AN EXPERIMENTAL DATABASE ON THE EFFECTS OF SINGLE AND SPLIT INJECTION STRATEGIES ON SPRAY FORMATION AND SPARK DISCHARGE IN AN OPTICAL DISI ENGINE FUELLED WITH GASOLINE, ISO-OCTANE AND ALCOHOLS

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ABSTRACT

This paper presents results from a comprehensive optical study of a direct-injection spark-ignition research engine running on gasoline, *iso*-octane, ethanol, *n*-butanol and E10 fuels injected from a multi-hole injector located centrally in the combustion chamber. The analysis was based on images of spray development, spark discharge and combustion to understand the effect of injection strategy early and late in the cycle on in-cylinder phenomena. Specifically, ‘single’ injection strategies at different timings from early to late intake stroke, as well as multiple ‘split’ injection events with triple pulses in the early intake stroke or double pulses in the intake stroke and late compression stroke were investigated. The engine was run at 1500 RPM at part and full-load conditions (0.5 bar and 1.0 bar inlet plenum pressure, respectively). Engine coolant temperatures of 20–90 °C were employed to understand how the fuel’s volatility was related to the phenomena observed. The sprays were imaged for a series of cycles primarily by Laser-sheet illumination on one vertical and two horizontal planes to identify three-dimensional aspects of the spray’s development and its interactions with the incoming flow, valves, piston and liner. There was a clear trade-off between the earliest single injection timings in the intake stroke that reduced valve interactions but increased piston impingement and the latest single injection timings towards the end of the intake stroke which reduced piston impingement but increased valve interactions and resulted in less time for fuel evaporation and mixing before ignition timing. The triple injection strategy eliminated effects related to impingement. The spark discharge was also imaged for a series of cycles with all injection strategies and clear differences were observed. Selective combustion imaging provided insights into the flame’s growth and motion with early and double early-late split injection strategies. The double early-late injection strategy demonstrated the potential for control of the mixture formation and flow field over the early flame development stage of combustion.

ABBREVIATIONS

AFR	Air-to-Fuel Ratio
AEOI	After End of Injection
AIT	After Ignition Timing
ASOI	After Start Of Injection
ATDC	After Intake Top Dead Centre
BDC	Bottom Dead Centre
BTDC	Before Compression TDC
BMEP	Brake Mean Effective Pressure
CA	Crank Angle
COV	Coefficient Of Variation (Standard Deviation/Mean)
DISI	Direct Injection Spark Ignition
EOI	End of Injection
ETU	Engine Timing Unit
EVC	Exhaust Valve Closure
EVO	Exhaust Valve Open
FFID	Fast Flame Ionisation Detection
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closure
IVO	Intake Valve Open
MFB	Mass Fraction Burnt
PIV	Particle Image Velocimetry
PLIF	Planar Laser Induced Fluorescence
RMS	Root Mean Square (standard deviation)
RON	Research Octane Number
RPM	Revolutions Per Minute
SI	Spark Ignition
SOI	Start of Injection

INTRODUCTION

The Direct-Injection Spark-Ignition (DISI) engine has several practical advantages over the port-fuel injected engine, such as greater precision in fuel metering and significant potential for better fuel economy, as well as larger charge cooling potential that allows higher compression ratios and increased thermal efficiency. However, successful implementation of all these benefits has been more difficult to achieve in practice than originally expected [1]. The main reason behind this has been the early-design direct-injection fuelling systems that failed to atomise the fuel fully in the time available before ignition and avoid fuel wall impingement; this produced non-optimal levels of tailpipe emissions. Latest design high-pressure multi-hole injectors located in close spacing arrangement with the spark plug aim to resolve these issues by employing a number of nozzle holes which can be designed to target the fuel delivery where necessary inside the cylinder and by flexibility of when and how fuel is injected into the cylinder that allows, for example, independence over the piston's position [2–4]. The majority of work published to date on multi-hole fuel nozzles concerns Diesel injectors. However, the different fuel properties and demands of the DISI combustion system merit specific attention. DISI injectors must inject fuel at conditions of low in-cylinder pressure, typically from ~0.2 bar at low load with early injection strategies for homogeneous mixture formation, to ~5 bar for late-injection strategies under stratified engine operation, or even more under supercharged operation; furthermore, fuel temperatures can vary from below 0 °C when at cold-start engine conditions to over 150 °C at the injector tip under high-load firing conditions.

Although quite significant phenomena are coupled over this operation regime, very few in-cylinder experimental results exist to explain such effects that can alter the shape and penetration of the spray, especially with a variety of fuels at the same conditions. Similarly, the effect of various multiple, *i.e.* ‘split’, injection strategies on in-cylinder spray formation and combustion has not been reported extensively in the literature. Such strategies have great potential to deal successfully with the peculiarities of the atomisation process of different fuel types and the requirements for different levels of mixture homogeneity and/or stratification. Most of the previous work of this type has primarily focused on the effect of such strategies on exhaust emissions [5], cold-start improvement [6], combustion work output [7], *etc.*, with no detailed comparison of the in-cylinder sprays themselves with different fuels and at different engine operating conditions. Primarily, spray work has been carried out in quiescent injection chambers [8–14]. It also needs to be mentioned that despite the routine use of spark-ignition systems in gasoline engines, our understanding of the physics of ignition and flame initiation is surprisingly poor, especially with different injection strategies in DISI engines. Observing spark discharges and early flame kernel development *in-situ* can be key to the development of greener engines, especially with injection strategies that can promote different degrees of homogeneity and, in the process, also influence the in-cylinder flow field and interact directly with the spark itself.

PRESENT CONTRIBUTION

The main objective of the current work has been to visualise and study the mechanism of spray formation in-cylinder and its potential effect on spark discharge in the context of injection strategies that can promote optimised mixture formation under various engine running conditions. A range of strategies, including ‘single’ injection pulses per cycle during various timings in the intake stroke, triple ‘split’ injections per cycle for better mixture homogeneity, as well as double ‘split’ injections in the intake and late in the compression stroke for mixture stratification. Gasoline, *iso*-octane, ethanol, *n*-butanol and E10 (blend of 90% gasoline with 10%

ethanol) fuels were used in order to decouple effects related to fuel type and volatility at both low-load and full-load engine operating conditions. Analysis of these data is done in the current paper in conjunction with in-cylinder flow and fuel concentration data that have been obtained in three optical engines of identical geometry and at the same operating conditions using Particle Image Velocimetry (PIV) [15–17], Planar Laser Induced Fluorescence (PLIF) [18, 19] and in-cylinder Fast Flame Ionisation Detection (FFID) at the spark-plug location and pentroof wall at ignition timing [7].

To the best of the authors' knowledge, this is the first time that such a complete set of spray data is presented for several fuels at comparable conditions over many variables in a latest geometry DISI combustion system, and discussed as a whole and against existing data of in-cylinder flow and fuel concentration obtained from optical engines of nominally identical geometries. A number of horizontal and vertical in-cylinder planes were imaged by Laser-sheet tomography to provide extensive information on three-dimensional effects. This is an extension of fuel spray research with the same injector reported earlier in-cylinder [3, 7, 20–22], as well as in a quiescent chamber [23–27], and makes an attempt to bring together in one document a range of detailed spray development data for multi-component and single-component fuels, statistically analysed. It was the authors' intention from the outset to publish and discuss these data in one entity rather than in a series of individual papers, hence it is acknowledged that this paper may be longer than usual. However, it is believed that this approach can be more useful for the engine research community and it contributes notably towards a public database of in-cylinder observations which are essential for developing our knowledge of the underlying fundamental mechanisms under realistic engine conditions.

Such experimental data are also essential for modellers because accurate simulation of sprays and mixture preparation with fuels of different properties in DISI engines is still very challenging, especially if validation is sought over a range of injection strategies and engine operating conditions. In that respect, all nominal angles of the injector's holes are also provided in the current paper, a scarcity in existing literature.

Moreover, experimental spray observations presented in terms of mean and standard deviation maps on various in-cylinder planes can be valuable for engine modellers, especially for validation of numerical predictions of cyclic variations that can be performed over a series of consecutive engine cycles by latest large eddy simulation codes with existing computing power. Such codes have focused primarily on homogeneous mixtures using gaseous fuels, *e.g.* propane in [28–31], hence any future predictions of cyclic variations that will include spray formation can benefit greatly by a database of in-cylinder sprays.

Previous work has offered some useful insights into the spark's behaviour in DISI engines [32–35] and the present paper also aimed at reporting on the spark's behaviour in conjunction with the same set of injection strategies employed for spray development analysis.

EXPERIMENTAL APPARATUS AND PROCEDURES

RESEARCH ENGINE

A single-cylinder optical research engine was used to investigate the in-cylinder spray behaviour under realistic operating conditions. The engine employed a 4-valve pentroof head based on a V8 4.5L prototype four-stroke DISI engine, with a vertical centrally mounted multi-hole direct-injection system in close-spacing arrangement with the spark plug. The valvetrain was comprised of double-overhead cam-shafts with direct acting lobes. The engine valve timings were set to give a small negative valve overlap by late Intake-Valve Opening and Closing

(IVO and IVC, respectively) and early Exhaust Valve Opening and Closing (EVO and EVC, respectively). The geometric parameters of the engine are summarised in **Table 1**. This table also contains the engine operating conditions used throughout this paper, as will be detailed later. With regards to intake geometry, the engine was fitted with an ‘inverted-U’ tumble-inducing manifold upstream of the inlet ports. For the purposes of the current paper, the engine was run with a fully optical liner and an optical piston crown in motoring mode and with a metal water-cooled liner and optical piston crown in firing mode. An in-house designed water system was used to circulate cooling water round the engine head and cylinder liner and incorporated a temperature control system comprising of a three-way valve to divert water to a heat-exchanger and immersion heater. A Eurotherm 3216 PID controller was used to select the temperature set-points and calibrate the temperature control dynamics. The main influence of engine temperature was observed by using engine head coolant temperature of 20 °C and 90 °C. Some measurements were also acquired at 50 °C and 70 °C as discussed later. Enough heat-soak time (30–40 min) was allowed so that the engine acquired sufficient thermal inertia at the different settings. It is noted here that the terms ‘engine head temperature’ and ‘fuel temperature’ are used interchangeably in the text that follows, as changing and controlling the engine head coolant temperature was essentially a means of altering the fuel’s temperature relevant to its distillation curve or boiling point. This follows the practice used in previous publications to link engine data and quiescent injection chamber data and it is also employed here for consistency, *e.g.* see [3, 21, 23].

The ignition system used was a transistorised coil-on-plug type driven via a 12 Volt DC, 25 A power supply which also powered the injector driver. A commercial J-type ground electrode with a V-grooved central electrode, NGK laser-iridium spark plug inclined at ~15° from vertical was used (NGK IKAR6_IX9), having a heat rating of 6 and a 0.9 mm spark gap. An ignition charge time of ~4 ms was used. The ignition and injection timings were controlled using an AVL 427 Engine Timing Unit (ETU) which generated crank-angle or time-based controlled trigger signals. The ETU required two inputs to define a ‘clock’ term and a ‘reference’ signal term. Both signals were supplied from the output of an optical encoder placed on the engine crank-shaft (Leine-Linde model 503) with 1800 pulses/revolution and a once/revolution TDC signal. The ETU used the TDC signal to reset its internal clock every crank shaft revolution and thereby maintain optimum synchronization accuracy on a cycle-by-cycle basis. The A/F Ratio (AFR) was measured using a heated zirconia-based high-speed oxygen sensor (ECM AFR Recorder 1200) installed in the exhaust manifold ~150 mm along the exhaust pipe downstream the engine head. The sensor allowed programming of the H-C and O-C ratios to accommodate different fuels and was calibrated for oxygen sensitivity using compressed gas bottles. Other details about the engine test bed arrangement can be found in previous publications by the current authors [3, 20].

INJECTOR AND FUELS

A six-hole injector designed for vertical installation in the engine head in close spacing arrangement with the centrally located spark plug was used for this investigation. The injector had six nozzle holes in an asymmetric arrangement with different angles with respect to the vertical axis. **Fig. 1** shows a schematic of the injector and its spray plumes through two views; all projected nozzle-hole angles are included. Plumes #1 and #6 have been designed to pass on either sides of the spark plug *i.e.* one at the intake side and the other at the exhaust side of the engine. More details about the injector’s geometry and spray formation in a quiescent injection can be found in previous studies [23–27, 36]. For all work presented in this paper the fuel pressure was maintained at constant

150 bar supplied by a pneumatic Heypac GX40 pump, a check-valve (Swagelock SS-4CPS-1) and a 2 μm ceramic element fuel filter (Swagelock SS-4FW4-2). Two fuels were primarily investigated, gasoline (RON95) and *iso*-octane. A standard commercial grade European gasoline contains several hundred hydrocarbons, typically about 25–30% C_5 or lower, 30–40% C_6 – C_8 and the remainder C_9 – C_{10} hydrocarbon chains. *Iso*-octane is a single component of gasoline with a boiling point of 99 °C when at atmospheric pressure. Two alcohols were also studied, ethanol and *n*-butanol. Ethanol boils at ~78 °C and *n*-butanol at ~117 °C. In order to investigate the combined effects of ethanol in gasoline at levels that have become recently commercially available, an imaging set was also acquired with a blend of 90% gasoline with 10% ethanol (E10); this was supplied in blended form by a petroleum company, *i.e.* it was not splash-blended by the authors. **Table 2** summarises some important physical and chemical properties of the fuels used. The exact distillation curve of the gasoline and E10 fuels employed in the current study are shown in **Fig. 2**. This figure also shows the boiling points of some of the main single components present in gasoline, as well as those of the two alcohols tested.

INJECTION STRATEGIES

The baseline injection strategy used a single injection event with timing set early in the intake stroke to promote homogeneous mixture formation. The duration was set to fit the required stoichiometric AFR as observed by the lambda sensor. For this strategy, the effect of injection timing over the intake stroke was studied with Start Of Injection (SOI) set to 40°, 60°, 80°, 100°, 120°, 140° and 160° ATDC, with SOI=80° CA ATDC selected as the optimum one as will be discussed later. Injector characterisation by [24] showed that the needle response time and the appearance of first fuel at the nozzle tip for 100 consecutive injections was very stable at about $300 \pm 5 \mu\text{s}$ after the rising edge of the TTL trigger signal. The minimum pulse duration was also found to be ~200 μs before any fuel was actually injected, thus setting a minimum limit on the different strategies that could be applied. For a strategy of three ‘split’ injections in the intake stroke at 1500 RPM stoichiometric operation, the pulse-widths used were 0.3 ms, 0.3 ms and 0.32 ms in duration, with SOIs of 60°, 70°, 80° CA ATDC, respectively. It is noted that the last injection pulse was intentionally selected to start at the same time in the intake stroke of the cycle as the single injection strategy of SOI=80° CA ATDC. Finally, a hybrid homogeneous/stratified injection strategy was devised to try to control the flow field and mixture around ignition timing. This consisted of injecting ~75% of the fuel in the intake stroke at the same timing as the standard single injection strategy (SOI=80° CA ATDC) and the final 25% of the fuel injected just before spark timing (SOI=45° CA BTDC). It is noted that the double hybrid strategy was not only related to control of mixture stratification at ignition timing but also relevant to ‘catalyst heating operation’ [34–35]. The typical pulsewidths for stoichiometric gasoline low-load operation (0.5 bar inlet plenum pressure) are summarised in **Table 3** for all injection strategies. For both alcohols the injection pulsewidth had to be increased due to their much lower stoichiometric AFR than gasoline’s (**Table 2**). Specifically, 1.3 ms was typically used for ethanol with the single-injection strategy and 1.1 ms for butanol; *iso*-octane’s pulsewidth was marginally smaller than gasoline’s due to their similar value of stoichiometric AFR.

