MECHANISMS OF SPRAY FORMATION AND COMBUSTION FROM A MULTI-HOLE INJECTOR WITH E85 AND GASOLINE

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ABSTRACT

The spray formation and combustion characteristics of gasoline and E85 (85% ethanol, 15% gasoline) have been investigated using a multi-hole injector with asymmetric nozzle-hole arrangement. Experiments were carried out in a quiescent optical chamber using high-speed shadowgraphy (9 kHz) to characterise the spray sensitivity to both injector temperature and ambient pressure in the range of 20–120 °C and 0.5, 1.0 bar. Spray tip penetrations and ‘umbrella’ spray cone angles were calculated for all conditions. Phase Doppler anemometry was also used to measure droplet sizes in the core of one of the spray plumes, 25 mm below the injector tip. To study the effect of fuel properties on vaporisation and mixture preparation under realistic operating conditions, a separate set of experiments was carried out in a direct-injection spark-ignition optical engine. The engine was run at 1500 RPM under cold and fully warmed-up conditions (20 °C and 90 °C) at part load and full load (0.5 and 1.0 bar intake pressure). Floodlit laser Mie-scattering images of the sprays on two orthogonal planes corresponding to the swirl and tumble planes of in-cylinder flow motion were acquired to study the full injection event and post-injection mixing stage. These were used to make comparisons with the static chamber sprays and to quantify the liquid-to-vapour phase evaporation process for both fuels by calculating the projected ‘footprint’ of the sprays at different conditions. Analysis of the macroscopic structure and turbulent primary break-up properties of the sprays was undertaken in light of jet exit conditions described in terms of non-dimensional numbers. The effects on stoichiometric combustion were investigated by imaging the natural flame chemiluminescence through the engine’s piston crown (swirl plane) and by post-processing to derive flame growth rates and trajectories of flame motion.

NOMENCLATURE

AFR	Air-to-Fuel Ratio
ASOI	After Start of Injection
ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
CA	Crank Angle
d	Nozzle Diameter
DISI	Direct Injection Spark Ignition
EOI	End of Injection
L	Length
MFB	Mass Fraction Burned
μ	Dynamic Viscosity
Oh	Ohnesorge Number
PDA	Phase Doppler Anemometry
Re	Reynolds Number
RON	Research Octane Number
RPM	Revolutions Per Minute
ρ	Density
SI	Spark Ignition
SMD	Sauter Mean Diameter ($D_{3,2}$)
SOI	Start of Injection
σ	Surface Tension
TDC	Top Dead Centre
u	Velocity
W	Width
We_G	Weber Number, Gas Phase
We_L	Weber Number, Liquid Phase

INTRODUCTION

Background

The gasoline Direct-Injection Spark-Ignition (DISI) engine has many well-known practical advantages over the port-fuel injected engine, such as the greater precision in fuel metering and significant potential fuel economy benefits. Other advantages, such as greater charge cooling potential allow higher engine compression ratios to be used, increasing the thermal efficiency. However over the last 10 years it has become clear that the successful implementation of all of these benefits has been more difficult to achieve than previously expected. The main reason has been the ‘inadequate’ design and performance of the direct-injection systems, as well as the engine calibration strategies used, which have not delivered the level of improvements in power, fuel economy and emissions promised by manufacturers. For example, first generation side-injection systems relied on a piston-bowl to achieve the ‘stratified charge’ concept, but were plagued by pool fires on the piston as the fuel failed to evaporate fully in the time available before ignition, thus producing unacceptable levels of particulate emissions and unburned hydrocarbons [1].

Latest design spray-guided multi-hole injectors located in close spacing arrangement with the spark plug aim to resolve these issues by providing accurate spray directionality as a result of the number of nozzle holes which can be designed to target the fuel delivery where necessary and by full flexibility of when and how fuel is injected into the cylinder *i.e.* independence over piston position and use of multiple ‘split’ fuel injection strategies [2–4]. The majority of work published to date on multi-hole injectors concerns Diesel nozzle geometries, with particular experimental features and analysis targeted at this combustion system. The sensitivities to geometrical differences of the injector nozzles and piston design, the varied operating conditions for the DISI combustion system and its particular mixture preparation requirements merit specific attention and are the motives behind continued research in this field.

