Mixture distribution in a multi-valve twin-spark ignition engine equipped with high-pressure multi-hole injectors

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Abstract: Laser-induced fluorescence has been mainly used to characterise the two-dimensional fuel vapour concentration inside the cylinder of a multi-valve twin-spark ignition engine equipped with high-pressure multi-hole injectors. The effects of injection timing, in-cylinder charge motion and injector tip layout have been quantified. The flexibility in nozzle design of the multi-hole injectors has proven to be a powerful tool in terms of matching overall spray cone angle and number of holes to specific engine configurations. Injection timing was found to control spray impingement on the piston and cylinder wall, thus contributing to quick and efficient fuel evaporation. It was confirmed that in-cylinder charge motion plays a major role in engine’s stable operation by assisting in the transportation of the air-fuel mixture towards the ignition locations (i.e. spark-plugs) in the way of a uniformly distributed charge or by preserving stratification of the charge depending on operating mode of the engine.

1. Introduction

As part of the effort to reduce vehicle fuel consumption and exhaust CO\(_2\) emissions, the first-generation direct-injection spark-ignition (DISI) engines are already in production in Europe and Japan. Recent emissions legislation in major industrialised nations have appeared to be even more stringent, thus adding additional challenges to the development of gasoline direct-injected engines. Exploring the potential of other known combustion concepts coupled with new-generation high-pressure injectors is one approach. There are different combustion concepts for direct-injection gasoline engines, classified according to (i) the relative position of the injector towards the spark plug and the piston crown, as a function of injection timing and air motion, and (ii) the approach to mixture preparation. They are categorised as wall- air-, or spray-guided combustion systems [1]. Further expansion to that system of classification can be done by recognising the various types of charge motion employed, namely, swirl and tumble [2]. In all concepts, good combustion is associated with the formation of a stable and ignitable mixture around the spark plug at the time of ignition.

Swirl pressure atomisers, known as first-generation injectors, have been the main fuel injection equipment (FIE) for DI gasoline engines in production so far and, as such, they have been investigated thoroughly [3-14]. The preferred combustion concept for this injector is the wall-guided
configuration employing side-injection (wide-spacing) over a moderate range of injection pressures (50-120bar). The disadvantage of the above FIE system is the sensitivity of the sprays generated from swirl pressure atomisers, to the chamber pressure. In addition, wall-guided systems use a piston crown cavity to direct the mixture towards the spark-plug, which can easily result in increased HC and soot emissions in the unfortunate case of fuel being injected outside the piston cavity [2, 15-17]. Through continuous development, the second-generation DI gasoline engines have emerged. Main differences between the two known generations are the type of FIE used and the main combustion concept employed. In an attempt to overcome the spray instability at high chamber pressures, a multi-hole nozzle was proposed incorporating several uniformly spaced holes in a configuration simulating diesel injector nozzles. As a result of this, the wall-guided concept has been replaced with the spray-guided one, taking full advantage of the stability of multi-hole injector sprays [2, 15, 18-25].

In the present investigation a Yamaha purpose-built single-cylinder research engine has been used, in order to explore the potential of multi-hole nozzles in creating stratified as well as homogeneous charge for engine part- and full-load operation, respectively. To realise this, central-injection has been employed by two different multi-hole nozzle designs. In addition, two extreme in-cylinder flow patterns have been tested, minimum and maximum swirl. The fuel vapour distribution at the time of ignition has been visualised for the aforementioned cases using the laser induced fluorescence (LIF) technique at the engine speed of 1500rpm. The following sections describe, in turn, the experimental arrangement and instrumentation, the results and their implications, and summary conclusions.

2. Experimental arrangement and instrumentation

2.1. Injector nozzle geometries

Two prototype multi-hole injectors were tested. The first one was a 6-hole nozzle having all holes symmetrically arranged on the periphery of an imaginary circle (Figure 1a). Its nominal overall spray cone angle was set to 40° and the L/D ratio was 2.14. The second nozzle was based on a 12-hole design which additionally incorporates a central hole while two of the side holes are blocked, creating space for possible positioning of the spark-plug (Figure 1a). In this case the overall spray cone angle was 90° and this nozzle has twice the L/D ratio of the 6-hole nozzle. The reason of this change in the L/D ratio is the fact that in order to maintain similar static flow-rates for both nozzles, the hole diameter of the 12-hole injector had to be reduced by half while the remaining design parameters were kept constant. From this point onwards, the first injector will be referred to as the 6-hole nozzle and the second one as the 12-hole nozzle.