EXPERIMENTAL TECHNIQUES

The experimental configuration consisted of in-cylinder pressure acquisition and high-speed imaging; typically 100 cycles were used for imaging studies but 200–300 cycles were used for in-cylinder pressure mapping purposes. Each of the techniques and methodologies used are presented individually in the following sections. The baseline engine condition approximated to a well-known reference engine mapping point, referring to low-

load, low-speed inner city driving conditions of 1500 RPM with 0.5 bar intake pressure. The spark advance was set to 35° CA for all fuels in an attempt to fix the nominal mean in-cylinder flow and thermodynamic conditions related to piston's motion at ignition timing. However, the engine was mapped for all fuels at ignition timings in the range 50°–30° CA BTDC and the reader may refer to [7] for further details if needed; the fixed 35° CA advance was a reasonable compromise for all fuels within the objectives of this paper.

In-Cylinder Pressure Measurement and Analysis

The in-cylinder pressure measurements for the present study were conducted with a water-cooled piezoelectric pressure transducer (Kistler 6041A). The sensor was installed flush with the engine pent-roof walls and was connected to a Kistler 5011B10 desktop charge amplifier. Absolute pressure referencing was done by employing another pressure transducer installed at the bottom of the cylinder barrel so that when the piston was close to BDC the transducer recorded the absolute in-cylinder pressure. Specifically, a water-cooled piezoresistive pressure transducer (Kistler 4075A10) was mounted on the exhaust side of the cylinder 75 mm below the head gasket plane. This also required amplification and a Kistler 4618A0 amplifier was employed. When the piston was above the barrel transducer the pressure measured was atmospheric as a result of the 'floating' cylinder design of extended piston optical engines.

The output signals of the amplifiers were digitised at a sampling rate of 45 kHz on a cycle-to-cycle basis with a 12-bit analogue-to-digital converter (National Instruments PCI-MIO-16E-4) using Labview interfacing. This rate corresponded to digitisation every 0.2° CA at 1500 RPM. The uncertainty due to electrical interference was a maximum of 0.05 % of the full scale value for the in-cylinder pressure and 1% of full scale value for the intake plenum and barrel pressures, corresponding to an uncertainty of ± 5.0 mbar and ± 10 mbar respectively. The crank angle pulse train from the crankshaft encoder was used via the AVL ETU unit to provide the clock source for the Labview data acquisition program. The pressure data were post-processed to calculate the Indicated Mean Effective Pressure (IMEP) and Mass Fraction Burned (MFB). Details about the processing methods and related uncertainties can be found in [7, 37].

In-Cylinder Spray Imaging

The test matrix involved Mie scattering imaging of the spray as this penetrated a laser-sheet 0.5 mm thick positioned horizontally (*i.e.* on the 'swirl' plane, entering the combustion chamber through the pentroof window), as well as vertically (*i.e.* on the 'tumble' plane, entering the combustion chamber through the 45° mirror and optical crown). The illumination source was a New Wave Research Pegasus Nd:YLF laser at 527 nm. This consisted of a diode-pumped high-repetition double cavity unit (10 kHz maximum Q-switch frequency per cavity) and was run with both heads firing simultaneously at 9 kHz. The laser pulse was essentially used to 'freeze' the imaged spray during the injection period. A Photron APX-RS high-speed camera was used to obtain images with 640×480 pixel resolution with a typical frame rate of 9 kHz, *i.e.* 1 °CA resolution at 1500 RPM. The camera was triggered externally with a TTL pulse supplied by the ETU and 100 consecutive cycles were imaged with up to 60 images/cycle. The acquired images were stored in 8-bit resolution. More details about the imaging equipment can be found in previous studies by the current authors [3, 20].

An image of the optical crown arrangement is shown in **Fig. 3**. The same figure also illustrates Laser-sheet images of the spray's development in the engine at 'static' engine conditions, *i.e.* into the engine cylinder at 0 RPM, with the convention of plume numbering clearly indicated (see also the schematic of **Fig. 1** for reference).

The horizontal locations of laser-sheet illumination have been included in the tumble plane images of row 1 of **Fig. 4** at running engine conditions of 1500 RPM, 0.5 bar intake pressure; they are illustrated by the green lines. Specifically, horizontal laser-sheet spray imaging inside the pent-roof (whose location is illustrated by the top horizontal green line in row 1) is shown in row 2. This plane allowed spray/valve-flow interactions to be captured particularly for plumes #1 and #6, but also illumination of the spray footprint produced by plumes #2, #5, #3 and #4 (plane equivalent to the static conditions shown in **Fig. 3**). The second in-cylinder horizontal plane view is shown in row 3 of **Fig. 4**. This was used to provide an insight into spray plume break-up as the spray penetrated further into the cylinder and interacted with the bulk in-cylinder flow motion. The location of this plane has been illustrated by the second green line from the top in the first image of row 1. The tumble plane images of row 1 in **Fig. 4** were obtained by laser-sheet illumination across the centreline of the injector, as defined by the vertical line shown in the first image of row 2. This was to image the vertical development of spray plumes #2 and #5. The results will be discussed on the basis of images from typical single cycles like those shown in **Fig. 4**, but also from mean (*i.e.* average) and RMS (*i.e.* standard deviation) images with an intensity range scaled as necessary to allow direct comparison of behaviour between fuels and conditions. This approach was selected because it could highlight the spray-flow interactions and their cyclic variability in accordance with the objectives of the work, and in contrast to the spray-tip penetration data quantified with the same injector in a quiescent injection chamber [23–25]. In this context, arrows have also been drawn on the tumble and pentroof swirl images of **Fig. 4** which represent the mean flow patterns as have been quantified on the same planes by PIV. The respective flow data have been presented in detail elsewhere [15–17] and are only used here to aid interpretation of our spray imaging observations. The three aforementioned planes of spray imaging were studied primarily with gasoline and *iso*-octane fuelling. Then the laser light was diffused through lenses before entering the cylinder from the side pentroof window and ‘global’ spray formation imaging was performed through the optical piston crown for all fuel types: gasoline, *iso*-octane, *n*-butanol, ethanol and E10.

Spark Discharge Imaging

One disadvantage of imaging on the horizontal (or ‘swirl’) planes was that it did not provide adequate information about the spark event, particularly since the current engine used tumble flow conditioning rather than swirl, which increased the vulnerability of the spark to flow field conditions at ignition timing. The spark was therefore imaged initially from an isometric view, angled at $\sim 60^\circ$ to the horizontal. In order to access this region of the combustion chamber the high-speed camera was positioned at the opposite side of the pent-roof window (under the pressure-sensor location) and optical access was achieved using a quartz annulus ring. The sparks were imaged sequentially for 200 cycles under static and motoring conditions at low-load and full-load conditions (0.5 bar and 1.0 bar intake pressure, respectively). These tests were done without the addition of fuel to provide benchmark insight into the effect of in-cylinder flow on the spark discharge event and its cyclic variability. The data will be discussed by analysis of mean and RMS images. A selected number of tests were also carried out by imaging the spark from the side pentroof window (*i.e.* on the tumble plane) for fuelled cycles using all three injection strategies (single, triple, and double hybrid). These images were obtained with simultaneous in-cylinder pressure sampling to allow conditionally averaged statistical analysis. Finally, selective imaging of combustion was performed through the piston crown for the single and double hybrid injection strategies to study the flame kernel’s growth and motion. The temporal and spatial resolution was nominally the

same to that used for spray imaging and in-cylinder pressure was simultaneously acquired. The image processing methods for quantification of the flame's growth and motion have been described in detail elsewhere for various fuels and are not repeated here for brevity [3, 20, 37, 38].

RESULTS AND DISCUSSION

The results are presented in a sequence of main sections as follows:

- Firstly, the mechanism of spray formation of the single injection strategy with SOI=80° CA ATDC is discussed using gasoline and *iso*-octane fuels at 20 °C engine head temperature, 0.5 bar intake pressure.
- Secondly, the effect of fuel volatility is discussed with the same single injection strategy by comparing gasoline and *iso*-octane sprays at 90 °C engine head temperature, 0.5 bar intake pressure.
- Thirdly, the effect of timing the single injection in the intake stroke and up to intake valve closure with SOI=60–220° CA ATDC and 0.5 bar intake pressure is discussed with relevance to spray impingement on piston crown and liner surfaces at different engine head temperatures.
- Fourthly, the effect of engine load on spray formation is discussed using the single injection strategy with SOI=80° CA ATDC by comparing spray images at 0.5 bar and 1.0 bar intake pressure.
- Fifthly, the effect of single and triple split injection strategy is discussed with focus on spray development and wall impingement at 20 °C and 90 °C engine head temperature, 0.5 bar intake pressure.
- Sixthly, the effect of alcohol fuelling on spray formation is compared directly to gasoline and *iso*-octane sprays at 20 °C and 90 °C engine head temperature, 0.5 bar intake pressure.
- Finally, the effect of injection strategy on spark discharge and combustion is discussed, with particular focus on gasoline and *iso*-octane fuelling at 0.5 bar intake pressure.

SINGLE INJECTION STRATEGY – SPRAY FORMATION AND FLOW INTERACTIONS

A comparison of the instantaneous spray images on the horizontal plane shown in **Figs. 3–4** illustrates the effect of the incoming 'valve jet' flow. It is clearly seen that spray plume development is affected almost immediately after the start of injection and that the spray plume break-up has significantly improved by ~10° CA ASOI (in comparison to static engine conditions). The initial spray/flow interaction at 5° CA ASOI (85° CA ATDC) was such that the tip of spray plume #6 was visibly deflected towards the bottom of the image, corresponding to the exhaust side of the combustion chamber. Plume #1 also interacted with the spark-electrode as it was deflected towards it by the valve jet. Clearly the characteristics of the flow over the intake valves formed an important factor in the spray's break-up. To understand the incoming flow, the pressure wave dynamics in the intake manifold had to be firstly understood. Information about the flow conditions in the intake manifold and inside the pent-roof of this engine was available from PIV work [15, 16]. Specifically, the PIV work had shown that the strength and location of several in-cylinder flow features, such as the vortex located just above the piston crown on the intake side of the combustion chamber (see **Fig. 4**), varied significantly from cycle-to-cycle. To illustrate the effect of intake flow variability, **Fig. 5**, shows a series of spray images on a cycle by cycle basis. Considering that spray formation imaging under quiescent chamber conditions has shown very small injection-to-injection variations [24], **Fig. 5** suggests that the incoming flow variations were responsible for the differences observed in the spray images and the variability in the levels of piston impingement and plume break-up. This is discussed further in the following subsections for the different planes of in-cylinder spray imaging.

Tumble Plane

The tumble plane mean and RMS spray images are shown for low-load motoring conditions at 20 °C in **Fig. 6**. In the first mean image, shown at 3.6° CA ASOI (83.6° CA ATDC), spray plumes #2 and #5 can be clearly seen exiting the nozzle and saturating the image (indicated by the high intensity red regions). This initial saturation was a result of the high density of liquid fuel emerging out of the nozzle and was tolerated in these images because good plume definition was still possible at this stage and sufficient intensity was also required later as the spray broke-up and scattering levels were significantly reduced (the laser source power was kept the same throughout the experiments to aid interpretation). The downward momentum of the spray plumes was much higher than the flow over the intake valves and therefore the imaged plumes for both fuels initially showed little interaction with the incoming airflow. In the regions of the spray tips however, there were clear cycle-to-cycle variations due to the more exposed liquid fragments and the direct effects of aerodynamic interactions and drag. The first clear spray/flow interactions can be seen for *iso*-octane for the plume on the exhaust side (spray plume #5) where the red region at the spray tip clearly thinned out into a channel and began to break up, as the intake valve flow tried to deflect the spray across the pent-roof. For gasoline this feature was not present, most probably because the spray had already atomised and broken-up faster at the spray tip, indicated by the lower pixel intensity levels and wider areas. Reference to the RMS images confirmed higher levels of intensity variability for gasoline than *iso*-octane at the spray leading edge, suggesting that the spray in this region was already more finely atomised (on the assumption that both fuels experienced nominally the same flow-field over the test run). This observation was considered to have roots in a combination of factors, ranging from variation in the type of fluid flow phenomena inside the injector nozzle itself to the different liquid transport properties of each fuel. In general however the initial spray penetrations were very repeatable for both fuels and there were only small differences in penetration (~1 mm), plume widths (< 1 mm) and cone angles (< 1°), with peak COV in the spray tip intensities of ~35% for gasoline and ~20% for *iso*-octane. The intensity scale is shown in % of the full 8-bit range (0–255), hence the colour map in the RMS images allows an insight into the cyclic variations of the spatial intensity.

As injection progressed, the sprays became fully formed and the features observed at 3.6° CA ASOI became more pronounced. At 7.2° CA ASOI, roughly in the middle of the injection event, the mean images still showed very similar features between the two fuels inside the pent-roof, although the intensities for gasoline were generally lower than those of *iso*-octane, particularly for the right plume (#5), again indicating better overall break-up. In terms of spray tip penetrations, plume #2 (below the intake valve) showed very similar average values for both fuels $\sim 45 \pm 2$ mm along the plume axis; however, plume #5 on the exhaust side showed a lower penetration and a thinner, more compact plume for *iso*-octane than for gasoline. In fact, the RMS image – which represents more clearly the full envelope of liquid presence on a cycle-to-cycle basis – showed a higher spray tip penetration by ~10 mm for gasoline. This higher penetration for gasoline was mainly attributed to the spray tip beginning to detach, ahead of the main spray as the effect of strong air entrainment along the surface of the plume began to break-up it up. It appeared that recirculation vortices around the spray tips caused by shear motion, combined with bulk flow structures around the plumes, promoted air entrainment into the plume core, which in turn gradually thinned the plume until the spray tip was completely detached. These characteristics were also prevalent for gasoline's plume #2 and clear in its RMS image. The spray images for *iso*-octane

suggested that these flow interactions were increasing the rate of evaporation of this fuel noticeably as these features could not be observed due to lower Mie-scattering levels, consistent with the expected relationship between Mie-scattering intensity and droplet field surface area. Since the bulk flow might not be expected to change significantly by the injection of a different fuel, it was clear that even subtle changes in the fuel transport properties could have a noticeable effect on the local in-cylinder spray break-up characteristics and initial mixture preparation.