Because of the ability of multi-hole injectors to target fuel directly to the spark-plug for stratified operation and to different areas around the cylinder for robust mixture formation under early injection homogeneous mode, it is important to understand whether strict design criteria are maintained when fluids other than those used to test the injectors at design and optimisation stage are employed in real engines. These requirements are further emphasised by the trend towards using sustainable renewable fuels such as ethanol in increasing proportions with gasoline. One of the drawbacks of ethanol as an automotive fuel is its lower energy density compared to gasoline and lower stoichiometric Air/Fuel Ratio (AFR) which increases the fuel mass that must be injected per cycle to achieve the same fuel equivalence ratio (typical AFR at stoichiometry for gasoline is 14.7, for pure ethanol 9.0). To deal with these issues, either the flow rate of injectors has to be increased, which may reduce the accurate fuel metering capabilities required for ‘split’ injection strategies and stratified operation, or standard injectors are typically used and the fuel is injected over a longer period of time during the cycle, increasing the probability of wall-wetting and reducing the effective time available for mixing and evaporation before ignition. There is also a limited understanding about the role that liquid transport properties such as surface tension (σ), viscosity (μ), density (ρ), boiling point and vapour pressure (p_v) have

on overall spray development, and how these are affected by the wide operating envelope of DISI engine injectors; *e.g.* DISI injectors must inject fuel into ambient pressures from ~0.2 bar to ~5 bar or more and with liquid temperatures of -10 °C or lower, to over 150 °C at the injector tip under high-load firing conditions.

Present Contribution

The current study presents macroscopic imaging of spray structure and quantification of spray properties, including spray tip penetration, cone angles and droplet sizes from tests carried out in a static pressure chamber with a multi-hole injector and two multi-components fuels, namely standard commercial grade gasoline (RON 95) and a commercial grade of E85 (85% ethanol mixture with 15% gasoline) at 0.5–5.0 bar gas pressure and 20–120 °C injector temperature conditions. Spray images are also presented for the same injector and fuel types in a single-cylinder optical engine under motoring condition with injection in the intake stroke to promote homogeneous mixture formation under 0.5 bar and 1.0 bar intake pressure with 20 and 90 °C engine coolant temperature. The evaporation history of the two fuels was also tracked by monitoring the scattering intensity during and after the injection period. Finally combustion data, such as in-cylinder pressure history and Mass Fraction Burned (MFB), as well as flame-kernel radius growth and motion from image processing, are presented for the same engine operating conditions in order to compare the high-ethanol content fuel with gasoline. A set of experiments is also presented with *iso*-octane, a single-component fuel with known properties typically used in engine testing, spray experiments and modelling.

The work is believed to contribute significantly towards a comprehensive database of spray development data which are essential for developing our knowledge of the underlying mechanisms responsible for atomisation under realistic engine conditions. Such a database will also assist spray modellers because accurate simulation of spray break-up mechanisms from first principles is very challenging. The discussion also makes an attempt to review fundamental theories and non-dimensional numbers from established background literature and to link these to practical combustion systems with different fuels. To the best of our knowledge, currently there is no other major optical study in the literature presenting and discussing spray formation and combustion phenomena with E85 and gasoline from a multi-hole injector both in a quiescent injection environment and in a firing engine under a range of operating conditions.

EXPERIMENTAL APPARATUS AND PROCEDURES

Fuels

Standard commercial grade gasoline (RON95) and a commercial grade of E85 (85% ethanol mixture with 15% gasoline) were used for the present study. Table 1 shows a number of selected properties which are relevant to the analysis of spray formation, evaporation and combustion of the fuels used; they are given for 1.0 bar ambient pressure. The properties of E85 were obtained from the fuel supplier specification sheet, except for the latent heat of vaporization which was estimated on the basis of values for pure ethanol (0.925 MJ/kg) and those of gasoline (0.360 MJ/kg). Figure 1 shows typical distillation curves of gasoline and E85 at 1.0 bar pressure. The distillation curve of ethanol is also shown in Figure 1 by a vertical line on the assumption of 100% purity and, thus, a fixed boiling temperature is indicated. It needs to be noted here that

although the E85 data in Figure 1 correspond to the exact fuel used in the current study, the gasoline curve does not correspond to the exact gasoline type used in this study but it has been reproduced from a previous publication to aid visual interpretation of data in Table 1 [5]. A standard commercial grade European gasoline contains several hydrocarbons, typically ~35–40% C5 or lower, similar levels of C6–C8 and the remainder C9–C10 hydrocarbon chains. On the basis of these fractions, it was also decided to select specific single components to aid the discussion and provide some interpretation of the results to be presented in later sections. From typical gas-chromatography analysis data, the reference single-component fuels were chosen to be *n*-pentane, *iso*-octane, and *o*-xylene, as also shown in Figure 1. The first one, *n*-pentane, is a typical low boiling point component found in gasoline, with boiling temperature 36.1 °C. The second one, *iso*-octane, is a mid-range boiling point component with boiling temperature 99 °C and finally *o*-xylene is a heavier gasoline component with boiling temperature 144 °C. These together with similar components like *iso*-pentane, *n*-octane, heptanes, *p*-xylene, *etc.* account for over 95% of the constituents by volume. A small set of spray imaging data was also acquired with pure *iso*-octane (typical single-component fuel used in engine research) to be studied as a benchmark set against the data acquired with the two multi-component fuels.

