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EFFECTS OF INTAKE FLOW ON THE SPRAY STRUCTURE OF A MULTI-HOLE INJECTOR IN A DISI 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.

KEY WORDS : Mie scattering, Intake swirl, Spray structure, Multi-hole injector, Direct injection, Gasoline engine

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 (Wirth et al., 1998; Nouri and Whitelaw, 2002). These “first generation” injection systems, with swirl pressure atomizers have been shown lower fuel consumption by up to 20% in the case of stratified, overall-lean part-load operation, but showed no significant improvements in HC and NO_x emissions (Fraidl et al., 1996). 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 in 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 start of ignition.

Most recent studies have focused on an alternative strategy to the wall- and air-guided mode of mixture preparation for producing stratified fuel 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 (Wirth et al., 2004; Shim et al., 2008). 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 distribution of the droplet velocities and diameters as a function of nozzle design, and 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 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 (Li et al., 2004; Nouri and Whitelaw, 2006).

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, based on the expectation that they produce stable fuel sprays with fine fuel droplets independent of the time of fuel injection (Wirth et al., 2004). Multihole injectors have been studied because of their potential for achieving good fuel stratification, thus extending the lean limit further (Preussner et al., 1998). They also offer the highest possible flexibility in adapting the spray pattern layout to a particular combustion chamber design. The investigations of (Ortmann et al., 2001; Lippert et al., 2004; Mitroglou et al., 2006, 2007) 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 as a result of an enlarged surface area produced by separated spray jets, enhanced flexibility to direct the sprays towards the proximity of the spark plug and improved matching with the injector, generated spray and combustion chamber design. Recently, a series of detailed experimental investigations in a high pressure chamber and a DISI engine have been carried out and reported in (Mitroglou et al., 2005,

2006, 2007) regarding gasoline spray characteristics and mixture distribution. The constant high pressure chamber was equipped with a high-pressure multi-hole injector at injection and chamber pressures up to 20 MPa and 1.2 MPa, respectively. The test results in the constant chamber confirmed that the overall spray angle relative to the axis of the injector was independent of injection and chamber pressure. The effects of injection and chamber pressure on droplet velocities and diameter were also quantified. From the experimental results in the engine, 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 (Birth et al., 2006; Nouri et al., 2007) 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 (Zhao et al., 1997). 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. It includes a 4-valve modern pent roof cylinder head designed to allow spray guided operation. 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 1(a), downstream of the throttle valve, there is a second valve installed at the inlet of one of the ports, named the Swirl Control Valve (SCV). When this valve is closed, in-cylinder swirl is generated. The position of this valve can be varied manually from fully open to closed using an external gauge controller. Without SCV, this cylinder head was designed to generate high tumble flows; and the TVRo (steady flow tumbling vortex) values measured were 1.38 for 1000 rpm in a steady flow rig test (Karaiskos, 2005).

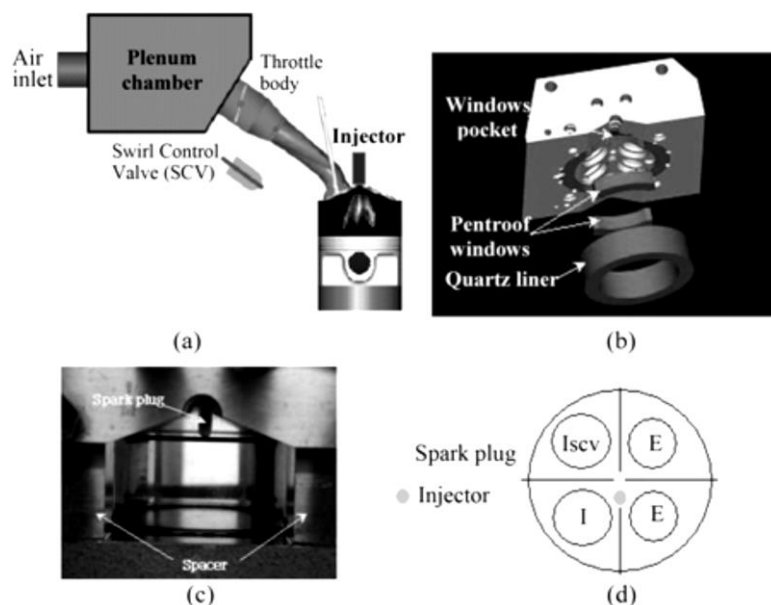


Figure 1. Engine set up: (a) Schematic of engine set-up; (b) Optical access arrangement; (c) Front view optical access (d) Cylinder head configuration.