At $\sim 11^\circ$ CA ASOI the injection event ended (0.9 ms pulse duration) and the spray completely detached from the nozzle. The spray was now fully airborne and its momentum was quickly reduced. The bulk in-cylinder flow continued to break-up the spray into smaller ligaments and droplets and both the mean and RMS images showed a very similar envelope of Mie-scattering intensity for *iso*-octane and gasoline at this stage, albeit with clearer differences in the intensity levels of the RMS images. For gasoline the exhaust side plume #5 exhibited no discernable ‘intact’ core and significant impingement already on the piston crown surface, compared to *iso*-octane where some spray plume structure was still visible. A similar trend was seen on the intake side for plume #2, where flow interactions appeared to be aiding the break-up of the gasoline plume in a more pronounced manner resulting in a wider area with RMS levels $\sim 10\%$ larger than for *iso*-octane. However, it is believed that faster evaporation of the atomised *iso*-octane droplets contributed to these differences, as described in the preceding paragraph. ‘Flapping’ of the spray plumes from cycle-to-cycle also resulted in larger regions of high RMS along the plume surfaces for both fuels, in particular under the intake valves for plume #2.

After the End Of Injection, specifically at $\sim 14^\circ$ CA AEIOI, it was confirmed that different levels of evaporation were indeed taking place for the two fuels as a result of their different volatilities. The mean image for *iso*-octane showed significantly lower intensity levels than gasoline by $\sim 4\text{--}6\%$ with corresponding lower levels of RMS intensity, indicative of smaller surface area which could only stem from smaller or fewer droplets participating in the scattering process. It was possible to speculate therefore that the reduced spray tip penetration observed for the exhaust plume of *iso*-octane was attributable to the removal of the spray leading edge and/or faster entrainment and evaporation of the spray tips. This would also be compatible with the strong tumble flow present in this region of the cylinder, observed by PIV in [15, 16]. The higher intensity droplet field for gasoline at 25° CA ASOI indicated that the wide range of gasoline’s volatility was probably responsible for the lower evaporation rates, *e.g.* associated with components such as toluene (boiling point 110°C), xylenes (boiling point $\sim 140^\circ\text{C}$) and other heavy fractions which together made up nearly 27% of the gasoline blend used by volume. This highlights the importance of the distillation curve in helping to interpret the potential of a gasoline blend to perform well in a DISI engine, especially under cold start conditions.

Overall the local interactions of the sprays with vortices in the bulk flow were observed to play an important role in the mechanism of initial mixture preparation because they affected how quickly and reliably the sprays were atomised and entrained into the bulk flow, where complete evaporation could then take place more efficiently. Such interactions also governed to a large extent the levels and probability of liquid impingement on in-cylinder surfaces, such as the piston crown (demonstrated in **Fig. 5**) and the cylinder walls, *i.e.* phenomena that can have an adverse impact on robust mixture preparation.

Pent-Roof Swirl Plane

The swirl plane images of **Fig. 7** show the spray footprint that was produced as the plumes penetrated through the horizontal laser-sheet that was positioned 1 mm inside the pent-roof (see **Fig. 3** for reference). The temporal resolution used for these images was higher than for the tumble plane because the spray penetrated very quickly through the sheet; spray images were thus produced for the first consecutive crank angle degrees after first contact with the sheet, 4°–6° CA ASOI, in order to provide a detailed illustration of the transient nature of spray/flow interactions occurring inside the pent-roof. Two final images at 8° and 14° CA ASOI are also shown to illustrate the changes in the spray's development mid-point through the injection event and to compare the Mie-scattering intensities after the end of injection. The mean and RMS images are shown with similar colour scales to those used for the earlier tumble plane images, with the mean images' lowest 10% intensity bins set to black (*i.e.* classified as 'background'), to facilitate visual interpretation of the spray on the laser-sheet plane and reduce out of plane scattering contributions. The RMS images had the colour scale compressed to show 30% of the full 256 intensity range for optimum contrast.

The plumes visualised in the images at 4° CA ASOI were plumes #2, #3, #4 and #5 in an anti-clockwise direction (refer to **Fig. 1** and **Fig. 3**). Plumes #1 and #6 had a shallow nozzle exit angle directed towards the spark-plug and therefore took longer to reach the light-sheet. It was immediately clear that, in agreement to the tumble plane images, the plume penetrations for both fuels were generally quite similar on the basis of the average size of their footprints (red areas) at 4° CA ASOI, although marginally larger for gasoline particularly for plumes #2 and #3 under the intake valves. The area around the four visible plumes was also noticeably brighter for gasoline (more blue colour), which indicated the presence of more or larger droplets (*i.e.* liquid phase) around the peripheries of the plumes contributing to added scattering. With respect to cyclic variations, both fuels showed similar areas of variability around the core of the plumes, with values reaching 30% of the mean in both cases, *i.e.* levels very similar to those produced on the tumble plane. For gasoline the edges of the plumes under the intake valves showed some 'streaks' in the RMS image that could be mistaken for different features in the break-up of these plumes. However, investigation of the sequence of raw images showed this to be an artefact of the cold motoring engine operating conditions and the imaging arrangement which had a relatively large depth of field. Careful examination of fuel residue of the piston crown after testing showed that fuel films had spread over the piston surface towards the centre of the piston crown window. For gasoline the effect was worse on the intake side than the exhaust side, but was not seen for *iso*-octane, supporting the observations made earlier from the tumble plane images. While tumble images already indicated higher levels of piston impingement for gasoline, it is believed that this effect on the horizontal plane images was particularly noticeable for gasoline due to its multi-component nature; *e.g.* the fuel films left on the piston crown were 'oily' and more viscous than those of *iso*-octane. This optical effect was seen for all the subsequent RMS images of gasoline at 20 °C on the same side of the engine chamber.

As the spray penetrated further, 5° CA ASOI, the main four plumes were no longer as symmetrical and began to be noticeably affected by the intake flow over the valves and the swirling flow in the cylinder. The spray plume footprints looked clearly distorted and there was a more progressive intensity gradient from the centres of the plume footprints to their peripheries; this was also clear from the RMS images. Plumes #1 and #6 also became visible, albeit still above the laser sheet as indicated by their low intensity levels. Plume #6 for *iso*-octane appeared to be closer to the laser sheet (brighter) than the equivalent plume for gasoline, indicating a marginally

faster penetration. It was also interesting to note how well defined plumes #1 and #6 were for *iso*-octane, indicating that there was probably weaker atomisation on the periphery of these plumes at this stage. The RMS images confirmed that the spray tips of plumes #1 and #6 for *iso*-octane had clearly penetrated further by this time. The higher level of variability at these locations is an indication of interactions with the valve flow which had enough momentum to deflect the spray tips into or away from the light-sheet.

Such observations raised questions about the atomisation process for fuels with different distillation characteristics and in particular whether this had an effect on the average droplet sizes produced. In a parallel study, Aleiferis and van Romunde [25] used the same injector and fuel injection pressure in a quiescent chamber at 20 °C and 0.5 bar gas pressure and quantified by shadowgraphy spray tip velocities of the order of 80 ± 1.5 m/s between 400–500 μ s ASOI (about 5 °CA equivalent at 1500 RPM) for both *iso*-octane and gasoline fuels. The subtle differences observed between the two fuels with the laser-sheet technique in the engine suggested that this imaging method captured the atomisation process in greater detail. Aleiferis and van Romunde [25] also carried out droplet sizing by Phase Doppler Anemometry (PDA) with both *iso*-octane and gasoline sprays. They found that gasoline produced very similar average droplet sizes to those of *iso*-octane at 20 °C, 0.5 bar gas pressure, with SMD of about 15 μ m for both fuels. The atomised spray tips of gasoline in the engine exhibited lower levels of RMS and were therefore highly likely to represent real differences in the initial evaporation rates of the higher volatility components in the gasoline, such as pentanes. Indeed, the PDA measurements of the same injector using pure *n*-pentane did produce smaller droplet sizes with SMD of ~ 13 μ m at 20 °C and 0.5 bar gas pressure [25].

The interactions with the intake valve flow developed further by 6° CA ASOI for both fuels and spray plume #6, was clearly deflected towards the exhaust side of the combustion chamber with the plume showing a distinct ‘kink’ along its length as its decreasing momentum due to drag was overcome by the magnitude of the intake flow. On the intake side of this interaction, the plume surface was quite ‘straight’ and well-defined, exhibiting little variation. On the other side of the plume however, significant break-up had taken place with droplets from the leading edge transported away and upstream of the spray tip, due to the combined effects of roll-up vortices around the leading edge and the effects of the local in-cylinder flow field. The intake side plume #1 was also affected by the intake flow with similarly increased break-up levels observed in the direction of the intake flow. However, it was not deflected as a whole because it penetrated below one of the intake valves and therefore did not experience directly the full force of the valve-jet cross flow. With regards to the other four plumes, *iso*-octane produced a spray with continued compactness of each plume and clear separation between them, while for gasoline sprays there were always clear interactions at the surfaces of the plumes thought to stem from improved break-up and secondary atomisation of the high volatility components.

Finally at 8° CA ASOI, around mid-point through the injection event, both fuel sprays appeared more similar to each other, as was also observed in the tumble plane, and visual differences averaged out as spray break-up and flow interactions increased for both sprays. Noticeably though, the area of plume #6 for gasoline was clearly still larger than for *iso*-octane, indicating more efficient spray break-up. After the end of injection $\sim 14^\circ$ CA ASOI (94° CA ATDC) there were higher intensity levels measured for gasoline than for *iso*-octane. However, at this location inside the pent-roof it was difficult to speculate whether these were due to increased levels of

atomisation and entrained droplets for gasoline or due to slower evaporating droplet field of gasoline, as suggested by the tumble plane results of **Fig. 6**.

In-Cylinder Swirl Plane

Although the pent-roof swirl plane imaging was useful in describing the initial spray formation process and the interactions of the spark-plug plumes with the incoming flow over the valves, it could not provide information on how the spray developed as it penetrated into the cylinder. To investigate the robustness of the individual spray plumes over time and as they began to interact with the bulk cylinder flow, a second horizontal, *i.e.* ‘swirl’, plane imaging view was used, positioned ~15 mm below the engine head gasket (see **Fig. 4** for reference). The swirl plane images are shown in **Fig. 8** and from 6° CA ASOI onwards as this was the first frame where the spray plumes became visible through the laser sheet. At 6° CA ASOI there was an immediate and obvious difference between *iso*-octane’s and gasoline’s spray plume footprints. Whereas the four spray plumes could be clearly identified for gasoline, only three were visible for *iso*-octane and even these were only barely visible with a much lower intensity (~50%) than gasoline. The RMS image also showed that whereas gasoline produced high levels of variation around the spray cores, the *iso*-octane sprays had the highest RMS in the plume centres, indicating cyclic variations in the penetration of the spray tips reaching into and out of the laser-sheet. Lower penetrations were also recorded on the tumble plane for *iso*-octane, particularly for the exhaust side, and these swirl plane images clearly confirmed these observations. Spray penetration measurements in the quiescent chamber of [23–25] also produced similar results, with *iso*-octane spray plume penetrations on average 1–2 mm lower than those of gasoline at ~6° ASOI, thus agreeing well with the engine observations. This was an interesting result as it suggested that the vertical spray plume momentum between these fuels was relatively unaffected by the addition of in-cylinder flow and that fluid dynamic phenomena due to different in-nozzle flow or secondary atomisation characteristics were responsible for the differences observed, *e.g.* as already suggested with regards to evaporation rate.

The initial spatial distribution of the plume tip footprints was also of importance as it revealed information about the angle of penetration for each plume, *i.e.* information as to how the nominal spray nozzle-exit trajectories changed throughout the injection process. In this case the plumes of *iso*-octane had a clearly wider footprint than those of gasoline, indicating larger penetrating angles with respect to the cylinder axis. By calibrating the images with a known grid using image analysis software and by knowing the vertical distance of the light sheet below the injector (~35 mm), the initial spray trajectory angles (with respect to the vertical) were calculated for both fuels. For plume #2 which showed the largest difference in location there was a difference of $5^\circ \pm 1.2^\circ$ between *iso*-octane and gasoline, giving trajectory angles of 20.6° and 14.5°, respectively.

Considering that the nominal nozzle design angle for plumes #2 and #5 was 20.8°, the calculated values revealed that *iso*-octane’s plumes were not deviated from their nominal geometry on this plane, while gasoline’s plume trajectory changed by 25% as it was drawn about 5° inwards towards the centre of the chamber. For plume #3 the two fuels were surprisingly similar with both giving trajectory angles lying in the range $14.9^\circ \pm 1.2^\circ$, compared the nominal design angle of 15.3°. For plume #4, gasoline was very similar to its sister plume #3 at 6° CA ASOI but at 7° CA ASOI it was clearly deflected by the cylinder flow towards the cylinder wall. For *iso*-octane plume #4 was not visible at 6° CA ASOI but 1° CA later it too appeared to have been displaced towards the walls in the same direction to that of gasoline. The changes in plume trajectory from the nominal 15.3° for this plume were

calculated using image processing to be $+2^\circ$ (13%) for gasoline and $+4^\circ$ (26%) for *iso*-octane. These observations are summarised in **Fig. 9**. The large movement of plume #4 for *iso*-octane resulted in it being partially clipped by the piston window, clearly not the case for gasoline. Analysis of the PIV images of [15, 16] in this region indicated that a large recirculation zone with mean velocities of about 7–10 m/s existed in the direction of the cylinder walls (as shown schematically in **Fig. 4**), explaining in part the observed plume behaviour. It should be remembered that different groups of plumes had different nozzle turning angles, which have been shown to affect fluid flow inside the nozzle holes and namely the extent of formation of cavitation [39–40]. It was therefore possible that such pronounced differences between the fuels for plume #2 in particular might be explained by such phenomena.

As the injection progressed from 8.0° – 14° CA ASOI, the centroids of luminosity for all plumes developed in a similar manner for both fuels, indicative that flow field interactions on this plane probably had comparable effects on the spray formation of both fuels. Throughout the injection process the gasoline spray plumes remained closer to each other than those of *iso*-octane, but by the end of injection, at 11° CA ASOI, the differences between the two fuels were much smaller; this was also the conclusion drawn from the observed tumble plane results. Finally at 14° CA ASOI, the spark-plug plumes #1 and #6 just came into sight for gasoline, with $\sim 10\%$ variability on the RMS image, indicating that *iso*-octane's plumes were still above the light-sheet. Indeed what showed up on this image for gasoline was likely a form of droplet clustering and ligaments breaking away from the spray tip and being entrained in the cylinder flow; indeed the earlier pent-roof images had already pointed to the larger levels of break-up for these gasoline plumes compared to *iso*-octane.

A final interesting feature of the sprays of both fuels was the gradual disappearance of parts of plumes #2 and #3 on the intake side as injection progressed, particularly clear in the RMS images. This presumably resulted from interactions with one of the intake valves, particularly for plume #3, *i.e.* part of the plume was effectively deflected partially upwards towards the intake port, impinging on the back of the intake valve as the valve lift increased. This occurred because of the quite retarded intake valve open timing employed on this engine operating point. In order to confirm this hypothesis, the phenomenon was investigated further and is discussed later.