Injector Geometry

The multi-hole injector produced six spray plumes or jets in a ¾ moon pattern as shown in the schematic illustration of Figure 2. The injector has been designed for vertical installation in the engine and in close spacing arrangement with the spark plug. The six nozzle holes have different angles to direct fuel to different areas of the combustion chamber. It can be seen that there is one line of symmetry in the spray pattern in the axis perpendicular to the engine ‘crossflow’ axis, *i.e.* intake to exhaust side, with plumes 1 and 6 directing the fuel towards the spark plug, as shown in Figure 2(a). Two nominally identical injectors of this type were used for the purposes of the present study; one for experiments in a quiescent optical chamber and the other one for motoring and firing tests in an optical engine. In both rigs the injector was operated with 150 bar fuel pressure and the fuel was provided by a pneumatic piston ram pump which avoided pressure fluctuations in the fuel rail. The injector used in the engine was also checked in the quiescent chamber to ensure consistent operation.

Quiescent Chamber

A pressure chamber was used to de-couple the effects of engine intake flow on spray break-up. The octagonal shape of the chamber allowed simultaneous multi-technique characterisation, in this case imaging and off-axis Phase Doppler Anemometry (PDA). More details about the chamber can be found in [6].

Imaging

The injector was mounted at 19° to the vertical in order to optimise the imaging arrangement (in contrast to the vertical engine mounting shown earlier). Figure 2(b) shows a schematic of this arrangement with the two imaging views adopted. The spray was recorded at a frame rate of 9,000 frames per second, equivalent to one spray image at each Crank Angle degree (°CA) for an engine running at 1500 RPM. The camera allowed a maximum spatial resolution of 640×480 pixels at this frame rate (160 µm/pixel). The shutter was set to 1 µs.

Illumination was achieved with backlighting using a Multiblitz Variolite 500 photographic flash-lamp diffused through a semi opaque Perspex sheet. As a result, each imaged jet was the superposition of two liquid jets for the ‘side’ view, and three for the ‘end’ view, as shown in Figure 2(b). The image was a shadowgraph of the spray, therefore only the liquid phase was visualised with this technique. 100 injection events and an average background image were captured. The background image for each time interval in the imaging sequence was removed from the spray images to account for differences in lighting over each run. Each corrected image was then thresholded at a value based on the mean of background pixels to leave a binary image. The image was then rotated to align each plume pair with the vertical axis and each plume pair was ‘scanned’ to find the plume tip. The distance from the plume tip to the nozzle was scaled to calculate a plume pair length. To determine the optimum threshold for automated processing, detailed sensitivity analysis was carried out on the penetration based on different threshold levels and the uncertainty in the calculated values was found to be better than 4%. A similar methodology was used to calculate overall spray ‘umbrella’ cone angles.

Droplet Sizing

Droplet sizing of the spray was carried out using a TSI Phase Doppler Anemometry system consisting of a Coherent Innova 70C Argon-Ion laser (5 W) coupled to a TSI beam splitter and Bragg cell. Both the transmitter and receiver had an optical focal length of 250 mm. The forward scattering angle was set to 40°. Droplet measurements were recorded 25 mm downstream from the injector tip, along the chamber’s central axis. Measurements were taken in the centre of plume 2, which was the central plume nearest the camera as the spray was seen from the side view in Figure 2(b). Droplet size measurements were taken over 200 injections for all conditions. The total number of samples collected for each injection event was of the order of 25 samples per millisecond for each of the measured injections, resulting in more than 7500 valid droplet size and velocity measurements over the duration of the injection event for each test condition. The effect of temperature dependence of the refractive index of the fuels was also considered on the basis of previously published work in this area, *e.g.* [7, 8].

Test Conditions

To mimic the conditions of the injector body when mounted in an engine, the injector body mounting was heated to temperatures of 20, 50, 80, 100 and 120 °C and its temperature was monitored by a thermocouple embedded into the housing of the injector. A settling time of at least 30 min and an injection frequency of less than 1 Hz was used to allow the fuel inside the injector to reach similar temperatures to that of the surrounding injector body. Therefore, in this paper, the term “fuel temperature” corresponds to the temperature reading of this thermocouple, *i.e.*, the temperature of the injector as close to the injector tip as possible, not the temperature of the fuel itself [5, 6].