Table 1. Test engine specifications.

Cylinder head	Pent roof	Ports	Tumble/Swirl
Bore×Stroke (mm)	83 ×92	In. Vavle timing	6°BTDC/ 50°ABDC
Compression ratio	10.5	Ex. Valve timing	50°BBDC/ 6°ATDC

Optical access to the combustion chamber was provided from the side (vertical images) via a fused silica cylinder liner (Figure 1(b) and 1(c)). As shown in Figure 1(b), there are also two quartz windows on both sides of the cylinder head to provide access to the pent roof area. The piston crown has a flat design so that an optical window can be fitted to obtain horizontal images. The injector and spark plug are oriented longitudinally, as shown in Figure 1(d); the line of the spark plug and injector is in the middle of the pent roof, between the intake and exhaust valves. All tests were carried out without combustion, and with the engine being motored. 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°CA. Engine control was achieved by using an advanced timer card (NI PCI-6602) with in-house software (Labview), which controlled injection and ignition.

The prototype injector used in the present experiments had been designed and manufactured by Bosch specifically for DISI engines, and it is a high pressure six hole injector with the holes symmetrically arranged on the periphery of an imaginary circle, as shown in Figure 2. The detailed specifications of the multi-hole injector, which operates with injection pressures up to 20 MPa, are described in Table 2. In view of the very short time available for fuel atomization and vaporization in DISI engines, particularly in the case of injection late in the compression stroke, the electromechanical response of the injector becomes an important consideration. Therefore, prior to the acquisition of spray images for the characterization of the spray development during the injection period, the injection delay, defined as the time between the rising edge of the triggering signal and first appearance of liquid at the nozzle exit, was quantified. Throughout preliminary injection testing, the injection delay time was found to be 0.7 ms, and during convenient spray image acquisition, the real injection delay time was adjusted to 0.2 ms with a 0.5 ms delay in the triggering signal.

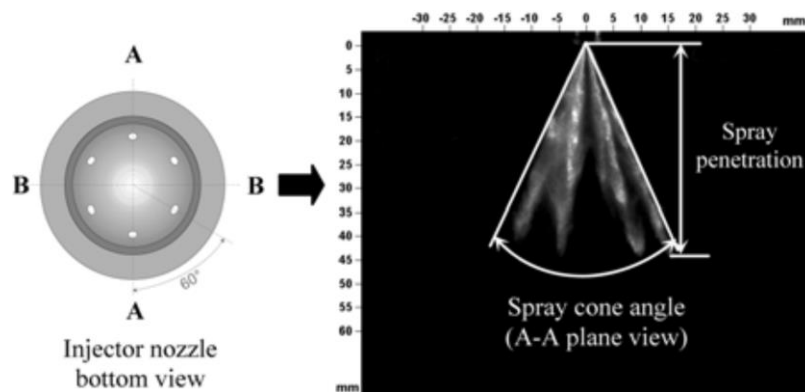


Figure 2. Injector nozzle and spray view.

Table 2. Specifications of multi-hole injector.

Hole numbers	6	Cone angle	42°
Hole diameter	140 μm	L/D ratio	2.14
Manufacturer	BOSCH	Production type	Gasoline DISI Proto type

2.2. Mie Scattering System

The optical set-up 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 fibers to the area of interest.

Qualitative and quantitative information about the spray was extracted from high-resolution forward illuminated images recorded with a non-intensified 12 bit CCD PCO SensiCam camera, offering a resolution of 1024 × 1240 pixels and low readout noise, in conjunction with a Nikkor telescopic zoom lens (75~300 mm 1/4.5~5.6). Image acquisition timing is controlled by the engine control system, which is equipped with two general purpose triggers. An active cycle frequency of image acquisition and fuel injection was set in such a way as to allow sufficient time (15 s) for the xenon flash light to recharge fully.

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 3. The injection duration was kept at 1 ms for all of the test conditions.

Table 3. Experimental conditions.