SINGLE INJECTION STRATEGY – FUEL TEMPERATURE EFFECTS ON SPRAY FORMATION

Tumble Plane

When the engine operated at fully warm conditions of 90°C engine coolant temperature, gasoline experienced significant changes to its spray formation. In **Fig. 10**, immediately out of the nozzle at 3.6° CA ASOI, the gasoline plumes ‘swelled’ and were drawn closer together, significantly changing the intended spray directionality. Similar observations have been noted in a quiescent injection chamber with the same gasoline fuel sprayed from the same injector at subatmospheric pressure with high fuel temperatures [23–25]. This was a consistent effect for each injection event, originating from flash-boiling of the high volatility components in gasoline and termed ‘spray collapse’. Even in the engine, the envelope of these swollen plumes was very robust on a cycle-to-cycle basis and the RMS images showed low levels of intensity deviation from the mean, indicative of a more homogeneous droplet field. For *iso*-octane, high fuel temperatures did not have the same effect and the two plumes were still individually identifiable, retaining their nominal injection trajectories and geometry.

Specifically, the images showed that most of the intended spray directionality was lost for gasoline sprays by 7.2° CA ASOI. Instead, the plumes penetrated nearly vertically down. Although difficult to show by still images here, movies of the spray's formation illustrated that the effect of rapid evaporation and drag caused the plumes to swell first and then collapse into the central region of the cylinder, with roll-up vortices developing on the intake and exhaust sides of the cylinder transporting droplets away from the spray's tips and to upstream locations around the sides of the plumes. These effects combined reduced the maximum liquid penetration of gasoline by ~10% compared to *iso*-octane. The higher RMS values at the leading edge of the gasoline sprays, particularly under the intake valve, highlighted the strong interactions with the flow at these locations and demonstrated the effect of variations in size and magnitude of the bulk flow structures. Spray impingement on the piston crown at 90 °C was much weaker for both fuels and occurred later than that observed at lower temperatures. In this context it may be speculated that the momentum of the cylinder flow overcame the spray easier at 90 °C and became the dominant mechanism of the transport and evaporation of atomised fuel around the cylinder.

For *iso*-octane, the thinning of the spray plume just behind the spray tip, especially for the plume under the intake valves, was seen clearly in the RMS image at 10.8° CA ASOI and was indicative of the positive effects of air entrainment and momentum exchange in breaking up the spray. This feature also resembled the same effect seen for gasoline at 7.2° CA ASOI at the 20 °C condition in **Fig. 6**. After the end of injection, 11° CA ASOI, the intensity levels for *iso*-octane quickly faded and at 25° CA ASOI the droplet field had completely evaporated from view. For gasoline, the intensity levels at this point were similar to those at 50 °C but spatially more liquid was detected because of the increased concentration of the fuel field below the injector as a result of 'spray collapse'. Interestingly the higher levels of RMS suggested that the final location of the droplet field on a cycle-to-cycle basis appeared to be more dependent on the instantaneous bulk flow structures encountered by the spray rather than fluid dynamic variations in the spray formation itself which were more relevant at colder temperatures. This was consistent with the previous observations that at lower temperatures the delicate balance between spray and airflow momentum determined to a large extent the degree of spray break-up, mixing and wall impingement.

Finally, it was interesting to observe that the early spray development for *iso*-octane at 90 °C was very similar to gasoline at 50 °C, although gasoline was still marginally more swollen even at 50 °C. The 50 °C images are not shown here for brevity but it is noted that the ligament at the spray tip observed on the exhaust side plume at lower temperatures reduced in intensity and was also similar visually to gasoline at 50 °C. The initial tip penetrations for both fuels at 3.6° CA ASOI were not higher at 90 °C than at colder temperatures, however the areas of high intensity (red) did increase in penetration. This was likely to be due to the increased surface area caused by increased atomisation but the possibility of greater levels of cavitation inside the nozzle holes speeding up the flow through the passages could not be discounted [27]. The similarity between gasoline at 50 °C and *iso*-octane at 90 °C indicated that some characteristic temperature could exist, relating the degree of superheat or the average boiling point of the fuel components, to the characteristics of spray formation and atomisation for different fuels. This hypothesis has been expanded in previous publications where other fuels were also studied with respect to non-dimensional number parameter space in a quiescent injection chamber [25, 27, 37]. It would be interesting to see at what point *iso*-octane began to behave in-cylinder like gasoline at 90 °C

but this was not practically possible due to various limitations of the optical engine. However, spray formation imaging in the quiescent chamber showed that a similar degree to gasoline's spray collapse was observed for *iso*-octane at injector temperatures of the order 160–180 °C [23]. It is also interesting to note that at 0.5 bar gas pressure the SMD of *iso*-octane remained almost constant at about 15 µm up to 90 °C injector temperature and was abruptly smaller by about 30% at 120 °C. In contrast, gasoline's droplets decreased in SMD almost linearly from 15 µm down to 9 µm as the injector's temperature was gradually increased from 20 °C to 120 °C at 0.5 bar [25].

Swirl Planes

With reference to **Figs. 11–12**, as with the observations from the tumble plane at 90 °C, the effects of increased evaporation at the leading edge of the gasoline spray plumes and increased drag, noticeably reduced the initial penetrations, which was also reflected in the higher variability of spray-tip arrival at the laser sheet, demonstrated by the RMS image. For *iso*-octane, the penetrations were both greater than gasoline's and also greater than what they were at 20 °C, demonstrated by the visibility of all four plumes at 90 °C. At this temperature the plumes of *iso*-octane also showed much greater 'homogeneity' throughout the injection event, with the main footprint areas being smaller and the centres of luminosity for each plume not changing appreciably until the end of injection. Around the spray cores little luminosity was observed and evaporation around the plumes appeared quite symmetrical, in particular under the intake valves. The spray areas for gasoline were larger and more variable than for *iso*-octane with a larger gradient in luminosity, probably as a result of the different break-up and evaporation rates that produced a broader spectrum of droplet sizes. The central location of the final droplet field for gasoline, shown at 14° CA ASOI, also confirmed the reason for the higher levels of intensity measured in the tumble plane in the centre of the bore at 90 °C, compared to low temperatures where the fuel distribution was 'more stratified' along the central axis of the different plumes.

The pentroof swirl plane clearly highlighted the different behaviour of the individual gasoline and *iso*-octane spray plumes immediately out of the nozzle. For gasoline, the spray collapse mentioned in the preceding discussion of the tumble plane was very clear, with the previously four distinct spray plumes at 20 °C merging to form one single region of luminosity for gasoline. However, the outlines of the plumes showed that some directionality was maintained and that the larger area of Mie scattering intensity was due to increased levels of atomisation around the plumes. This was also confirmed from the in-cylinder swirl plane images discussed later and shown in **Fig. 12**. These clearly showed that although the plumes were significantly distorted, they did not collapse completely in the simplistic manner that one might expect from 'pure' flash-boiling. Inside the pent-roof, the *iso*-octane plume footprint shown in **Fig. 11** was wider at 90 °C and also partially contracted. The footprint of plumes #3 and #4 was now clearly connected to those of the centre plumes #2 and #5. The RMS images of *iso*-octane at 90 °C showed just lower intensity values than those at 20 °C, but clearly lower values at 90 °C were observed for gasoline than at 20 °C (as also quantified on the tumble plane).

As the injection progressed, the directionality of the spark-plug plumes #1 and #6 were clearly seen to be little affected by temperature for *iso*-octane and indeed still experienced a similar deflection to that observed at lower temperatures. However, at 6° CA ASOI the mean intensity of the deflected plume #6 was much lower than at 20 °C, reaching values of only ~30% of the peak at 20 °C. Compared to gasoline the mean intensities for this plume were also marginally lower for *iso*-octane. By 8° CA ASOI the penetrations for both gasoline and *iso*-octane

were clearly lower than for 20 °C as the spray tips evaporated faster. This effect was also reflected in smaller RMS area for both fuels, highlighting the significant improvement in the break-up, atomisation and evaporation of the plumes at high temperatures. Similar observations were made for the intake side plume #1.

SINGLE INJECTION STRATEGY – TIMING EFFECTS ON SPRAY FORMATION

Fig. 13 shows the effect of injection timing on early flame burning (0–10% MFB) and engine stability (COV_{IMEP}). It was found that an injection timing close to 80° CA ATDC was optimum as this led to minimum duration of 0–10% MFB and also minimum levels of COV_{IMEP} . The reasoning behind this figure was not clear immediately though, especially when considering that there was a very sharp change in behaviour on both burning and stability for injection timings past 80° CA ATDC, especially at the colder engine conditions. Therefore, the spray formation was imaged over a range of injection timings during the intake stroke. Although injection timings as early as SOI 40° CA ATDC and as late as SOI 160° CA ATDC were investigated, earlier timings were found impractical and/or unsuitable for image post-processing due to the high levels of piston impingement and fouling of the optical piston crown. **Fig. 14** shows a comparison of spray development with gasoline fuel for two injection timings with SOI 60° CA and 160° CA ATDC at 20 °C. **Fig. 15** shows the spray images with SOI 120° CA at the same conditions, but throughout the injection process. The mean and RMS images illustrated not only the clear interactions between the left intake valve and plumes #2 and #3, but also the different characteristics of spray development and break-up for plumes #1 and #6. It was apparent that interactions with the momentum of the flow past the intake valves were responsible for the differences observed, especially when considering the PIV data from the same engine by [15, 16]). With intake valve open timing fixed at 24° CA ATDC in the current study, the earlier injections during the intake stroke had a pronounced effect on the deflection of plumes #1 and #6 due to higher velocities through the intake valves and produced a wider envelope of atomised fuel within the pent-roof, as illustrated by the mean and RMS spray images. Specifically, for SOI 60° CA ATDC the deflection of plume #6, at 6° ASOI was over 30° from its nominal trajectory, deflected towards the exhaust side. Atomisation and mixing of plume #1 was also significantly improved, although a similar ‘break’ of the plume’s core was not observed as the spray momentum was directly opposing the flow through the valves at this time. However, the high piston position at this point in the cycle and the strong spray momentum of the plumes resulted still in significant piston impingement. This is clear in the RMS image at 8° CA ASOI in the blue regions around the central plumes. There was therefore clearly a trade-off between the benefits of early valve phasing for improved plume break-up and later injection timing for reduction of direct fuel impingement on the piston crown.

When late injection timings were analysed (*e.g.* with SOI=160° CA ATDC), the spray plumes were seen to be nearly unaffected by the intake valve flow and the break-up of plume #6 was driven mainly by flattening of the leading edge due to drag and the associated roll-up vortices transporting atomised droplets up and around the sides of the plumes, *e.g.* as seen at 6° CA ASOI. This vortex showed a high level of repeatability on a cycle-by-cycle basis as illustrated by the clarity of its features in both the mean and the RMS images. In contrast, the area of RMS for SOI 60° CA ATDC was much greater, indicating larger variability in the spray break-up process.

For SOI 120° CA ATDC, the spray exhibited an intermediary behaviour, between the extreme break-up observed at SOI 60° CA and the more repeatable process seen at SOI 160° CA. Indeed data from PIV studies by [15, 16] in the cylinder and intake runner of an identical engine at the same operating conditions, showed that the

peak momentum through the intake runner occurred between 50°–60° CA ATDC. The real time history through the intake valves will lag by some time increment dependent on the velocity of the flow through the intake port. Mean velocities of up to 45 m/s were measured in the intake runner, hence using the distance from the PIV measurement location to the back of the intake valves, the time lag associated with this flow pattern travelling into the engine's pent-roof would be between 1–2 ms. This placed the peak momentum through the valves at about 65–75° CA ATDC which agrees with the observed spray behaviour between 66° CA and 86° CA ATDC.

In order to compare how the volatilities of the fuels affected the spray's development, *iso*-octane and gasoline were also investigated at 90 °C engine head temperature and SOI=120° CA ATDC; these results are shown in **Fig. 16**. The images of **Fig. 15** at 20 °C showed that although at the early stages of spray development the two fuels showed very similar behaviour, between 6°–8° CA ASOI, there were noticeable differences in the way that plume #6 was broken up and atomised. Although some subtle differences had already been observed even for SOI 80° CA ATDC, it had been unclear as to how interactions with the strong incoming flows were affecting break-up. The differences with later injection timing were much more obvious and clearly highlighted that the fuel properties on their own (regardless of flow interactions) had an influence on the natural break-up characteristics of the spray plumes. *Iso*-octane was observed to penetrate the laser-sheet with only some minor disturbances to its nominal spray development, as shown by the small area of higher intensity in the mean image at 8° CA ASOI. The RMS image also showed very small values of intensity variations. For gasoline however, at the same timing ASOI, plume #6 showed much greater signs of atomisation and spray tip break-up. These effects were again reflected in the RMS image which had higher values and larger area than *iso*-octane on the exhaust side. For 90 °C these differences were not as apparent because the rapid break-up and evaporation quickly altered the spray momentum for both fuels, showing very similar mean sprays, particularly for plume #6.

The timing of 14° CA ASOI is included in **Figs. 15–16** to highlight mechanisms of secondary atomization and spray interaction other than with the incoming flow. Specifically, although not immediately apparent from the mean images, the RMS images showed that on a cycle-to-cycle basis there was a large stream of droplets produced from the left intake valve side after the end of injection. This was seen for both fuels at 20 °C and appeared to be more severe for gasoline. It appeared that as the injection timing was phased towards maximum valve lift – which occurred at 149° CA ATDC – the interactions between plume #3 and the top of the intake valve resulted in part of the plume being deflected up into the intake port, ramping off the back of the valve. As injection ended, this fuel film was then stripped off the back of the valve and resulted in the stream of droplets observed. The areas of scattered light by these droplets were significantly above the pixel resolution and clearly visible. Considering the timescales involved, it is likely that these droplets did not evaporate much before impinging on either the cylinder liner or the piston crown. However, the nature of the Mie-scattering imaging technique means that no definitive statement can be made about the type of fuel splashing off the back of the valve, *i.e.* since Mie intensity is proportional to liquid surface area ($\propto d^2$) the imaged signal could have been produced by either large ligaments stripping off the valve's surface or from a larger number of smaller droplets produced by secondary wall atomization effects. In any case, this phenomenon may also be associated with the deterioration in combustion stability with this injection timing of SOI=120° CA ATDC, as seen earlier in **Fig. 13**. The fact that these droplets appeared in the RMS image indicated that the degree of spray/valve impingement and secondary droplet production was quite variable on a cycle-to-cycle basis, suggesting in turn that the spray's

interaction with the cyclically variable incoming flow altered the exact form of impingement from one cycle to the next.

At 90 °C the spray's interaction with the valves was nearly eliminated for a number of reasons. Firstly, the plumes were drawn inwards under the injector tip due to partial spray collapse. Secondly, increased evaporation reduced droplet sizes to levels where they were entrained more easily into the cylinder flow, and thirdly because the hot intake valves aided evaporation of any droplets or liquid film that formed on the valves.

In summary, under cold-start conditions, the selection of injection timing with such complex multi-hole spray patterns is critical for optimum spray formation, atomisation and mixing and should be chosen so as to increase flow interactions as much as possible whilst minimising potential contact with any in-cylinder surfaces and formation of fuel films that are detrimental to robust engine performance. Additionally, the selection of compatible injection and valve timing strategies is of utmost importance, particularly for cold-start conditions.