![Figure 1: a. Multi-hole nozzle designs. b. Triggering signal and typical needle lift diagram.](image)
The nominal injection pressure for both nozzles was 120 bar. Based on the needle lift curve for both injectors (Figure 1b), there was a minimum possible injection duration of around 0.7 ms. This is the result of the opening delay for both valves being set to 0.6 ms.

2.2. Engine configuration

A Yamaha single cylinder purpose-built research engine was used for this investigation. It features a full stroke length optical quartz liner, an elongated Bowditch piston with flat shaped crown and a pent-roof style combustion chamber. The cylinder-head utilises a 5-valve configuration, 3 inlet valves and 2 exhaust valves. The injector is located centrally, with a 1 mm offset to the geometrical centre of the cylinder. In order to overcome known issues of cocking and injector deposits the initial closed-spacing configuration has been replaced with a twin-spark arrangement, employing 2 spark-plugs at the two sides of the cylinder, installed one opposite to the other (Figure 2b). This is believed to be the best solution for reducing injector’s tip temperature during engine firing operation, which is the main source for cocking. The proposed configuration shares features from a close-spacing, spray-guided concept as well as from wide-spacing, wall-guided designs. Thus, it is named twin spark-plug, piston-guided engine.

As it can be seen in Figure 2a, downstream the throttle valve, which is mainly not used in this investigation, there is a second valve installed, named Swirl Control Valve (SCV). This valve, when closed, covers two-thirds (2/3) of the inlet port, which effectively blocks the flow of two inlet valves and directs the whole intake flow to the third valve (Figure 2b). In this way maximum in-cylinder swirling flow is achieved. The angle of this valve is variable and controlled automatically; so it is possible to achieve any desired swirl ratio in the cylinder by varying the SCV angle. From this point
forward the statement “SCV open” or “SCV90” means no swirl (valve fully open), while “SCV closed” or “SCV0” means maximum in-cylinder swirl (valve closed).

The engine could be run fired or motored. For all the experimental results presented here the engine was motored by a shunt dynamometer, as the study interest of this paper was on the preparation of the desired mixture prior to the combustion event. An in-house developed code, installed on a lab PC, simulated the electronic control unit of the engine, providing in this way maximum flexibility in the selection of injection/ignition timings. Synchronisation between the controlling PC and the engine was achieved through a crankshaft encoder, having a resolution of 0.36° crank angle (CA). The engine load was determined by a laminar flow meter that indicated the volumetric efficiency of the engine. A three-piston-type pump, coupled to an electric motor, was responsible for delivering high-pressure fuel (up to 120bar) to the common rail, which has been specifically built with one injector outlet. This common rail was connected to the injector via a pipe with specific diameter and length. A fuel pressure regulator is attached on the common rail and is controlled automatically. It regulates fuel pressure by throttling the return fuel line. A fuel flowmeter is installed inline with the injector and provides the fuel consumption under any operating conditions. Signals from the aforementioned instrumentation and three pressure transducers, installed in the inlet-, exhaust pipes and engine cylinder, are gathered and saved in the monitoring PC and further processing provides information on engine’s operating conditions. A summary of the engine’s specifications can be found in Table 1.

| Bore x Stroke | 73 x 59.6 (mm) | Valvetrain | 5valve DOHC |
|---------------|---------------|------------|-------------|
| Con. Rod Length | 116 (mm) | Inlet Valve Opening – Closing | -35° - 245° |
| Comp. Ratio ε | 8.7 | Exhaust Valve Opening – Closing | -245° - 35° |
| Max Speed | 3000rpm | Cylinder-head Ports | Low Tumble/Var. Swirl |

Table 1: Engine specifications.