SCV position	Open/Close	Injection pressure	7MPa/12MPa
Coolant temperature	40°C/ 90°C	Fuel	ISO Octane
Intake air temperature	~20°C	Operating mode	Homogeneous/ Stratified

Spray imaging was repeated three times for each time step of each test case. The spray cone angle and penetration, obtained from the Mie images, are defined and provided in Figure 2; images taken in the A-A plane view were used to obtain the spray cone angle. For the investigation of in-cylinder spray characteristics, the injected sprays of the 6-hole injector were visualized in the B-B plane view at engine speeds of 1000 rpm. There is a 0.2 ms delay between the injection trigger signal and first appearance of a spray. Therefore, spray evolution images were captured from 0.3 ms ASOI to 1.1 ms ASOI at 0.1 ms time intervals.

3. RESULT AND DISCUSSION

3.1. Early Injection for Homogeneous Stoichiometric Operation

Multi-hole injectors are known to have stable spray structures under various operating conditions. The 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 the spray's penetration in the combustion chamber. Homogeneous operation dictates early injection of the fuel during the induction stroke. A selection of early injection timing includes injection of fuel at ATDC 60°CA, 90°CA, and 120°CA.

From the previous LDV measurement of in-cylinder flow under 'SCV open' or tumble flow condition (Kariskos, 2005), it was realized that high velocities were generated during the intake process, rising to a maximum between ATDC 60°CA and 120°CA, 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 neither strong nor well defined with respect to cylinder axis. 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 the spray pattern at different injection timings of ATDC 60°CA, 90°CA, and 120°CA, with the SCV fully closed (maximum swirl), a fuel injection pressure of 7 MPa, and a coolant temperature of 40°C is displayed in Figure 3. As shown, there are two distinct features in the spray structures; one is that the multiple spray plumes (jets) from the multi-hole nozzle cannot 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 takes 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 are dispersed rapidly under a 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. These effects are more evident when the elapsed time goes over 0.7 ms ASOI, the whole spray is now inclined downstream and furthermore, the fuel droplets of the tip edge

start to be separated from the main plume jet towards the cross-flow direction; the latter effect may be a result of high swirling and turbulence. The extent of the separation increases with elapsed time after the start of injection, and those of ATDC 90°CA and 120°CA SOI are more pronounced than that of ATDC 60°CA.

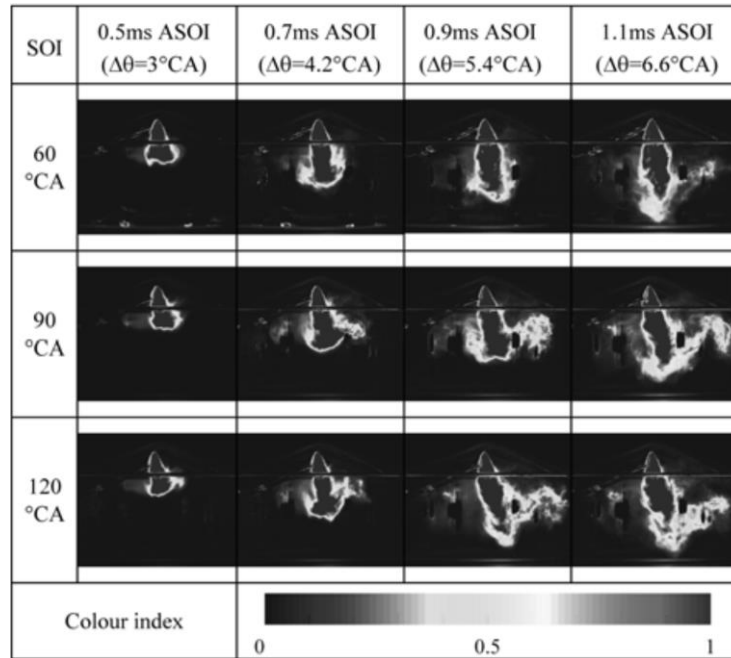


Figure 3. Mie images during the intake stroke under swirl flow, $P_{inj.}=7\text{MPa}$, and $T_{coolant}=40^\circ\text{C}$.