SINGLE INJECTION STRATEGY – ENGINE LOAD EFFECTS ON SPRAY FORMATION

Tumble and Swirl Planes

The effects of in-cylinder pressure and bulk flow magnitude on spray formation were also investigated by operating the engine at full load, *i.e.* under wide-open-throttle conditions with 1.0 bar intake plenum pressure. **Figs. 17–21** show the mean and RMS images of spray development at full load using all three laser-planes (tumble, pent-roof and in-cylinder swirl) at 20 °C and 90 °C with both gasoline and *iso*-octane fuels. Since the main features of spray formation have already been reviewed in the previous section, this section focuses on discussing specific differences observed between spray development at low load and full load (0.5 bar and 1.0 bar intake pressure, respectively). In order to represent realistic in-cylinder spray development at this condition, a fuel injection pulse of 1.5 ms was employed and the sprays compared to the 0.9 ms pulse used for low-load conditions. This also enabled interpretation of the spray images after 11° CA ASOI as the end of injection (EOI) moved to about 18° CA ASOI for full load.

The mean and RMS spray development on the tumble plane is shown in **Fig. 17** for 20 °C. This can be compared directly to **Fig. 6**. For 'first fuel' seen out of the nozzle at about 4° CA ASOI, the higher in-cylinder pressure did not seem to affect the spray formation significantly for either *iso*-octane or gasoline. This was demonstrated generally by near-identical cone angles and penetrations of the plumes for both fuels. Even features such as the thinning of the spray tip seen for *iso*-octane at 20 °C on the exhaust side at low load were replicated at full load. For gasoline, the intake side plumes were also visually very similar to those produced at low load. On the exhaust side however, there were some noticeable changes in the spray tip characteristics. Specifically, at low-load conditions, the leading edge appeared to be breaking up quickly, but for full load this edge was much more intact as illustrated by the longer penetration of the high intensity red regions. This feature appeared in both *iso*-octane and gasoline images with similar thinning of the leading 2–3 mm of the plume. This showed that the increase in cylinder pressure had an effect on the immediate atomisation characteristics, slowing down the break-up of the spray's leading edge. The similarities between the two fuels were also replicated in the RMS images.

At 7.2° CA ASOI it was already clear that peak spray tip penetrations were lower at full load by ~10% for both fuels when compared to low load. This was made obvious by comparing the RMS images. The spray's interaction with the stronger flow field resulted in faster break-up of the spray tip, particularly for *iso*-octane. Thinning of the spray plume on the intake side was observed immediately behind the leading edge for *iso*-

octane. The plume penetration on the exhaust side was shorter as the spray quickly evaporated. Gasoline on the other hand produced longer penetrations on the exhaust side and also higher levels of RMS intensity around the spray tips, indicating higher levels of variability in liquid fuel presence over those regions.

As injection progressed past 10.8° CA ASOI, *i.e.* past the EOI point of low-load conditions, the sprays at full load were still at least 10 mm above the piston surface. In comparison, at low load, the spray had already impinged on the piston surface by this time. As injection continued, the full-load sprays still showed limited levels of impingement onto the piston crown, which was initially surprising given the near doubling of the injection pulse duration. These observations were clear indicators of the higher drag from the increased gas density and the effectiveness of the stronger incoming airflow in breaking up the spray plumes and entraining the resultant droplets into the flow but also keeping them away from in-cylinder surfaces. The reduction in plume penetration was also observed in detail on the horizontal imaging planes shown in **Fig. 18**. This was illustrated by the round, more symmetrical plume footprints and the more compact areas of RMS (the reader may refer to **Fig. 8** for direct comparison with the low-load condition).

Spray plume #2 under the intake valves, also appeared to be displaced towards the injector axis, more for gasoline than for *iso*-octane at 7° CA ASOI. The RMS image also suggested that gasoline had broken up more than *iso*-octane, as demonstrated by its larger spray plume envelopes. However, as also mentioned earlier, the lower levels of intensity around the spray plumes of *iso*-octane may also reflect the faster evaporation of atomised droplets around the plumes. As injection progressed these differences became smaller and by 11° CA ASOI the two fuels had very comparable spray footprints. In the final frame shown on the tumble plane, 25.2° CA ASOI the mean images showed that *iso*-octane produced lower levels of luminosity in the final droplet field, indicating that this was a real effect likely to be associated with faster evaporation of the droplet field. This is believed to highlight how important the heavy fractions in the gasoline blend can be in determining the speed of evaporation and thus the initial quality of mixture composition. However, compared to low load, the differences were much smaller.

The effect of increasing the engine temperature on spray development under full-load conditions is shown first on the tumble plane in **Fig. 19** for 90 °C. This can be compared directly to **Fig. 10** of low-load conditions. The lower intensity levels in the mean image compared to that at 20 °C, by some 50% or more, illustrated the effect of temperature to higher evaporation rates. However, the envelope of liquid fuel remained unchanged. The levels of spray tip variability were comparable to low load, highlighting the robustness of the early spray formation process. Throughout the injection process strong interactions with the in-cylinder flow changed the nature of spray development, but even at 90 °C these were not sufficient to result in even a small degree of spray collapse for gasoline at full load, as clearly seen by the well-defined spray plume cone angles in the RMS image.

The full-load spray development at 90 °C on the horizontal planes of ‘pentroof swirl’ and in-cylinder swirl’ is shown in **Figs. 20–21**, respectively. These can be compared directly to the low-load conditions of **Figs. 11–12**. The images revealed some clear differences between *iso*-octane and gasoline. Specifically, although the initial footprints of both fuels were quite similar, plumes #1 and #6 broke up differently for the two fuels. For gasoline these plumes were rapidly atomised and quickly lost their definition, whilst for *iso*-octane they initially experienced a strong deflection by the valve jet, as shown at 5° and 6° CA ASOI, with the spray pattern breaking up considerably. However, while the plumes of gasoline disintegrated quickly as a result of these flow

interactions, the plumes of *iso*-octane remained well defined throughout the injection, as shown in the last frame of 14° CA ASOI (plumes #1 and #6 are clearly still ‘straight’ and unaffected when compared to gasoline). The in-cylinder swirl plane images also bore out the same trends and indeed by 14° CA ASOI atomisation appeared significantly better for gasoline. However, since it was known already from the tumble plane images that the average spray luminosity envelope was smaller for *iso*-octane than for gasoline, the definition of *iso*-octane’s plumes is presumably related to the fact that any atomised spray droplets evaporated quickly to a degree that they no longer scattering enough light, thereby giving the impression of poorer atomisation due to more compact spray plumes. The gasoline sprays clearly did not collapse immediately out of the nozzle and while the pent-roof swirl images might have given this impression, the in-cylinder swirl plane images showed that this was not the case. Compared to the low-load sprays, the RMS images of full-load operation showed that the footprint variability of plumes #2, #3, #4 and #5 was much lower as indicated by the clearly defined ‘black’ regions in the images. For plumes that interacted directly with the incoming valve jet however, plumes #1 and #6, the levels of variability were higher at full load, as indicated by the red regions.

This was a clear example of how beneficial flow interactions can be in contributing to the break-up process, under conditions that did not favour natural spray break-up. Indeed, the poorer spray break-up at these conditions was clear from the RMS images, which showed much smaller envelopes of atomisation around the plumes at 8° CA ASOI compared to low load. The plume penetrations were clearly reduced with higher in-cylinder pressures as shown by both fuels’ plumes being only just visible at 6° CA ASOI in **Fig. 21** (expected from gas density effects). At 7° CA and 8° CA ASOI the plumes of both fuels were very similar and had smaller footprints than at low-load conditions due to the reduced levels of atomisation at the periphery of the plumes. The higher levels of RMS around the core of the plumes however showed that there were noticeably higher levels of interaction with the in-cylinder flow highlighted by the large regions around the plumes cores.

For completeness, it should also be mentioned that work by other authors with a DISI engine injector (6-hole, bend angle ~15°, quiescent chamber conditions, 80 bar injection pressure) showed that the orientation of the spray plumes was ‘twisted’ by about 30° under flash-boiling conditions [41]. Such significant rotation phenomena were not distinctively observed in the current study, despite the extensive imaging planes. It is not clear why but it is plausible that the injector angle, nozzle-hole geometrical differences, as well as different injection pressure between the two injectors were associated with different in-nozzle cavitation phenomena that, in turn, led to differences in spray’s momentum and in the manifestation of spray rotation.

Probability of Fuel Presence

The discussion so far has focused on describing the general macroscopic characteristics of spray formation, and the related effects of temperature and in-cylinder pressure for fuels of different volatilities. The mean and RMS images in the previous sections also revealed that piston impingement was particularly prevalent at cold engine conditions, as viewed from the tumble plane results. In this section, images of probability of fuel presence are used to quantify the degree of liquid impingement over 100 consecutive engine cycles on the tumble plane for 20 °C, 50 °C and 90 °C engine head temperatures. **Fig. 22** shows the sprays of *iso*-octane and gasoline at 14° CA ASOI (94° CA ATDC or 3° CA after EOI); low-load and full-load conditions are compared directly. COV spray images are also presented in **Fig. 24**; these were calculated by dividing the RMS image by the mean image. The lower image boundaries have been cropped to correspond to the piston surface.

The probability images were calculated by binarising the instantaneous images using a thresholding operation and then averaging all the thresholded images at each crank angle ASOI, hence creating a probability map of the presence of liquid fuel with pixel values in the range 0–100% (the same method has been used in the quiescent chamber sprays [23]). The usefulness of this technique compared to the RMS image is that it provided a more direct interpretation of the boundaries of the liquid envelope in 2D and also allowed tracking of the liquid spray field as it evaporated during the intake stroke. While this approach reflects usefully the spatial probability presence of liquid fuel defined by an intensity threshold value, the COV images retain all the original information from the raw image intensity histograms, thereby giving a complementary quantification of the spatial variability envelope. Therefore where images showed regions of low fuel probability (blue-black regions) around the periphery of the spray plumes, the COV maps showed the highest percentage variation and where images showed 100% fuel probability presence (red regions) in the centre of mean spray plumes, the COV maps showed the lowest levels of variation (black regions).

The probability images indicated that for 20 °C and 50 °C there were measurable levels of piston impingement for both fuels at low-load conditions. Gasoline showed impingement on both the intake and exhaust sides for 20 °C while *iso*-octane showed only significant levels of impingement on the intake side at any time. The dark blue levels of probability above the piston surface on the exhaust side for gasoline showed that there were many cycles where liquid fuel still reached the piston crown surface and indeed this was highlighted by the yellow-colour footprint on the piston surface. This occurred at about 50–75% of the cycles in comparison to 0% with *iso*-octane fuelling. It should be emphasized that the bright ‘patches’ shown in the images on the piston surface were not produced as a result of reflections of penetrating fuel but were only observed when direct impingement occurred and formed fuel films of high levels of light-scattering.

The transition gradients in the probability images (from ‘black’ to ‘yellow’ to ‘red’ regions), especially at the spray tips for both fuels can be used as a measure of spray tip break-up and evaporation, since sprays that disintegrated and began evaporating quickly (reducing the droplet surface area) produced lower levels of scattering. At 20 °C the spray tips on the intake side were more variable for gasoline and showed larger areas of lower liquid probability compared to *iso*-octane. However, at 50 °C, the reverse was true, with *iso*-octane producing a wider transition area of probability from 0–100% fuel presence.

Impingement was always observed to be larger for the intake side plume, highlighting the more variable nature of break-up in this region of flow just past the valves. The higher levels of liquid impingement could be associated with lower drag levels in low-pressure wake regions. The gasoline sprays in general also appeared to produce wider plumes. However, inspection of the equivalent mean and RMS images showed that this was due to higher average levels of scattering intensity of the gasoline spray plumes, as these broke up into droplets differently to *iso*-octane’s. As the engine head temperature was increased to 90 °C, the result of keeping the same intensity threshold value highlighted how the spray envelope reduced due to evaporation. For gasoline, the envelope changed significantly as a result of spray collapse due to flash-boiling and significant piston impingement was eliminated at these conditions.

For full load, the differences between the two fuels in terms of probability of fuel presence were much smaller. At 20 °C it appeared that the effect of high volatility components in the gasoline improved atomisation and widened the spray envelope. However, as temperature increased it was clear that *iso*-octane produced the

smallest probability of presence area, indicating improved evaporation around the spray plumes and highlighting the adverse effect of low volatility fractions of gasoline in reducing the droplet evaporation rate. The levels of impingement were also eliminated even at this cold condition. Bearing in mind that for full-load conditions the injection event had not yet finished, this was quite remarkable; it is reminded that a 1.5 ms pulse width was used such that these results of **Fig. 22–23** show sprays about 3° CA before EOI). The COV images however, indicated that some fuel did make it to the piston on occasions even under full-load conditions, but that the levels of COV in these regions were very high ~100%. In **Fig. 23** almost all conditions exhibit some degree of piston impingement even at 90 °C. Once again the gradients of COV through the spray tips reveal significant information about the nature of spray plume break-up from cycle-to-cycle. In general, the peak COV reduced with increasing temperature for *iso*-octane but increased for gasoline, which still showed peak levels of COV over 100% at 90 °C. The highest levels of variation clearly occurred at cold conditions and were highest for *iso*-octane. The differences in volatility of the fuels were also manifested more clearly at higher temperatures, where the spray tips were more ‘homogeneous’ for *iso*-octane than for gasoline at both low-load and full-load conditions. The COV images at 90 °C low load also showed that the cylinder was almost completely filled with a fairly ‘homogeneous’ droplet cloud for gasoline, whilst the spray envelope was quite unchanged for *iso*-octane. These differences in mixture preparation can act as sources of combustion variation between fuels via changes in the cycle-by-cycle fuel concentration at ignition timing [7].

TRIPLE INJECTION STRATEGY – EFFECTS ON SPRAY FORMATION

The spray development for a triple injection strategy using three individual, *i.e.* ‘split’, injection events in the intake stroke (delivering in total the fuel mass required for stoichiometric operation) is shown in **Fig. 24** for gasoline. The mean and RMS spray development is shown in **Fig. 25** for *iso*-octane at 20 °C. These figures can be compared to the single-injection tumble plane sprays of **Fig. 4** and **Fig. 6**. Due to the imaging frame rate used for this plane (5 kHz or 1.8° CA/frame), it was not possible to compare the three injections at exactly the same crank angle after the start of each individual injection, thus the differences in the time stamp ASOI of the spray images for each injection. It is observed that the shorter injector pulse widths worked well in reducing the spray’s momentum and plume tip penetration. The gasoline ‘spray collapse’ features of the single injection strategy are also present with the triple injection strategy. Comparison of the triple injection *iso*-octane sprays of **Fig. 25** with those of the single injection in **Fig. 6** reveals that the macro-scale of the spray structure remained the same, *e.g.* the mean and RMS feature of a longer right-hand-side plume at about 7–10° CA ASOI in **Fig. 6** is clearly present in the first two injections of **Fig. 25**. However, one particular aspect of the triple injection strategy was that after the end of each injection shot, a trail of droplets was observed stemming from the injector tip (**Fig. 25**). Although some droplets were also seen for the single-injection strategy at 20 °C, there were clearly more of these with three sequential injection events. They were observed on a cycle-to-cycle analysis and also show up in the RMS image, as highlighted by the yellow dashed-line circle in **Fig. 25** for the second injection at 78° CA ATDC. It is also immediately apparent from this tumble plane imaging that splitting the single injection event into three injections of smaller duration, reduced the levels of liquid impingement on the piston crown significantly. It appears that the reduced spray penetration of each short injection event enabled the in-cylinder flow to ‘overcome’ the spray’s momentum and quickly entrain the liquid droplets. These phenomena were

studied in more detail using a full optical bore piston and wall-mounted heat flux sensing instrumentation and recently published elsewhere [21], hence not elaborated on further here.