2.3. Laser-induced fluorescence
During the planar LIF measurements the optical set-up depicted in Figure 3 was used. Laser light at 266nm was provided by a frequency quadrupled Nd:Yag pulsed laser at a repetition rate of 12.5Hz. A collimated laser sheet of 50mm width and 0.5mm thickness was produced using a 600mm biconvex spherical lens, a –25mm planoconcave cylindrical lenses and a 150mm planoconvex cylindrical lenses. The sheet entered the cylinder through the optical liner, having its plane almost aligned to the plane of the injector with a 6.5mm offset to the plane of the two spark-plugs. A two-component fuel was used for the experiments consisting of isoctane (base fuel) and 3-pentanone (tracer). The tracer concentration was chosen on the basis of achieving good signal-to-noise ratio and minimum incident light absorption. Therefore a tracer concentration of 20% by volume was chosen which offered satisfactory signal intensity at an acceptable laser light intensity and minimal light absorption. Calibration of the LIF signal was achieved with the aid of a premixing chamber (see Figure 2a), which served the purpose of preparing a homogeneous mixture of known composition. Images of the homogeneous charge were acquired and coupled with background and data images. Each image set consisted of 50 single-shot images and post-processing, on a pixel-by-pixel basis, revealed the spatial in-cylinder air/fuel ratio distribution for each case tested.
In an attempt to fully characterise the mixture distribution in a multi-valve, twin-spark DI engine and to explore the potential of multi-hole nozzles, a variety of different operating modes and conditions were tested. A summary of the most appealing ones can be found in Table 2.

| Engine Speed | 1500rpm | Injector | 6-/12-hole |
|--------------|---------|----------|-------------|
| Throttle Valve | Wide Open | Injection Pressure | 80bar |
| SCV | Open/Closed | Injection Strategy | Single |
| Coolant Temp. | 90°C | Fuel | 80% isoctane-20% 3-pentanone |
| Int.Air Temp. | ~40°C | Oper. Mode (Eng. Load) | Homogeneous/Stratified (Full/Partial) |

Table 2: Experimental cases.

3. Results and discussion

Multi-hole injectors are known to have stable spray structure under various operating conditions [2, 15, 19-21, 23-25]. 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. A typical characteristic of multi-hole sprays is the increased penetration, relative to fan sprays or hollow cone sprays from outward opening nozzles. An immediate result of the high penetration momentum is the insensitivity of these sprays to in-cylinder charge motion. By elimination of possible charge motion control mechanisms, engine management is significantly simplified.

Based on previous experiments and CFD calculations, the atomisation controlling mechanism of multi-hole injectors is cavitation. In conjunction with nozzle design parameters, such as L/D ratio, cavitation controls the dynamic fuel flow rate as well as the shape of the individual fuel jet plumes. Figure 4 presents the volumetric capacity of the tested injectors as a function of injection pulse duration. The 12-hole nozzle appears to have increased flow rate, at injection durations greater than 1ms, compared to the 6-hole one. This is a result of the difference in L/D ratios. 6-hole nozzle incorporates an L/D ratio of 2.14; in this case hole cavitation is extended from the inlet all the way to the exit of the hole, resulting in a “brushy” shaped spray and reducing the effective flow area of each hole. On the other hand, the 12-hole nozzle has twice the L/D ratio of the 6-hole one. Hole cavitation is present again but not extended to the exit of the hole, resulting in a “slim” fuel jet shape that occupies almost 100% of the hole flow area.
Known advantages of stoichiometric homogeneous operation of DI engines over their conventional PFI counterparts, such as increased volumetric efficiency and reduction of charge temperature (allowing higher compression ratios to be used), create the need of exploring the potential of multi-hole injectors in creating a well-homogenised, stoichiometric mixture. The first part of the LIF data presented here are oriented towards this direction. The engine speed was set to 1500rpm and the global AFR around stoichiometric (~15).

3.1. Homogeneous stoichiometric operation

Injection timing

Homogeneous operation dictates early injection of the fuel during the induction stroke, in the case of single injection strategy. A selection of early injection timings for this study includes injection of the fuel at 50° and 70°CA after TDC as well as at later stages during the induction stroke, such as 90° and 120°CA. The general injection timing scheme had to be modified slightly according to each injector’s specifications. For the case of the 12-hole nozzle, which features a 90° overall spray cone angle, an earlier timing than 50° was also tested (30° after TDC), while the latest timing of 120° did not show any interest. The reason is that when injecting late, because of the large spray cone angle relative to engine’s bore, spray impinges on the liner. Similar effect can be seen in early injection timings, where fuel impinges on the piston. The latter behaviour is preferred, compared to liner wall wetting, because of better behaviour in smoke and HC emissions.