At 0.9–1.1 ms ASOI, the spray tilt is even more recognizable, with downstream injected fuel droplets 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 the multi-hole nozzle to accommodate the homogeneous charge mixture. At 1.1 ms of elapsed time, a small portion of the separated fuel droplets reaches the cylinder wall, which is undesirable. The spray evolution with tumble flow, with a fuel injection pressure of 7 MPa and a coolant temperature of 40°C at the start of injection at ATDC 60°CA, 90°CA, and 120°CA, is displayed in Figure 4. Similar to the spray pattern under swirl flow, and for the same reasons, the multiple spray plumes cannot be distinguished. The injected fuel spray plumes cannot avoid the strong influence of the incoming air cross flow during intake valve opening due to the injector position in the cylinder head. Generally, the tumble flow does not deflect the spray pattern as strongly as the swirl flow, and there is no fuel droplet separation phenomenon; the latter indicates no impingement on the liner. The larger spray deflection and droplets separation with swirl flow, as seen in Figure 3, clearly suggest the presence of centrifugal force acting on the fuel droplets away from the centre of the cylinder. Overall comparison with the spray patterns under the flow of Figure 3 indicates that the spatial distribution of the injected fuel spray under tumble flow is apparently less than that of swirl flow, especially over the elapsed time of 0.7 ms. In addition, the tilt of the overall spray in the direction of the intake cross-flow is not as much as the swirl. Therefore, for a well distributed, homogenized and stoichiometric mixture, it is more important for swirl flow to be generated in the cylinder than tumble flow.

3.2. Late Injection for Stratified Lean Operation Mode

The concept of stratification needs to be clarified according to the 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. 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 ATDC 270°CA, 285°CA, and 300°CA, which were defined as medium and late injection timings. During this period, tumble motion still existed, but swirl flow decayed and at ATDC 300°CA, the turbulence intensity

increased linearly across the cylinder while the weak main flow 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 may be less affected by the tumble motion. The evolution of the spray pattern at the start of injection at ATDC 270°CA, 285°CA, and 300°CA, with the SCV fully closed (swirl), fuel injection pressure of 7 MPa, and coolant temperature of 40°C, is displayed in Figure 5. Not like the spray pattern of the 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 as to overcome the spray plume momentum, and therefore there is much less deformation and dispersion of fuel droplets. Until an elapsed time of 0.9 ms ASOI, the spray plume patterns were similar regardless of SOI timing. However, when the elapsed time exceeds 0.9 ms ASOI, the front shape of the tip of spray plumes can no longer maintain its straight penetration, and is distorted slightly perhaps due to the RMS component of swirl flow. With respect to the start of injection timing, the growth of spray penetration is restricted by the upward moving piston and higher chamber pressure. The spray penetration of ATDC 300°CA SOI was strongly affected, and a shorter spray penetration can be observed.

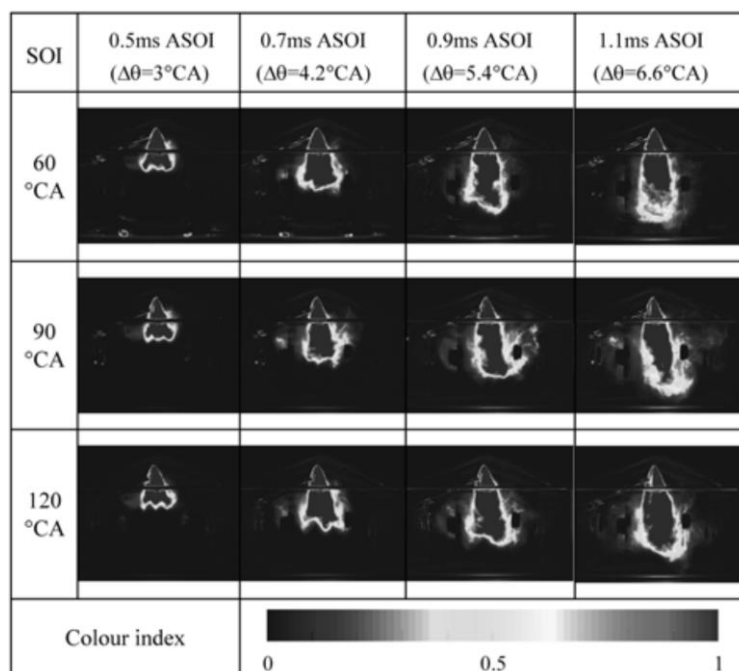


Figure 4. Mie images during the intake stroke under tumble flow, $P_{inj}=7\text{MPa}$, and $T_{coolant}=40^\circ\text{C}$.