Analysis of the combustion performance matrix for the triple-injection strategy showed just a marginal decrease in IMEP for both *iso*-octane and gasoline fuels; in terms of combustion stability though, the effect was clearly negative for most conditions, with higher COV levels recorded typically throughout [7]. To understand this, the differences in mixture stratification amongst fuels and injection strategies were studied by in-cylinder FFID at the spark-plug location and pentroof wall [7]. The results showed that differences in the gaseous mole fractions did exist. Reference to gaseous mole fraction is important here because it was found that there were some clear signs of a ‘wetted’ spark-electrode and pent-roof from the injection event, even under firing conditions, especially at 20 °C. The degree of this type of stratification was generally higher for cold engine conditions. At 20 °C, at the spark-plug location, the value of λ was 0.96 for *iso*-octane and 1.09 for gasoline. Gasoline also exhibited a lower degree of stratification than *iso*-octane, with a difference in λ of about 0.1 and 0.2, respectively. At 90 °C, gasoline showed a very similar value of λ and degree of stratification to that at 20 °C, whilst *iso*-octane was significantly affected with a λ value of ~ 1.03 at the spark plug and leaner by ~ 0.12 at the wall. This means that *iso*-octane’s fuel concentration field was more homogeneous (or less stratified) than at 20 °C and more similar to gasoline’s. Those trends also agreed with the PLIF measurements of [18–19] in an identical optical engine run at the same operating conditions. Specifically, the cyclic variability in the FFID’s ‘plateau’ levels at the spark plug location had a COV of $\sim 10\%$, compared to $\sim 20\%$ quantified by PLIF. However, the mixture in-homogeneity measured by PLIF was just less than $\sim 10\%$ when spatially averaged over a scale of 350 μm at spark timing. The FFID measurements were repeated with the triple injection strategy and it was found that the triple injection tests produced leaner equivalence ratio for both *iso*-octane and gasoline at the spark-plug location, with both fuels exhibiting a very similar value of λ , ~ 1.12 at 90 °C, and a lower degree of stratification than the single-injection strategy.

SINGLE INJECTION STRATEGY – ALCOHOL FUELLING

The mean ‘global’ spray formation images obtained through the piston crown with gasoline, *iso*-octane, ethanol and *n*-butanol fuels using diffused Laser-light illumination are shown in **Figs. 26–27** for 20 °C and 90 °C, respectively. The corresponding RMS images are shown in **Figs. 28–29**. The COV images at a fixed frame ASOI are also shown in **Fig. 30** as these removed any bias in the RMS images arising from scattering intensity variations between fuels. Quantification of imaged spray areas in this engine has been presented elsewhere with two different injectors [3, 20, 37], hence analysis here is focused primarily on the statistically processed images of **Figs. 26–30**.

Alcohols spray plumes were slower to exit the injector nozzle and clearly produced thinner and more compact spray plumes. At 20 °C it was also clear from the RMS images that the levels of plume ‘flapping’ were greater with gasoline and *iso*-octane than for both alcohols, as indicated by the regions of higher RMS values along the outer region of the plumes. For ethanol and *n*-butanol, the locations of highest variation tended to be limited towards the spray tips, and indeed this highlighted the compactness observed for the alcohols plumes in general. Specifically, for *n*-butanol, clear regions of droplets/ligaments ahead of the spray tips (plumes #3 and #4) were seen on the RMS image, which were too large to be entrained into the cylinder flow and impinged on the piston crown. The overall RMS areas were also much smaller for *n*-butanol, emphasizing the poor atomisation and lack

of air entrainment in the spray plumes at cold conditions. At 90 °C, greater similarities were observed between the locations of variation for ethanol and gasoline, although gasoline was seen to collapse marginally more. The sprays of *n*-butanol and *iso*-octane did not collapse and *n*-butanol showed larger regions of variation than at 20 °C due to improved atomisation and air entrainment. In general, the core sprays of all fuels were robust on a cycle-to-cycle basis, as indicated by the dark regions of low RMS in the relevant areas, defining the ‘spine’ of the individual plumes.

The COV images in **Fig. 30** are shown with a color scale where dark blue indicates 5% variability and the start of the red region corresponds to about 25% variability; the red range extends to a maximum variability of 75% and 115% at 90 °C and 20 °C, respectively. This was done to maintain a high level of contrast in the images at the lower intensity areas and clearly identify regions where higher levels of variation existed with respect to the mean. It was immediately clear from the sprays at 20 °C that there were significant differences between fuels. Both alcohols showed much lower levels and areas of plume COV, highlighting their compactness and weak atomization features. Although ethanol sprays exhibited some red regions where COV was high, around the spray tips and edges of some plumes, *n*-butanol sprays had very little COV. In comparison, *iso*-octane and gasoline showed much higher levels of COV, with large red zones all around the spray perimeter. It is believed that these red zones were created by the way plumes began to break up and disintegrate through greater interactions with the bulk flow.

Overall, the COV images produced quite an interesting result on their own, because they highlighted that even fuels with higher boiling points *e.g.* *iso*-octane, could have significantly better atomization, break-up and subsequent evaporation characteristics due to advantages in other relevant fuel properties. Indeed, the differences in surface tension at 20 °C between *iso*-octane (boiling point ~ 76 °C at 0.5 bar) and ethanol (boiling point ~62 °C at 0.5 bar) are of the order of 35%, with lower surface tension improving break up and leading to a 12% larger Weber number for *iso*-octane [25]. The Ohnesorge number that combines the Weber and Reynolds numbers, *i.e.* surface tension and viscosity effects, may be used as a metric to determine on first approximation how different fuels actually atomize and approach flash boiling without having to test them. At colder conditions the differences between fuel properties are greater, meaning that alcohol fuels which typically have higher surface tension and viscosities than low carbon number aliphatic hydrocarbons (C₅–C₁₀) have shown significant differences in their Ohnesorge numbers and in their respective atomisation characteristics [25, 37]. At 90 °C the differences between fuels are reduced significantly. Specifically, as viscosity and surface tension decreased drastically at high temperatures, the sprays of *n*-butanol showed greater levels of variation around the plumes and a larger projected spray area. In fact it is interesting to note that the similarities of *n*-butanol’s Weber number at 90 °C and *iso*-octane’s at 20 °C [25] coincide with the visual characteristics of the in-cylinder sprays.

As far as the two multi-component fuels, gasoline and E10, are concerned, at 20 °C the E10 spray was found to be macroscopically no different from gasoline. However, as the temperature was increased, more noticeable differences appeared. The distillation curve for E10 is very different in the 20–70 °C range compared to gasoline’s, and E10’s vapour pressure is also higher from about 40 °C onwards (**Fig. 2**). The mean and RMS E10 spray images are shown for 70 °C and 90 °C in **Fig. 31**. It is noted that the 70 °C set point was specifically selected for E10 to observe superheat effects related to ethanol’s boiling point of ~62 °C at the 0.5 bar low-load condition used here. It is immediately clear that the sprays of E10 at 70 °C resemble closely those of gasoline at

90 °C; this was probably caused by the higher levels of superheat experienced by E10 components as a result of the higher vapour pressure. At 90 °C, the E10 sprays were clearly more collapsed than those of gasoline, immediately out of the nozzle, and only the two spark-plug plumes (#1 and #6) were still vaguely identifiable. This higher degree of spray collapse is further emphasized by the absence of the ‘cross-shape’ in the RMS formed by plumes #2–#5, which was still observed at 70 °C. This evidence also means that 10% ethanol addition can result in higher levels of in-nozzle cavitation which, combined with superheating, can generate greater levels of spray collapse. Previous work by the current authors with E10 and gasoline run through real-size optical injector nozzle holes has highlighted supporting observations [27]. In this context, it is also clear that in-nozzle phase change can alter the spray characteristics for all fuels and, although one of the main drivers of cavitation is the fluid’s vapour pressure, the liquid viscosity can counteract certain phenomena through damping effects of turbulent eddies inside the nozzle, since these are typical sources of local low pressure regions that can give rise to cavitation [27, 42, 43].

SINGLE AND TRIPLE INJECTION STRATEGIES – EFFECTS ON SPARK DISCHARGE

The spark event is clearly important in defining some of the initial conditions for a flame kernel to develop in an SI engine. In order to obtain baseline information about the spark event in this engine at the main conditions of interest, a high-speed imaging study was undertaken with focus on understanding the effect of flow and injection strategy on the spark discharge itself. Initially, spark images were obtained at static engine conditions, *i.e.* 0 RPM, 1.0 bar in-cylinder pressure. Then the spark imaging experiments were repeated with the engine motoring at full load, *i.e.* with 1.0 bar intake pressure, for direct comparison with the 1.0 bar static engine conditions. Engine motoring experiments were also carried out at low-load conditions with 0.5 bar intake pressure, as this formed the main operating point throughout the current work. At low-load operation, the spark was also imaged with different injection strategies. Specifically, the single injection strategy of baseline SOI 80° CA ATDC was employed, as well as another one with SOI 120° CA ATDC. The latter was employed because injection timing sweeps had shown that such late SOIs led to deterioration in combustion stability. The spark was also imaged with the triple injection strategy (*i.e.* with split SOIs at 60°, 70° and 80° CA ATDC).

Mean and RMS spark images are shown in **Fig. 32** at 5° CA after ignition timing. A spark advance of 35° CA was fixed throughout this work, hence these spark images correspond to 30° CA BTDC. It should be noted that the images for both the static and full-load tests are shown with a 51/255 pixel intensity scaling, whilst the low-load images are shown with a 21/255 scaling. This highlights that the spark’s luminosity was significantly reduced at low-load operating conditions. For this view the spark-electrode was inclined towards the camera so that the top landing of the ground electrode could clearly be visualized. This was the ‘isometric view’ explained earlier and also shown photographically in **Fig. 32** for reference. A schematic of the electrodes has also been superimposed on the spark images to aid interpretation.

The images showed the significant effect the flow field had on the behaviour of the spark plasma. At static conditions, the shape of the spark plasma was highly repeatable and the mean image showed a compact region of high luminosity corresponding to the tip of the central electrode. The RMS image also showed some strongly localized areas of variability around the tip and near zero variation in the strike zone, which was defined by a very clear thin spark channel. On the other hand, the motoring tests showed that the mean spark luminosities were rather lower and that the envelope of the strike zone on the electrode was significantly widened. The RMS

image under motoring conditions showed localized red regions at the base of the electrode indicating the variation in strike zones, and their wide envelope both spatially and in intensity terms, demonstrating that sparks were clearly stretched, distorted and of variable strength from cycle to cycle. These effects were only augmented under full-load conditions, with both the mean and RMS images showing much larger spatial envelopes. The RMS image in particular was able to highlight the wide range of discharges that occurred as a direct result of local variations in the flow field and that sparks could be stretched by large distances of about 1–3 mm while still remaining anchored to the central and ground electrodes.

The imaging study was extended to firing conditions so that sparks could be analysed under real-world operating conditions in the presence of hot residual gas and in association with the performance of their respective combustion cycles. The sparks were imaged through the pent-roof window this time, *i.e.* the tumble plane; the photographic view of the spark plug itself is also shown for reference. Motoring tests were initially carried out with this imaging arrangement to obtain baseline spark images for direct comparison with the firing tests via mean and RMS statistical analysis. These tests are shown in **Fig. 33** at 5° CA after ignition timing. A schematic of the electrodes has been superimposed again to aid interpretation. It should be noted that the mean intensity levels for these images should not be directly compared to **Fig. 32** due to the different imaging arrangement and settings used. However, it is clear from the motoring spark images of **Fig. 33** that these did not exhibit the high levels of localised intensity gradients regions observed in **Fig. 32** that were associated with different cycle-to-cycle strike locations on the ground electrode. High-speed movies though showed that such strikes were still taking place but were being masked by the ground electrode due to inclination of the spark plug away from the vertical axis. Nonetheless, the view was still useful because it allowed analysis of the directionality of distortion of the spark by the conventional tumble motion.

Spark images for firing engine conditions with different injection strategies are shown in **Fig. 34**. In particular, the RMS maps demonstrated a wider area of luminous sparking for the standard single injection strategy of SOI=80° CA ATDC, indicating that a number of cycles produced very elongated and stretched sparks. For the later injection timing of SOI=120° CA ATDC, it was interesting to see a different RMS envelope; the intensity levels here were concentrated in a very central region with lower numbers of strikes occurring on the near edge of the ground electrode *i.e.* the spark was not so greatly influenced by the flow-field. The concentrated channel of high intensity for the triple injection case also indicated that this produced the most repeatable spark event on a cycle-to-cycle basis. This resulted in a smaller envelope of variability and together the results appeared to suggest that different injection timings might be able to influence not only mixture preparation processes but also the mean bulk flow at ignition timing and potentially influence the initial direction and burn rate of the flame kernel.

To investigate these trends further, the data for single injection were conditionally averaged based on the cycles' P_{max} to try and assess whether there was indeed a correlation between improved/reduced combustion 'quality' and the spark behaviour. For 100 consecutive cycles of data with a mean P_{max} of 16.5 bar, sparks from cycles with $P_{max} > 18$ bar (10 cycles with mean $P_{max} = 18.4$ bar) and $P_{max} < 16$ bar (16 cycles with mean $P_{max} = 14.7$ bar) were averaged and the RMS imaged also calculated. The results shown in **Fig. 35** indicated that fast cycles, which produced higher peak cylinder pressures nearly always produced highly stretched sparks with different strike locations, while slow cycles had a much more 'neutral' central sparks with low displacements. For a full analysis

of these effects, simultaneous measurements of spark current and voltage would be necessary to assess the contribution of these factors on spark energy and breakdown characteristics versus the spark behaviour observed. However, such trends have also been reported in [44] where the spark was optimised to become deflected and elongated so that it could ignite a stratified mixture. Similarly, spark characterisation was carried out in [45] where it was reported that for cycles with the same spark energies those with longer duration produced faster cycles. This was in agreement with conclusions cited therein by [46–47], that a large ignition window led to fast early flame kernel development, because it masked the effects of in-cylinder variation in local AFR. These findings were also in agreement with other researchers [48–50] and have also been discussed more recently in [32, 51].