Start of injection process at 30°CA after TDC, at induction stroke, is quite early injection timing. Liner wetting is totally avoided, all the fuel impinges and stays on the piston though. The injection duration for achieving stoichiometric charge is roughly 35°CA (including injector opening delay) at 1500rpm engine speed, considering the flow rate of the 12-hole injector. This fact leads to an end of injection (EOI) timing around 65°CA after TDC. As a result, there is enough time left for the fuel to vaporise completely of the piston surface and form a uniform ignitable mixture cloud around the vicinity of the spark-plug locations at the time of ignition (~330-360°C). As the injection timing is retarded at 50°CA after TDC, always having the SCV fully open (no swirl), liner wetting appears towards the end of injection, it is minimum though. The effective time for mixing is reduced at this case and it can be seen in the 2-D AFR distribution image taken at 340°CA, otherwise 20°CA before compression TDC. There is an AFR gradient along the laser path; mixture distribution, from left to right, appears slightly

![Figure 4: a. Volumetric capacity of tested injectors at 120bar injection pressure, as a function of injection pulse duration under atmospheric conditions. b. Schematic representation of in-hole flow pattern for the two nozzle designs.](image-url)
lean in fuel (AFR~16-17) in the beginning and passed the centre line of the liner, it becomes richer to AFR values close to 12. In the vertical axis of the image, the upper half of the combustion chamber is richer in fuel than the bottom half, especially right below the two outer inlet valves. This is happening due to a combined effect of intake valves having their maximum lift at around 100°CA after TDC and the relatively large spray cone angle for this engine size. As fuel is being injected at such large angle (90°) around the maximum lift timing of the intake valves, spray inevitably starts interfering with the intake valves. According to operating conditions and the amount of fuel actually hitting and staying on the valves, it is possible that a small percentage will eventually be vaporised towards the end of compression stroke, where combustion chamber temperatures rise considerably. At such late timings, fuel recently vaporised, stays around the vicinity of the valves due to lack of in-cylinder motion. Spray interfering with the intake valves, starts becoming an issue at later injection timings, of 70° and 90°CA after TDC, where the main injection process finishes on or after the maximum valve lift event. In these two cases, there is also extensive liner wetting, an undesired situation mainly because of the limited vaporisation conditions that cylinder walls can offer and the further damage that fuel causes to the lubricant. The aforementioned facts, combined with the weak airflow of a 3-intake valve system and the low engine speed (1500rpm), cause a rather uneven fuel vapour distribution, as it can be seen in Figure 5.

Figure 5: Two-dimensional air/fuel ratio (AFR) distribution at 340°CA (20° before compression TDC) of the 12-hole injector at injection timings of 30°, 50°, 70° and 90°CA after induction TDC.

As the 12-hole injector proved to have a relatively large overall spray cone angle for our engine specifications, the second nozzle tested features a much smaller nominal spray cone angle of 40°. Although liner wall wetting is not an issue with the narrow spray from our 6-hole injector, fuel impinging on the flat piston crown is evident. Provided that piston impingement is less harmful for an engine’s emission levels than liner wetting, the initially set protocol, regarding injection timings, was followed. Starting from 50°, 70°, 90° and finally 120°CA injection timings, the effect of fuel impinging on the piston is evident on AFR distribution across the centre plane of the cylinder. At first, there is no direct liquid fuel interaction with intake valves. At early timings, such as 50° and 70°CA after TDC, piston impingement rules the final fuel vapour distribution. In both cases, most of the fuel injected lies on the piston and evaporation starts from the hot piston surface as compression temperature rises; a very similar mechanism to what was mentioned before, for the case of early injection of the 12-hole injector. As a result, fuel vapour distribution between the two nozzle designs is comparable and features areas rich in fuel and steep AFR gradients across the plane of the image. Although the 6-hole nozzle produces a more compact and narrow spray shape, compared to the one of the 12-hole injector, rich areas in fuel at the time of ignition can be found at the sides of the cylinder.
Air flow does not enhance atomisation and ultimately evaporation at 1500rpm, as mentioned before, due to valve configuration of the cylinder-head and increased penetration momentum of the individual fuel jet plumes. Therefore, the mechanism that justifies transportation of vapour clouds to the sides of the cylinder can be found in the spray itself. The 6-hole injector utilises an L/D ratio of 2.14, half the value of the 12-hole nozzle, thus individual fuel jets appear to have a “brushy” shape that enhances air entrainment. The efficiency of this mechanism becomes lower as L/D ratio increases. Higher L/D ratios cause increased liquid jet penetration, minimising possible air entrainment due to spray’s high velocity flow field interaction with the surrounding environment.