The evolution of the spray pattern under conditions of tumble flow, fuel injection pressure of 7 MPa, and coolant temperature of 40°C is displayed in Figure 6. Similar to the spray pattern under swirl flow, the whole spray pattern was kept straight regardless of SOI. With respect to the start of injection timing, growth of the spray plumes maintains its straight penetration, unlike that of the swirl flow. From the spray pattern of late injection during the compression stroke, it can be argued that the spray shape and penetration were affected by the RMS component of in-cylinder flow and piston movement. In particular, the spray penetration of the latest start of injection is strongly restricted by the upward moving piston.

3.3. Temperature Effect on Spray Droplet Vaporization

Since the Mie scattering technique is based on scattered light by liquid droplets only the remaining non-yet-vaporized spray could be captured. More specifically, assuming that the base spray image for characterizing evaporation would be at the lowest available temperature, then the combination of images taken at the base and at a higher temperature would provide important qualitative information on the relative percentage of liquid already vaporized, as was suggested by (Mitroglou, 2005). The principle of this approach is shown schematically in Figure 7, and the outcome would represent the probability density function of the liquid fuel droplets that are most likely to be evaporated.

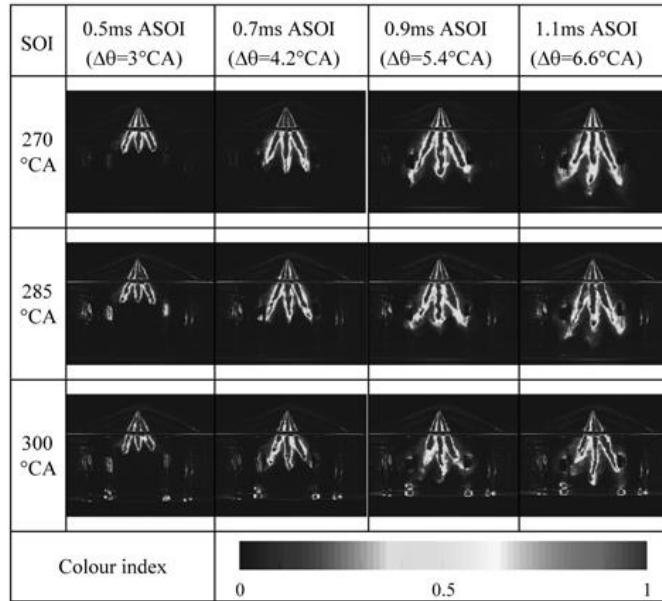


Figure 5. Mie images during the compression stroke under swirl flow, $P_{inj.}=7\text{MPa}$, and $T_{coolant}=40^\circ\text{C}$.

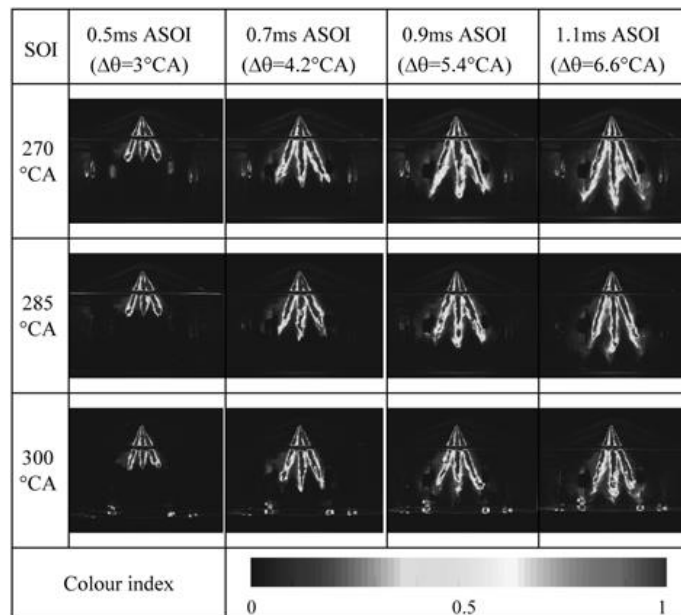


Figure 6. Mie images during the compression stroke under tumble flow, $P_{inj.}=7\text{MPa}$, and $T_{coolant}=40^\circ\text{C}$.