The study of [45] also reported on the fluctuating nature of the spark and the distortion of the discharge channel due to flow-field effects and hypothesized that based on the results, large spark energies and short durations, associated with high levels of IMEP, were a result of higher mean velocities at the spark-plug gap around ignition. The results of the current work seem to lend support to this hypothesis, and suggested that different direct-injection strategies in the intake stroke had the ability to influence the flow field around ignition timing. Based on the injection strategies investigated, and knowledge of the temporal evolution of the flow field during the intake stroke gained from [15, 16], it would appear that phasing the injection process in accordance with the peak flow velocities through the intake valves ($\sim 80^\circ$ CA ATDC for the current valve timing) might add energy to the bulk flow and generate higher velocities at the spark-plug location at the end of the compression stroke. Direct-injection timings might therefore not only be able to affect combustion through the mixture preparation process by changes to the spray-break up, mixing and evaporation time available to produce a fully mixed charge, but also through ‘energizing’ or partially ‘neutralizing’ scales of the flow. These effects need to be investigated further but pose a possible new avenue of research and development for direct-injected spark-ignition engines, namely the use of the injector to control the flow-field at spark timing and potentially even the turbulence intensity. To examine potential effects, the PIV study of [17] was carefully considered. These authors found that a single injection event of 1 ms duration per cycle in the early intake stroke (same strategy to that of the current study) produced $\sim 10\%$ lower RMS velocity at ignition timing than a ‘split’ injection strategy with three sequential injection events of 0.3 ms duration each early in the intake stroke; the effect of an injection strategy with two sequential injection events led to 5% increase in RMS velocity at ignition timing. However, the differences in RMS velocity between the different strategies just 10° CA AIT had dropped to levels below 4–5%. Considering these small differences in RMS velocity soon after ignition timing, the differences observed between the single and triple injection strategy in **Fig. 34** seem consistent. In contrast, the later single-injection strategy with SOI 120° CA ATDC exhibited a narrower mean spark envelope and a wider RMS envelope that may suggest differences in the flow field at the spark plug location at ignition timing to that with SOI 80° CA ATDC. However, the exact origins require further work and analysis, possibly in conjunction with the fuel concentration field (e.g. leaner mixture and larger cyclic variability in flame initiation).

DOUBLE HYBRID INJECTION STRATEGY – EFFECTS ON SPRAY, SPARK AND COMBUSTION

Fig. 36 shows a typical spray event of late injection stage from a single cycle 5° CA away from TDC, without engaging the spark. Plumes #2–#5 are seen clearly hitting the piston crown; they appear as ‘O’ rings around the saturated liquid spray at 7° CA ASOI and they move outwards as lower luminosity ‘clouds’ from 8° CA ASOI.

Plumes 1 and 6 are seen passing by the spark plug electrode and also hitting the crown at 9° CA ASOI. Considering these effects and as a consequence of the discussion in previous sections, it was decided to try a ‘hybrid’ injection strategy. Specifically, a double homogeneous/stratified injection strategy was devised to try to control the flow field and mixture around ignition timing; this consisted of injecting ~75% of the fuel in the intake stroke at the same timing as standard single injection (SOI 80° CA ATDC) while the final 25% was injected just before ignition (SOI 45° CA BTDC) such that a very short and precise fuel spray was delivered past the spark plug by plumes #1 and #6, leaving not only a trail of fuel droplets but also a turbulent field of eddies behind the spray, in which enhanced ignition could take place.

The spark images of this strategy are shown in **Fig. 37**. In this figure, the spark is being rapidly displaced in the direction of the spray plumes *i.e.* across the combustion chamber, orthogonal to the tumble plane (*i.e.* into the page relative to the figure). This was indicated by the clear shadow of the central electrode tip. The sparks were displaced by eddies in the wake of the two spray plumes. However, as will be shown later, once the flame kernels grew to larger sizes, these were convected across to the exhaust side of the combustion chamber, similarly to the standard single injection strategy. Although the second injection was timed to affect the ignition event, it was clear that the different scales of the flow affected the flame in different ways, depending on its size. Indeed this bias was shown by the RMS image, where the outline of the central electrode is clearly identifiable and the sparks can be seen to grow into the page and then to the right side of the image, corresponding to the exhaust side of the combustion chamber. Multiple discharges at the base of the weld of the ground electrode were also seen for 3% of the cycles, probably due to fuel impingement in these locations combined with excessive local flow velocities in the spark-plug gap. The tests showed that such flow field conditions at ignition timing, combined with the heterogeneous nature of the mixture, resulted in significantly improved rate of pressure rise, which resulted in higher peak pressures by ~17% compared to single injection, at otherwise similar operating conditions and ignition timing.

Combustion images with the single and double-hybrid injection strategy are shown in **Fig. 38**. These illustrate that the flames of the double-hybrid strategy were brighter and exhibited ‘hot’ spots presumably due to the partially stratified nature of the mixture, including regions which did not have time to fully vaporize and mix before the combustion event and burned as small pool fires, particularly on the piston crown. **Fig. 39** shows the in-cylinder pressure traces and MFB of both injection strategies. An equivalent lean operating condition is also shown for comparison to highlight how such strategies could be used to obtain improved engine performance even under overall lean air/fuel ratios *i.e.* with further potential for fuel consumption and CO₂ emission reduction benefits. The level of IMEP also increased by about 30–40%, depending on λ . Figures quantifying the improvement in the rate of burning via the calculation of the ensemble average flame size (as viewed through the piston crown) and the effect on the average flame motion in the cylinder for these conditions versus standard single injection, are shown in **Fig. 40**. In this figure, it is clear that the flames with the hybrid injection strategy had a preferential direction of motion that seems to have been dictated by the directionality of plumes 1 and 6 as they passed by the spark plug (**Fig. 36**), and this further highlights the control potential of such hybrid injection strategies in terms of both fuel concentration and local flow characteristics.

SUMMARY AND CONCLUSIONS

This paper presented results from a research programme set up to resolve the effect of fuel type and injection strategy on spray development, wall wetting, mixture formation and combustion in DISI engines. The focus of the investigation was a combustion system with a centrally located multi-hole injector in close proximity to the spark plug. Specifically, a detailed imaging study of spray development with gasoline, *iso*-octane; ethanol, *n*-butanol and E10 was carried out. Several key operating conditions were examined, *e.g.* single and multiple, *i.e.* ‘split’, injection strategies, as well as different engine head temperatures (20–90 °C) and engine loads (low load and full load with 0.5 bar and 1 bar inlet plenum pressure, respectively). The injection pressure was fixed at 150 bar and the engine speed at 1500 RPM. The main conclusions of this work can be summarised as follows:

- Spray development was quite robust on a cycle-by-cycle basis, with flow interaction effects lowest along the plume periphery and highest at the spray tips. However, the cycle-to-cycle differences in the spray’s development were larger than the very small injection-to-injection differences observed in a quiescent chamber, hence previously quantified cyclic variations in the incoming air flow were mainly responsible for the spray’s in-cylinder behaviour. Strong in-cylinder tumble flows were effective in breaking up the spray by cross-flow momentum exchange. Certain spray plumes were noticeable distorted and deflected by the intake valve flow by as much as 30° from their original trajectories.
- *Iso*-octane’s spray plumes demonstrated the closest agreement with their nominal injector nozzle hole angles, whilst gasoline’s plumes showed mean trajectories that deviated by as much as 25% from their nominal design angles towards a vertical direction. This is believed to be due to better break-up and atomisation of gasoline’s plumes when interacting with the incoming flow.
- Significantly altered spray development occurred for gasoline at low-load engine conditions (0.5 bar inlet plenum pressure) combined with high fuel temperatures (90 °C). Flash-boiling of the high volatility components of gasoline disrupted the spray plumes causing them to ‘collapse’ into the low pressure regions below the injector axis. This behaviour was not observed for *iso*-octane at the same conditions, which was found to produce similar macro-scale spray development to gasoline at 20 °C but with increased evaporation rates.
- Timing the single injection event per cycle to start towards maximum intake valve lift was found to result in direct spray impingement on one of the intake valves. This caused large amounts of secondary atomisation from some plumes being partially deflected into the intake port and re-entering the cylinder or as fuel films stripped off the back of the valve.
- There was a trade-off between earlier injection timings that reduced valve interactions but increased piston impingement and later injections in the intake stroke which reduced piston impingement but increased valve interactions and resulted in less time for fuel evaporation before ignition. An injection timing of 80 °CA ATDC was a compromise between spray liner/piston impingement, available time post injection for evaporation, as well as ‘optimum’ phasing with regards to valve timing that promoted strong spray/flow interactions for efficient break up and atomisation.
- At full-load operating conditions (1.0 bar inlet plenum pressure), the different volatilities of *iso*-octane and gasoline did not manifest themselves with the same clarity as at low load. The sprays of both fuels were in general very similar macroscopically. Specifically, the sprays did not break-up initially as quickly, but the spray tip penetration was reduced by about 10% due to the higher in-cylinder gas density. However, as the spray

developed, the interactions with the stronger in-cylinder bulk flow resulted in prominent momentum exchange, more air entrainment into the sprays and more rapid plume break-up than at low-load conditions

- Although injection pulsewidths were nearly double at full load to maintain stoichiometric operation, levels of liquid piston impingement were clearly reduced as the bulk flow tumble motion in conjunction with the lower spray penetration kept the atomised fuel away from in-cylinder wall surfaces. No spray collapse was observed for gasoline at 90 °C, although the spray plumes were noticeably more ‘swollen’ than those of *iso*-octane. Nonetheless, as the fuel atomised at the spray tips, *iso*-octane clearly evaporated faster, demonstrating the negative effect of gasoline’s higher boiling point components on this occasion.
- The alcohols were slower to exit the injector nozzle and produced thinner more compact spray plumes. At 20 °C, low load, the levels of plume ‘flapping’ were greater with gasoline and *iso*-octane than for both alcohols, as indicated by the regions of variation along the outer region of the plumes. The locations of highest variation tended to be limited towards the spray tips for the alcohols.
- Similarities were observed between the locations of spray variation for ethanol and gasoline at 90 °C low load, although gasoline was seen to collapse marginally more. The sprays of *n*-butanol did not collapse even at 90 °C low load and showed larger regions of variation than at 20 °C, suggesting improved atomisation from lower surface tension and viscosity effects coupled to stronger air entrainment.
- At 20 °C low load the E10 spray was found to be macroscopically no different to gasoline. However, the sprays of E10 at 70 °C resembled closely those of gasoline at 90 °C, an effect related to the ~8 °C superheat experienced by the ethanol component of E10 at 70 °C when injected into the gas pressure of 0.5 bar in-cylinder. At 90 °C, the E10 sprays were clearly more collapsed than those of gasoline immediately out of the nozzle.
- Splitting the injection event into three smaller ‘split’ injections in the early intake stroke to achieve the same overall stoichiometry was observed to improve fuel evaporation and mixture homogeneity. Each one of the short split injection events showed clear spray collapse for gasoline at 90 °C, similarly to the longer injection event of the single injection strategy.
- The spark discharge exhibited a highly variable geometry that was significantly influenced by the flow field on a cycle-to-cycle basis, as demonstrated by close-up imaging at quiescent and running engine conditions. Changing the injection timing and strategy in the intake stroke showed differences in the spark’s behaviour at late compression, which appeared to be coupled to changes in the flow field at spark timing.
- A double hybrid homogeneous/stratified injection strategy (75% of the fuel delivered in the intake stroke and 25% just before spark timing) indicated that there was potential to manipulate the characteristics of both fuel stratification and local flow field at ignition timing. Combustion imaging analysis quantified the preferential direction of flame development for stoichiometric and lean fuelling with this strategy.
- Optimisation of the number of split injections in the intake stroke and close to ignition timing merits attention as this can promote mixing, eliminate wall wetting, and enhance flame growth, targeting reduced CO₂ emissions and particulate matter with existing injector technology.

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REFERENCES

1. Zhao, F., Lai, M.C., and Harrington, D.L., “Automotive spark-ignition direct-injection gasoline engines”, *Progress in Energy and Combustion Science*, 1999, Vol. 25, pp. 437–562.
2. Stach, T., Schlerfer, J., and Vorbach, M., “New generation multi-hole injector for direct-injection SI engines”, SAE Paper 2007-01-1404, 2007.
3. Serras-Pereira, J., Aleiferis, P.G., Richardson, D., and Wallace, S., “Mixture Formation and Combustion Variability in a Spray-Guided DISI Engine”, *SAE Transactions, Journal of Engines*, Vol. 116, No. 3, 1332–1356, SAE Paper 2007-01-4033, 2007.
4. Serras-Pereira, J., Aleiferis, P.G., Richardson, D., and Wallace, S., “Spray Development, Flow Interactions and Wall Impingement in a Direct-Injection Spark-Ignition Engine”, SAE Congress Brazil, Sao Paulo, Brazil, SAE Paper 2007-01-2712, 2007.
5. Oh, H., Bae, C., Park, J. and Jeon, J., “Effect of the Multiple Injection on Stratified Combustion Characteristics in a Spray-Guided DISI Engine”, SAE Paper 2011-24-0059, 2011.
6. Fan, Q. and Li, L., “Transient Characteristics of Cold Start Emissions from a Two-Stage Direct Injection Gasoline Engines Employing the Total Stoichiometric Ratio and Local Rich Mixture Start-up Strategy”, SAE Paper 2012-01-1068, 2012.
7. Serras-Pereira, J., Aleiferis, P.G., and Richardson, D., “An Analysis of the Combustion Behaviour of Ethanol, Butanol, *iso*-Octane, Gasoline and Methane in a Direct-Injection Spark-Ignition Research Engine”, *Combustion Science and Technology*, 2013, Vol. 185, pp. 484–513.
8. Schmitz I. and Leipertz, A., “Comparison of the Flash Boiling Influence on the Spray Structure of a High Pressure Swirl Injector and of a Multihole Injector for GDI engines”, 10th ICLASS, Kyoto, Japan, 2006.
9. Arora, R., Morgan, C.J., Naber, J.D., and Lee, S.-Y., “Flash Boiling Spray Characterization of a Gasoline Multi-hole Injector in a Heated Pressure Vessel”, 23rd ILASS Americas, Ventura, USA, 2011.
10. Blessinger, M. and Meijer, M., Pickett, L.M., Manin, J., and Skeen, S., “Liquid/Vapor Penetration and Plume-Plume Interaction of Vaporizing Iso-octane and Ethanol SIDI Sprays”, 25th ILASS Americas, Pittsburgh, USA, 2013.
11. Zeng, W., Xu, M., Zhang, G., Zhang, Y., and Cleary, D.J., “Atomization and vaporization for flash-boiling multi-hole sprays with alcohol fuels”, *Fuel*, 2012, Vol. 95, pp. 287–297.
12. Zhang, M., Xu, M., and Zeng, W., “Flow Field Characterization of Superheated Sprays from a Multi-hole Injector by Using High-speed PIV”, SAE Paper 2012-01-0457, 2012.
13. Zhang, M., Xu, M., Zhang, Y., Zhang, G., and Cleary, D.J., “Flow-field investigation of Multihole Superheated Sprays using High-speed PIV. Part I. Cross-Sectional Direction”, *Atomization and Sprays*, 2012, Vol. 22, pp. 983–995.