**Figure 6:** Two-dimensional air/fuel ratio (AFR) distribution at 340°CA (20° before compression TDC) of the 6-hole injector at injection timings of 50°, 70°, 90° and 120°CA after induction TDC.

The last two cases in Figure 6 stand for 90° and 120°CA injection timings. Distribution of AFR is not as uniform as expected, but the contrast between too rich and too lean, in fuel, areas is missing. This happens mainly due to the fact that there is no fuel impinging on the piston in both cases. Once again, at the given engine speed and intake port design, sprays from 6-hole nozzles show the trend to occupy both sides of the cylinder, in a more uniform way than 12-hole sprays, which showed preference towards the right half of the cylinder.

**In-cylinder charge motion**

So far, it has been proven that the lack of strong airflow is partly responsible for the poor mixing presented in this study. Given the fact that our engine features low tumble intake ports, another device has been installed that increases swirl levels in the cylinder. An interesting piece of equipment offering the possibility of fully customisable swirl levels by varying the angle of the SCV. Introduction of moderate swirl (swirl control valve closed 30°, relative to fully open position and zero swirl) immediately improved mixing and fuel vapour distribution, as seen in Figure 7. Moving one step forward and introducing maximum swirling motion in the cylinder charge did not improve more the final AFR distribution. As it is evident in the third frame of Figure 7, AFR values given from the LIF image exceed the nominal air/fuel ratio by 65-70%. Although it is probably the most homogeneous distribution we have acquired so far, it appears significantly rich in fuel; giving a false reading. This is an extreme situation and swirl levels achieved with the SCV closed, are by far for larger engine designs. The immense swirling motion of the in-cylinder air has centrifuged finely atomised fuel droplets resulting in a certain amount of fuel being on the quartz liner. Scattered laser light from liquid present in the laser plane causes the fuel on the liner to fluorescence, thus providing a false idea of pixel intensity in the data image. Therefore, it is wise to momentarily keep aside the quantified information of an LIF measurement and concentrate on the qualitative result.
Figure 7: Comparison of three different in-cylinder swirl levels. a. No swirl (SCV open), b. Medium swirl (SCV 30° closed) and c. Maximum swirl (SCV closed) in the case of 12-hole injector at 50°CA injection timing.

The effect of in-cylinder air motion to the shape of multi-hole sprays is rather insignificant, as mentioned already before and can be seen in Figure 8. A direct comparison between two extreme swirl conditions shows that individual liquid fuel plumes are very little affected by the air motion.

Figure 8: Effect of in-cylinder swirl on multi-hole sprays.

3.2. Stratified lean operation

Injection timing

A whole different concept applies to the stratified operation. 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 a 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, thus, fuel consuming throttling of the engine is avoided and the load is controlled quantitatively by the amount of fuel injected. This pattern has been followed in the present study and partial engine load was simulated with an overall AFR of ~30.

The concept of stratification needs to be clarified according to the engine design. In the case of a twin-spark ignition engine, good stratification means each spark-plug is surrounded by ignitable mixture. Therefore, for the purposes of this study, 2 distinct mixture clouds are expected to be present at the two sides of the cylinder (image left-right), where the 2 spark-plugs are located.
The most common technique to achieve stratification is by injecting fuel during the compression stroke of the engine and after the closure of the inlet valve. In this study, injection during compression stroke has been divided into 3 subcategories; early timings (220°-260°CA after induction TDC or 100°-140°CA before compression TDC), medium timings (270°-280°CA or 80°-90°CA before compression TDC) and late injection timings (290°-310°CA or 50°-70°CA before compression TDC). LIF images from stratified operation have undergone certain post-processing and the normalised deviation of the nominal AFR is presented. Nominal AFR is set during tests according to the fuel quantity injected. As it can be seen in the colourmap, right of the images, value 0 means the AFR has its nominal value, while 1 (red colour) and –1 (blue colour) stand for the richest and leanest, in fuel, areas detected in the images.

Early and late injection timings during compression stroke did not provide the expected results for the 12-hole injector. In turn, liner wetting and extensive piston impingement for early and late timings caused undesired stratification. For the case of late injection, the lack of time available for vaporisation is evident, causing the fuel to stick on the piston. Very good results are obtained at medium injection timings. The relative position of the piston to the fuel spray is such that directs the mixture cloud towards the spark-plugs. The time available for vaporisation is sufficient for both, free fuel spray and spray previously impinged on the piston. A sample development of injection and mixture formation process for medium injection timings can be found in Figure 9.

