------------|
| 60        |                      |                        |                        |                        |
| 90        |                      |                        |                        |                        |
| 120       |                      |                        |                        |                        |
| Colour index | 0                  | 0.5                  | 1                      |                        |

*Figure 5* Instantaneous Mie images during the induction stroke with tumble flow (B-B plane view); SCV is fully open.

Evolution spray pattern of start of injection at 270°CA, 285°CA and 300°CA after induction TDC, having the SCV fully closed (swirl), are displayed in Figure 6. Not like the spray pattern of intake stroke, the multiple spray plumes from a multi-hole nozzle can clearly be discriminated. As mentioned before, the axial and swirl mean velocities and also the turbulence level were not so large to overcome the spray plumes momentum and therefore much less deformation and dispersion of fuel droplets. Till the elapsed time of 0.9ms ASOI, the spray plumes patterns were similar regardless of the SOI timing. But, when the elapsed time goes over 0.9ms ASOI, the front shape of the tip of spray plumes can’t maintain its straight penetration and is distorted slightly perhaps due to RMS component of swirl flow. As start of injection timing retards, growth of the spray penetration is restricted by the upward moving piston, i.e. higher chamber pressure. The spray penetration of 300°CA SOI was strongly affected and the shorter spray penetration can be observed than that of others.
The evolution of spray pattern under tumble flow condition is displayed in Figure 7. Similar with the spray pattern under the swirl flow, the whole spray pattern was kept straight regardless of SOI. As start of injection timing retards, growth of the spray penetration is restricted by the upward moving piston. Especially, the spray penetration at 300°CA SOI was strongly affected and the shorter spray penetration can be observed than that of others due to increased chamber pressure. But, the front shape of the spray plumes maintains its straight penetration unlike that of the swirl flow.

From the spray pattern of late injection during compression stroke, it can be argued that the spray shape and penetration were affected by RMS component of in-cylinder flow and piston movement. Especially the spray penetration of the latest start of injection is strongly restricted by the upward moving piston.

| SOI (°CA) | 0.5ms ASOI (Δθ=3°CA) | 0.7ms ASOI (Δθ=4.2°CA) | 0.9ms ASOI (Δθ=5.4°CA) | 1.1ms ASOI (Δθ=6.6°CA) |
|----------|----------------------|------------------------|------------------------|------------------------|
| 270      | ![Image](image1.png) | ![Image](image2.png)   | ![Image](image3.png)   | ![Image](image4.png)   |
| 285      | ![Image](image5.png) | ![Image](image6.png)   | ![Image](image7.png)   | ![Image](image8.png)   |
| 300      | ![Image](image9.png) | ![Image](image10.png)  | ![Image](image11.png)  | ![Image](image12.png)  |

**Figure 6** Instantaneous Mie images - compression stroke with swirl flow (B-B plane view); SCV is fully closed.

| SOI (°CA) | 0.5ms ASOI (Δθ=3°CA) | 0.7ms ASOI (Δθ=4.2°CA) | 0.9ms ASOI (Δθ=5.4°CA) | 1.1ms ASOI (Δθ=6.6°CA) |
|----------|----------------------|------------------------|------------------------|------------------------|
| 270      | ![Image](image13.png) | ![Image](image14.png)   | ![Image](image15.png)   | ![Image](image16.png)   |
| 285      | ![Image](image17.png) | ![Image](image18.png)   | ![Image](image19.png)   | ![Image](image20.png)   |
| 300      | ![Image](image21.png) | ![Image](image22.png)   | ![Image](image23.png)   | ![Image](image24.png)   |

**Figure 7** Instantaneous Mie images - compression stroke with tumble flow (B-B plane view); SCV is fully open.
### 3.3 Temperature effect on spray droplet vaporisation

Since the Mie scattering technique is based on scattered light by liquid droplets only the remained non-yet-vaporised spray could be captured. More specifically, assuming that the base spray image for characterising evaporation would be at the lowest available temperature, then the combination of images taken at the base and a higher temperature would provide important qualitative information on the relative percentage of liquid already vaporised as was suggested by [20]. The principle of this approach is shown schematically in Figure 8 and the outcome would represent the probability density function of the liquid fuel droplets that are most likely to be evaporated.

![Figure 8](image)

**Figure 8** Schematic representation of the image processing procedure described in [20].

| Cond. | (a) $P_{inj}=70$bar, SOI=300CA, tumble flow | (b) $P_{inj}=120$bar, SOI=300CA, tumble flow |
|-------|---------------------------------|---------------------------------|
|       | Mie image | Vaporizing region | Mie image | Vaporizing region |
| 0.9ms ASOI | ![Image] | ![Image] | ![Image] | ![Image] |
| 1.1ms ASOI | ![Image] | ![Image] | ![Image] | ![Image] |
| Colour index | ![Image] | ![Image] | ![Image] | ![Image] |

**Figure 9** The effect of increased coolant temperature on the fuel vaporisation of the injected spray (A-A plane view).