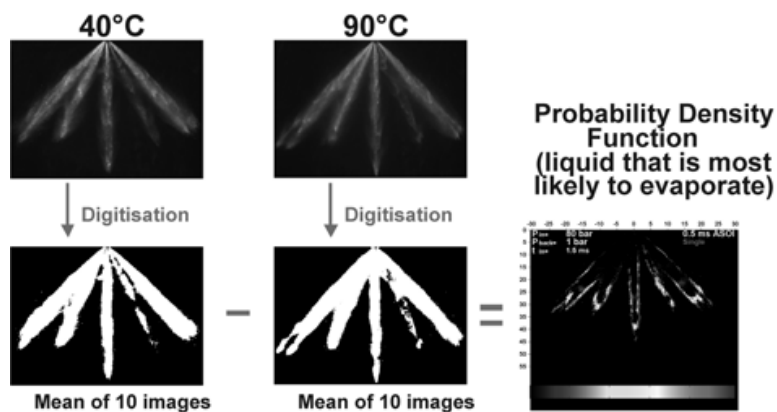


Figure 7. Mie image processing for vaporizing region.

Figure 8 illustrates the temperature effect on spray droplet vaporization for sprays injected at 7 MPa and 12 MPa into the cylinder. In general, the results show that a small amount of liquid fuel is vaporized for a temperature rise from 40°C to 90°C, this is perhaps expected since the boiling temperature of the fuel (isooctane) is 102°C~105°C @0.1 MPa; similar results was reported (Mitroglou, 2005) for the same increase in temperature. It is also evident that the amount of vaporize fuel is slightly more with higher injection pressure probably due to minor improvements in atomization and efficacy. A more specific analysis is needed to quantify the effect of a coolant temperature of 90°C in spray vaporization relative to 40°C. For example, taking plane Mie images rather than surface images will help considerably, along with taking extra images at temperatures above the fuel boiling point. Overall, the present results show that only small amounts of liquid are expected to vaporize during the injection, and this would most likely happen around the edges of the individual fuel spray jets, away from the injector exit.

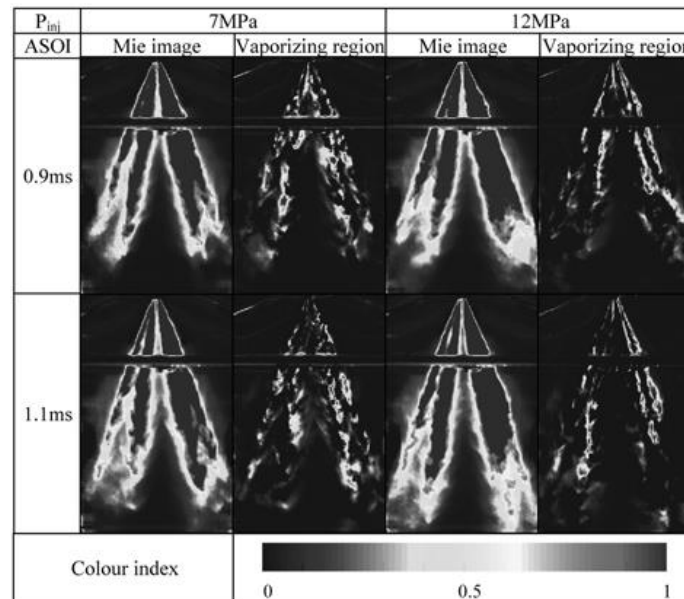


Figure 8. Effect of the coolant temperature on fuel vaporization at ATDC 300°CA SOI under tumble flow.

3.4. Spray Penetration and Cone Angle

The spray penetration and spray cone angle at different injection timing, injection pressure, and coolant temperature at 1000 rpm are plotted in Figure 9 and Figure 10. The spray penetration of 60°CA SOI and $T_{coolant}=40°C$ is shown in Figure 9(a). The penetration is affected by fuel injection pressure so that at initial stage till 0.5 ms ASOI, the spray penetration at 7 MPa is a little greater than that of 12 MPa, mainly because the mechanical operational delay time of the injector at 12 MPa is longer. However, the injected fuel droplets had a substantial momentum, as a result of the higher fuel pressure, and consequently, penetrated further into the cylinder than those injected at a lower injection pressure as the time ASOI increases.