![Figure 9: Injection and mixture formation processes for the 12-hole injector at injection timing of 270°CA after induction TDC, 80bar injection pressure and no in-cylinder swirl.](image)

![Figure 10: Injection and mixture formation processes for the 6-hole injector at injection timing of 270°CA after induction TDC, 80bar injection pressure and no in-cylinder swirl.](image)
The same tendency was found in the 6-hole injector. In details, late injection of fuel causes the mixture cloud to remain in the centre of the cylinder, causing undesirable stratification. The reason is mainly the compact, narrow spray shape produced of the 6-hole injector, coupled with the almost stagnant in-cylinder air motion that late crank angles, like 320°CA, dictate. On the other hand, combination of early injection during compression stroke and single injection strategy produces very similar results. Medium timings produced once again acceptable stratification as seen in Figure 10.

**In-cylinder charge motion**

Increased in-cylinder swirl is expected to homogenise the mixture rather than to produce the desired stratification. This behaviour has been confirmed from the LIF results for both injectors. As mentioned before, stratification is achieved by injecting fuel during the compression stroke and after the closure of the intake valves. Even though the valves are closed, swirling motion is preserved in the cylinder losing some of its strength. Additionally, it has been proven that sprays from multi-hole injectors are rather insensitive to in-cylinder air motion due to their increased penetration momentum. Therefore, in-cylinder swirl under these conditions is not strong enough to affect the spray development. Moving one step forward, after the fuel has been completely vaporised, with swirling motion still present in the cylinder, the result would be homogenisation of the charge rather than stratification (Figure 11).

**Figure 11:** Effect of in-cylinder swirl on charge stratification, in the case of 6-hole injector and 270°CA injection timing.

4. **Conclusions**

A research program has been conducted with the aim of exploring the potential of high-pressure multi-hole injectors in creating homogeneous as well as stratified mixture distributions in a five-valve twin-spark ignition engine. A motored single-cylinder optical engine was used, with a fully optical quartz liner, which featured a centrally mounted injector and two spark-plugs located at opposite sides of the cylinder. Laser induced fluorescence was employed to reveal fuel concentration and local air/fuel ratio measurements along a central plane of the cylinder, with a 6.5mm offset relative to the spark-plug plane.

The twin-spark combustion concept featuring central injection is generally offering advantages in HC and smoke emissions relative to its side-injection, wall-guided counterpart. Additionally, the previously observed problem of cocking in multi-hole injectors can be minimised due to reduced injector tip temperatures, as a result of twin-spark combustion. A second advantage of having two spark-plugs installed is the reduced danger of spark-plug fouling present in most spray-guided configurations. This is also ensured by the stable spray structure of multi-hole injectors and the
flexibility they offer in nozzle design, which helps in matching closely the spray shape to engine specifications. The characteristics of multi-hole sprays, carefully evaluated recently, is the increased spray tip penetration relative to the swirl or hollow cone sprays from outwards opening nozzles. This can, unfortunately, lead to extensive impingement of the fuel on cylinder walls and piston. Although there are ways for controlling this characteristic; such as careful selection of injection timing, there will always be a small percentage of fuel impinging on the piston. On the other hand, the increased penetration momentum can make multi-hole sprays less sensitive to the in-cylinder air motion. This leads to simpler engine management system eliminating the need for accurate control of the airflow. Nevertheless, results presented here have shown that airflow is vital for effective mixing and improved homogeneity implying that a compromise may be needed for complete combustion.

Overall, multi-hole injectors have demonstrated their ability to produce homogeneous and stratified charge in a multi-valve twin-spark ignition engine. It has been confirmed that injection timing controls wall impingement, resulting in quick and efficient evaporation for homogeneous engine operation. It is also the major parameter in achieving stratification by determining the time of piston impingement, which directs the mixture cloud towards the spark-plugs. Finally, in-cylinder swirl was shown to improve homogeneity significantly but to be an obstacle to any attempts for creating charge stratification. In concluding, it should be stated that combustion tests which represent the next stage in the research programme will conclusively determine whether the proposed combustion concept stand a chance to reach production.

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