Figure 9 illustrates the temperature effect on spray droplet vaporisation for sprays injected at 70bar and 120bar into the cylinder. In general, the results show a small amount of liquid fuel is vaporised for a temperature rise from 40°C to 90°C, this is perhaps expected since the boiling temperature of the fuel (isooctane) is 102°-105° @1bar; similar results was reported by [20] for the same increase in temperature. It is also evident that the amount of vaporise fuel is slightly more with higher injection pressure probably due to minor improvements in atomisation and effectively. A more specific analysis is needed to quantify the effect of coolant temperature of 90°C in spray vaporisation relative to 40°C. For example taking plane Mie images rather than surface images.
will help considerably and that taking extra images at temperatures above the fuel boiling point. Overall, the present results show that only small amounts of liquid are expected to vaporise during the injection and this would most likely happen around the edges of the individual fuel spray jets away from the injector exit.

3.4 Spray penetration and spray cone angle

The critical spray characteristic of multi-hole injector is the overall spray cone angle, which is defined during manufacturing by the relative angle between the injection hole axis and the injector axis of symmetry. The spray penetration and spray cone angle of experiments results at different injection timing, injection pressure and coolant temperature at 1000 rpm are plotted in Figure 10 and Figure 11. The spray penetration of 60°C SOI and T\textsubscript{coolant}=40°C is shown in Figure 10(a). The penetration is affected by fuel injection pressure so that at initial stage till 0.5ms ASOI, the spray penetration of 70bar is a little longer than those of 120bar this is mainly because the mechanical operational delay time of the injector at 120bar is longer as was shown by [20] who used the same injector. But, the injected fuel droplets, possessed large momentum by the higher fuel pressure, and consequently penetrate more into the cylinder than the lower injection pressure as the time ASOI increases. From the elapsed time of 0.6ms ASOI, the spray penetration of 120bar becomes longer than that of 70bar. The penetration increases straight till 0.8ms ASOI and from 0.8ms ASOI, the penetration stops at about 40mm due to loss of droplets momentum.

The spray penetration of 300°C SOI under swirl flow and tumble flow is respectively shown in Figure 10(b) and Figure 10(c). The spray penetration during compression stroke has similar trend to that of intake stroke. The penetration is also affected by fuel injection pressure. But additionally it is strongly affected by the chamber pressure (moving piston), which causes the maximum penetration of 35mm shorter than that of intake stroke. And after 0.9~1.0ms ASOI the spray tip starts to impinge on piston. So the injected fuel of the high pressure reaches on the piston earlier than that of the lower pressure. Therefore it is required to carefully consider the extent of the fuel impingement according to the fuel pressure. But, the temperature effect on the spray penetration is small and not as noticeable as the fuel pressure.

The plane (A-A) where the overall spray angle was calculated is shown in Figure 3, and the angle was measured between the extreme edges of the two outer jet sprays near the injector tip where the effects of the cross-flow was minimum. Figure 11 shows the spray cone angle during intake and compression stroke at different injection pressures and coolant temperatures. The results showed that the overall spray angle remained constant and almost independent of injection pressure, chamber pressure and coolant temperature. The results

![Figure 10](image1.png)

**Figure 10** Spray penetrations during intake and compression stroke.

![Figure 11](image2.png)

**Figure 11** Spray cone angle during intake and compression stroke.
also showed that there is a small and gradual reduction in the overall spray cone angle with the elapsed time ASOI, which is similar for all conditions tested, so that making the overall spray cone angle smaller than that of the nominal value like those reported by [10-11]. This can be related to the complex flow structure inside the nozzle hole especially the presence of different types of cavitation [13-14] depending on pressure difference across the nozzle due to the opening of the needle. In particular, there is a geometrical cavitation that forms on the upper part of the nozzle and can affect the trajectory of the exiting fuel jets by forcing them downwards.

4. Conclusions

Spray characteristics of a high pressure 6-hole multi-hole injector were investigated in an optical engine using Mie scattering. The results were obtained at an engine speed of 1000rpm and the effects of injection timing, in-cylinder charge motion, coolant temperature and injected fuel pressure were investigated. The most important findings are summarised below:

1. To obtain a homogeneous and stoichiometric mixture, in-cylinder swirl has proved to be far more effective than the tumble flow during the intake stroke. The results showed a clear shift of the spray jets in the direction of the intake cross-flow, with the smaller droplets strongly affected by the high velocity charged motion, swirl flow and the high level of turbulence. The end result was that individual spray jet plumes became indistinguishable presenting a very different spray pattern than expected.

2. The spray pattern of late injection during the compression stroke was little affected by tumble and swirl cross-flow. However, the effect of increased chamber pressure due to piston movement was considerable in limiting the spray jet penetration.

3. The effect of coolant temperature on fuel droplets vaporisation was found to be small when the temperature was raised from 40°C to 90°C.

4. Fuel pressure promotes spray penetration although, during compression stroke, it is strongly affected by the upward moving piston causing an increase in the air density in the cylinder.

5. The overall spray cone angle was found to be constant and almost independent of injection pressure, chamber pressure and coolant temperature. A gradual reduction in the overall spray angle was also found with elapsed time after the start of injection, which could be related to the development of cavitation in the nozzle holes.

Acknowledgement

The authors would like to thank Dr N. Mitroglou for his contribution to this research programme and Mr Tom Fleming and Jim Ford for their valuable technical support during the course of this work.

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