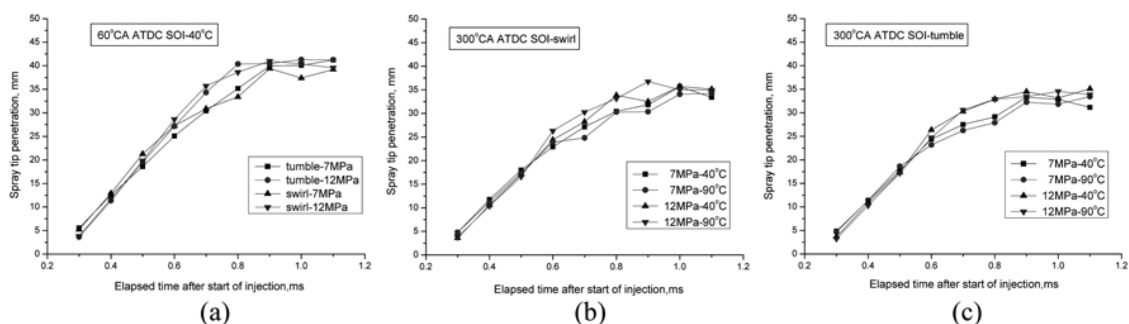


Figure 9. Spray penetration during intake and compression strokes.

From the elapsed time of 0.6 ms ASOI, the spray penetration at 12 MPa becomes greater than that at 7 MPa. The penetration continues to increase until 0.8 ms ASOI, and from 0.8 ms ASOI onward, the penetration stops at about 40 mm due to loss of droplet momentum. The spray penetration of 300°CA

SOI under swirl and tumble flow are shown in Figure 9(b) and Figure 9(c), respectively. The spray penetration during the compression stroke has a similar trend to that of the 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 a maximum penetration of 35 mm, shorter than that of the intake stroke. After 0.9~1.0 ms ASOI, the spray tip starts to impinge on the piston. The injected fuel of high pressure reaches the piston earlier than that of lower pressure. Therefore, it is necessary to carefully consider the extent of 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 2, 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 10 shows the spray cone angle during intake and compression stroke, and at different injection pressures and coolant temperatures.

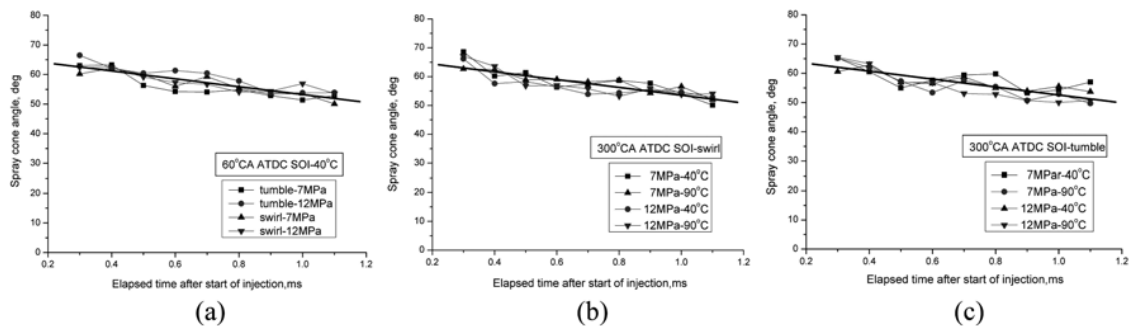


Figure 10. Spray cone angle during intake and compression strokes.

The results showed that the overall spray angle remained constant and almost independent of injection pressure, chamber pressure, and coolant temperature. There is also a small and gradual reduction in the overall spray cone angle with the elapsed time ASOI, which is similar for all conditions tested, making the overall spray cone angle smaller than that of the nominal value. This can be related to the complex flow structure inside the nozzle hole, especially in the presence of different types of cavitation, depending on pressure differences across the nozzle due to the opening of the needle. In particular, there is a geometric 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. CONCLUSION

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 1000 rpm, and the effects of injection timing, in-cylinder charge motion, coolant temperature, and injected fuel pressure were investigated. The most important findings are summarized below:

- (1) To obtain a homogeneous and stoichiometric mixture, in-cylinder swirl proved to be far more effective than tumble flow during the intake stroke. The results showed a clear shift of the spray jets in the direction of the intake cross-flow.
- (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 vaporization was found to be small when the temperature was raised from 40°C to 90°C.
- (4) Fuel pressure promotes spray penetration although, during the 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 can be related to the development of cavitation in the nozzle holes.

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