14. Zhang, M., Xu, M., Zhang, Y., Zhang, G., and Cleary, D.J., “Flow-field investigation of Multihole Superheated Sprays using High-speed PIV. Part II. Axial Direction”, *Atomization and Sprays*, 2013, Vol. 23, pp. 119–140.
15. Jarvis, S., Justham, T., Clarke, A., Garner, C.P., Hargrave, G.K., and Richardson, D., “Motored SI IC Engine In-Cylinder Flow Field Measurement Using Time Resolved Digital PIV for Characterisation of Cyclic Variation”, SAE Paper 2006-01-1044, 2006.
16. Justham, T., Jarvis, S., Garner, C.P., Hargrave, G.K., Clarke, A., and Richardson, D., “Single Cylinder Motored SI IC Engine Intake Runner Flow Measurement using Time Resolved Digital Particle Image Velocimetry”, SAE Paper 2006-01-1043, 2006.
17. Rimmer, J.E.T., Long, E.J., Garner, C.P., Hargrave, G.K., Richardson, D., and Wallace, S., “The Influence of Single and Multiple Injection Strategies on In-Cylinder Flow and Combustion within a DISI Engine”, SAE Paper 2009-01-0660, 2009.
18. Williams, B., Ewart, P., Stone, R., Ma, H., Walmsley, H., Cracknell, R., Stevens, R., Richardson, D., Qiao, J., and Wallace, S., “Multi-Component Quantitative PLIF: Robust Engineering Measurements of Cyclic Variation in a Firing Spray-Guided Gasoline Direct Injection Engine”, SAE Paper 2008-01-1073, 2008.
19. Williams, B., Ewart, P., Wang, X., Stone, R., Ma, H., Walmsley, H., Cracknell, R., Stevens, R., Richardson, D., Fu, H., and Wallace, S., “Quantitative planar laser-induced fluorescence imaging of multi-component fuel/air mixing in a firing gasoline-direct-injection engine: Effects of residual exhaust gas on quantitative PLIF”, *Combustion and Flame*, 2010, Vol. 157, pp. 1866–1878.
20. Serras-Pereira, J., Aleiferis, P.G., Richardson, D., and Wallace, S., “Characteristics of Ethanol, Butanol, *Iso*-octane and Gasoline Sprays and Combustion from a Multi-Hole Injector in a DISI Engine”, *SAE International Journal of Fuels and Lubricants*, 2008, Vol. 1, pp. 893–909, SAE Paper 2008-01-1591.
21. Serras-Pereira, J., Aleiferis, P.G., and Richardson, D., 2010, “Imaging and Heat Flux Measurements of Impinging Sprays of Liquid Hydrocarbons and Alcohols in a Motoring and Firing Direct-Injection Spark-Ignition Engine”, *Fuel*, 2012, Vol. 91, pp. 264–297.
22. Serras-Pereira, J., Aleiferis, P.G., Walmsley, H.L., Davies, T.J., and Cracknell, R.F., “Heat Flux Characteristics of Spray Wall Impingement with Ethanol, Butanol, *iso*-Octane, Gasoline and E10 Fuels”, *International Journal of Heat and Fluid Flow*, 2013, Vol. 44, pp. 662–683.
23. van Romunde, Z., Aleiferis, P.G., Cracknell, R.F., and Walmsley, H.L., “Effect of Fuel Properties on Spray Development from a Multi-Hole DISI Engine Injector”, *SAE Transactions, Journal of Engines*, 2007, Vol. 116, No. 3, pp. 1313–1331, SAE Paper 2007-01-4032.
24. van Romunde, Z. and Aleiferis, P.G., “Effect of Operating Conditions and Fuel Volatility on Development and Variability of Sprays from Gasoline Direct-Injection Multi-Hole Injectors” *Atomization and Sprays*, 2009, Vol. 19, pp. 207–234.
25. Aleiferis, P.G. and van Romunde, Z., “An Analysis of Spray Development with *iso*-Octane, *n*-Pentane, Gasoline, Ethanol and *n*-Butanol from a Multi-Hole Injector under Hot Fuel Conditions”, *Fuel*, 2013, Vol. 105, pp. 143–168.

26. Serras-Pereira, J., van Romunde, Z., Aleiferis, P.G., Richardson, D., Wallace, S., and Cracknell, R.F., Cavitation, primary break-up and flash boiling of gasoline, *iso*-octane and *n*-pentane with a real-size optical direct-injection nozzle, *Fuel*, 2010, Vol. 89, pp. 2592–2607.
27. Aleiferis, P.G., Serras-Pereira, J., Augoye, A., Davies, T.J., Cracknell, R.F., and Richardson, D., “Effect of Fuel Temperature on In-Nozzle Cavitation and Spray Formation of Liquid Hydrocarbons and Alcohols from a Real-Size Optical Injector for Direct-Injection Spark-Ignition Engines”, *International Journal of Heat and Mass Transfer*, 2010, Vol. 53, pp. 4588–4606.
28. Vermorel, O., Richard, S., Colin, O., Angelberger, C., Benkenida, A., and Veynante, D., “Towards the understanding of cyclic variability in a spark ignited engine using multi-cycle LES”, *Combustion and Flame*, 2009, Vol. 156, pp. 1525–1541.
29. Enaux, B., Granet, V., Vermorel, O., Lacour, C., Pera, C., Angelberger, C., and Poinso, T., “LES study of cycle-to-cycle variations in a spark ignition engine”, *Proceedings of the Combustion Institute*, 2011, Vol. 33, pp. 3115–3122.
30. Lacour, C. and Pera, C., “An Experimental Database Dedicated to the Study and Modelling of Cyclic Variability in Spark-Ignition Engines with LES”, *SAE Technical Paper 2011-01-1282*, 2011.
31. Granet, V., Vermorel, O., Lacour, C., Enaux, B., Dugué, V., and Poinso, T., “Large-Eddy Simulation and experimental study of cycle-to-cycle variations of stable and unstable operating points in a spark ignition engine”, *Combustion and Flame*, 2012, Vol. 159, pp. 1562–1575.
32. Peterson, B. and Sick, V., “High-Speed Flow and Fuel Imaging Study of Available Spark Energy in a Spray-Guided Direct-Injection Engine and Implications on Misfires”, *International Journal of Engine Research*, 2010, Vol. 11, pp. 313 0–329.
33. Fajardo, C. and Sick, V., “Flow field assessment in a Fired Spray-guided spark ignition direct-injection engine based on UV particle image velocimetry with sub crank angle resolution”, *Proceedings of the Combustion Institute*, 2007, Vol. 31, pp. 3023–3031.
34. Twiney, B., Stone, R., Chen, X., and Edmunds, G., “Investigation of Combustion Robustness in Catalyst Heating Operation on a Spray Guided DISI Engine, Part I - Measurements of Spark Parameters and Combustion”, *SAE Paper 2010-01-0593*, 2010.
35. Twiney, B., Stone, R., Chen, X., and Edmunds, G., “Investigation of Combustion Robustness in Catalyst Heating Operation on a Spray Guided DISI Engine, Part II - Measurements of Spray Development, Combustion Imaging and Emissions”, *SAE International Journal of Engines*, 2010, Vol. 3, pp. 426–439, *SAE Paper 2010-01-0603*.
36. Butcher, A.J., Aleiferis, P.G., and Richardson, D., “Development of a Real-Size Optical Injector Nozzle for Studies of Cavitation, Spray Formation and Flash Boiling at Conditions Relevant to Direct-Injection Spark-Ignition Engines”, *International Journal of Engine Research*, 2013, Vol. 14, pp. 557–577.
37. Aleiferis, P.G., Serras-Pereira, J., van Romunde, Z., Caine, J., and Wirth, M., “Mechanisms of Spray Formation and Combustion from a Multi-Hole Injector with E85 and gasoline”, *Combustion and Flame*, 2010, Vol. 157, pp. 735–756.

38. Aleiferis, P.G., Serras-Pereira, J., and Richardson, D., “Characterisation of Flame Development with Ethanol, Butanol, *Is*o-Octane, Gasoline and Methane Fuels in a Direct-Injection Spark-Ignition Engine”, *Fuel*, 2013, Vol. 109, pp. 256–278.
39. Gilles-Birth, Bernhardt, S., Spicher, U., and Rechs, M., “A Study of the In-Nozzle Flow Characteristics of Valve-Covered Orifice Nozzles for Gasoline Direct Injection”, SAE Paper 2005-01-3684, 2005.
40. Gilles-Birth, I., Rechs, M., Spicher, U., and Bernhardt, S., “Experimental Investigation of the In-nozzle Flow of Valve Covered Orifice Nozzles for Gasoline Direct Injection”, *Proceedings of the 7th International Symposium on IC Diagnostics*, pp. 59–78, Baden-Baden, Germany, 2006.
41. Heldmann, M., Knorsch, T., Schmitz, I., Wensing, M., and Leipertz, A., “Investigation of significant Spray Rotation Phenomena under Flash-Boiling Conditions studied on a Multi-hole DISI Injector for Bio-Ethanol E85 and Gasoline E5”, *ILASS – Europe 2011, 24th Annual Conference on Liquid Atomization and Spray Systems*, Estoril, Portugal, 2011.
42. Aleiferis, P.G., Hardalupas, Y., Kolokotronis, D., Taylor, A.M.K.P., Arioka, A., and Saito, M., “Experimental Investigation of the Internal Flow Field of a Model Gasoline Injector Using Micro-Particle Image Velocimetry”, *SAE Transactions, Journal of Fuels and Lubricants*, 2006, Vol. 115, No. 4, pp. 597–606.
43. Kolokotronis, D., Hardalupas, Y., Taylor, A.M.K.P., Aleiferis, P.G., Arioka, A., and Saito, M., “Experimental Investigation of Cavitation in Gasoline Injectors”, SAE Paper 2010-01-1500, 2010.
44. Schwarz, Ch., Schünemann, E., Durst, B., Fischer, J., and Witt, A., “Potentials of the Spray-Guided BMW DI Combustion System”, SAE Paper 2006-01-1265, 2006
45. Aleiferis, P.G., Taylor, A.M.K.P., Whitelaw, J.H., Ishii, K., and Urata, Y., “Cyclic Variations of Initial Flame Kernel Growth in a Honda VTEC-E Lean-Burn Spark-Ignition Engine”, SAE Paper 2000-01-1207, 2000.
46. Anderson, R.W. “The Effect of Ignition Power on Fast-Burn Engine Combustion”, SAE Paper 870549, 1987.
47. Herweg, R., Begleris, P., Zettlitz, A., and Ziegler, G.F.W., “Flow Field Effects on Flame Kernel Formation in a Spark-Ignition Engine”, SAE Paper 881639, 1988.
48. Cho, Y. S., Santavicca, D. A., and Sonntag, R. M., “The Effect of Spark Power on Spark Ignited Flame Kernel Growth”, SAE Paper 922168, 1992.
49. Kalghatgi, G.T., “Flame Initiation and Development from Glow Discharges – Effect of Electrode Deposits”, *Combustion and Flame*, 1989, Vol. 77, pp. 321–336.
50. Ziegler, G.F.W., Schaudt, R.A., and Herweg, R., “The Influence of Spark Plug Orientation on Flame Kernel Formation in Flowing Mixtures”, *Proceedings of XXI FISITA*, pp. 1.225–1.232, Belgrade, 1986.
51. Sick, V., Drake, M.C. and Fansler T.D., “High-speed imaging for direct injection gasoline engine research and development”, *Experiments in Fluids*, Vo. 49, pp. 937–947, 2010.
52. Scharlab S.L., Data Sheet 2,2,4-Trimethylpentane, anhydrous, 2011.
53. Shell Global Solutions (UK), Personal Communication, 2005.
54. Poling, B.E., Prausnitz, J.M., and O’Connell, J.P., “The Properties of Gases and Liquids”, 5th Edition, McGraw-Hill, 2001.

55. Owen, K. and Coley, T., "Automotive Fuels Reference Book", Society of Automotive Engineers, 1995.
56. Yaws, C.L., "Yaws' Handbook of Thermodynamic and Physical Properties of Chemical Compounds", Knovel, 2003.
57. Gupta, R.B. and Demirbas, A., "Gasoline, Diesel and Ethanol Biofuels from Grasses and Plants", Cambridge University Press, 2010.

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Table 1. Research Engine Specifications and Conditions.

Cylinders, Cycle	1, 4-Stroke
Valves	2 Intake, 2 Exhaust
Bore, Stroke	89.0 mm, 90.3 mm
Compression Ratio	11.15:1
Valve Timings	IVO 24°, IVC 274°, EVO 476°, EVC 6°
Optical Piston Crown	65 mm in diameter
Engine Speed	1500 RPM
Engine Load	0.5 bar and 1.0 bar inlet plenum pressure
Engine Head Temperature	20–90 °C
Spark Advance	35° CA
Air-to-Fuel Ratio	$\lambda=1.0$ and 1.2 (air excess ratio)

Table 2. Fuel Properties ^(a).

Properties	Gasoline	<i>iso</i> -Octane	Ethanol	<i>n</i> -Butanol
Chemical formula	C _{6.75} H _{12.99}	C ₈ H ₁₈	C ₂ H ₅ OH	C ₄ H ₉ OH
Density [g/cm ³]	0.72	0.69	0.79	0.81
Boiling point [°C]	~55–145 (10–90%)	99	78.3	117.8
Flash point [°C]	-43	-12	12	30
Reid vapour pressure [kPa]	56	11.8 ^(b)	16.1 ^(b)	2.2 ^(b)
Dynamic viscosity (20 °C) [cP]	0.37–0.44 ^(c)	0.51	1.2	2.95
Dynamic viscosity (80 °C) [cP]	–	0.27 ^(d)	0.43 ^(d)	0.73 ^(d)
Kinematic viscosity (25 °C) [cSt]	0.5–0.58	0.67 ^(d)	1.34 ^(d)	3.22 ^(d)
Kinematic viscosity (80 °C) [cSt]	–	0.42 ^(d)	0.59 ^(d)	0.96 ^(d)
Surface tension (20 °C, 80 °C) [mN/m]	~20, –	18.3 ^(d) , 13.6 ^(d)	24.05 ^(d) , 16.4 ^(d)	26.3 ^(d) , 19.3 ^(d)
Latent heat (T _{boil}) [kJ/kg]	364	272 ^(d)	855 ^(d)	584 ^(d)
Latent heat (25 °C) [kJ/kg]	380–500	300 ^(d)	874 ^(d)	669 ^(d)
Stoichiometric AFR	14.6	15.1	9	11.1
Heating value [MJ/kg], [MJ/lt]	42.7, 32 ^(c)	44.6, 30.8 ^(d)	26.9, 21.3 ^(d)	33.9, 27.5 ^(d)
Specific energy [MJ/kg air]	2.92	2.94	2.99	3.01
RON	95	100	129 ^(e)	96 ^(e)

(a) If not specified differently, data taken from [52, 53]; (b) [54]; (c) [55]; (d) [56]; (e) [57].

Table 3. Typical Injection Pulsewidths for Stoichiometric Fuelling (gasoline, 0.5 bar).

Injection Pulse Durations [ms]		
Single	Triple	Double Hybrid
0.90	0.3, 0.3, 0.32	0.75, 0.25

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