In-cylinder engine gas pressures at time of injection were replicated by inducing a vacuum (0.5 bar absolute for early injection strategies at part-load ‘homogeneous’ charge operation). Tests were also carried out at atmospheric pressure (nominally 1.0 bar). Previous tests have shown little deviation in the global spray form

for different multi-hole injectors at elevated gas pressures [5, 9], except the expected decrease in plume tip penetration due to increased drag, therefore, the tests presented here were carried out at ambient and reduced gas pressures, where the greatest differences in spray development were observed.

Injection timing and duration was held constant at 1.5 ms pulse-width for all conditions, with the camera recording for 2.5 ms each injection to capture initial, steady-state, end of injection and post injection spray characteristics. Due to obscuration of the spray tips by the pressure chamber windows, it was only possible to measure spray tip penetrations until approximately 1.0 ms After the Start Of Injection (ASOI, *i.e.* the time period following the rising edge of the trigger pulse sent to the injector driver unit). The average driver delay from the start of the injection pulse (Start of Injection, SOI) to fuel seen at the injector tip was of the order 300 μ s, as has been quantified in detail for a similar system in [5].

Single-Cylinder Optical Engine

Engine Installation

A single-cylinder optical engine was used to investigate the spray behaviour under realistic operating conditions. The flexible optical access allowed imaging through a piston crown window (swirl plane) and through a full-stroke quartz liner (tumble plane). Table 2 summarises details of the engine set-up; valve timings are given with respect to intake Top Dead Centre (TDC) as °CA After TDC (ATDC). More details about the engine can be found in [3, 4]. The engine was motored at part load or full load (0.5 bar or 1.0 intake pressure, respectively) and fired at part load continuously, *i.e.* with no skip-firing. A piezo-resistive absolute pressure transducer was installed in the inlet plenum to set the engine load by adjusting the throttle. An AFR meter was used in the exhaust to measure oxygen content and set the correct fuelling. The engine was run with stoichiometric AFR for all the tests presented in this paper. 100 consecutive cycles of in-cylinder pressure data were acquired after the engine was fired and had been allowed to stabilise for 20–30 s. The influence of fuel temperature was observed by using engine head coolant temperature of 20 °C and 90 °C. It should be noted that the engine coolant temperature is not an exact measure of fuel temperature but is relevant for practical comparisons of real-world engine operating conditions and for direct comparison with the injector heating arrangement in the quiescent chamber. Enough heat-soak time (30–40 min) was allowed at 90 °C so that the engine head components acquired enough thermal inertia to simulate fully warmed-up conditions. Synchronization of various control triggers for ignition, injection and camera/laser systems was achieved using an AVL engine timing unit and an optical encoder with 1800 pulses per revolution on the crankshaft. The in-cylinder pressure was also logged; then processing was carried out to calculate combustion statistics for each cycle using LABVIEW and MATLAB based software.

Injection Strategies

The injection timing was set for homogeneous mixture preparation mode, *i.e.* early in the intake stroke, with Start of Injection (SOI) set to 80° CA ATDC. This was done after initial tests and on the basis of recent results from injection sweeps and wall wetting studies in the same engine with a similar injection system [3, 4]. The different injection pulse-widths needed for the two fuels, with E85 requiring a longer pulse length

due to its lower AFR ratio at stoichiometry (14.7 for gasoline, 9.0 for pure ethanol), also meant that in order to compare the evaporation histories of the two fuels, tests were first carried out under motoring conditions with 0.5 bar intake pressure using the same injection pulse widths (0.8 ms). By monitoring the scattering intensity in the images after the end of injection it was possible to gain information about the evaporation process. The effects of realistic pulse widths on liquid wall-wetting and evaporation at the same load were also investigated (1.25 ms for gasoline, 1.6 ms for E85) and spray areas were calculated to confirm whether evaporation rates changed due to potential vapour saturation effects.

Image Acquisition

Spray imaging was performed on two orthogonal planes in the engine. A vertical plane, termed ‘tumble plane’ in Figure 2(a), captured the spray as it penetrated the cylinder, allowing comparisons to be made with the chamber’s ‘end’ view spray development in Figure 2(b). A horizontal plane, termed ‘swirl’ plane in Figure 2(a), imaged the projected ‘footprint’ of all the spray plumes and flame images, viewed through the piston crown. Flood lighting illumination with a high-repetition rate Nd:YLF laser was achieved using a series of divergent lenses. The same high-speed camera used in the pressure chamber was also employed for the engine study at a frame rate of 9 kHz (swirl plane) and 5 kHz (tumble plane), corresponding to 1° CA and 1.8° CA between frames at 1500 RPM. This was possible with an image resolution of 640×480 pixels and 512×1024 pixels, respectively. The lower temporal resolution in the tumble plane was necessary to increase the spatial resolution in order to capture the full spray development and post-injection mixing stage from injection timing to close to Bottom Dead Centre (BDC). At this resolution the on-board camera memory allowed just over 100 cycles to be imaged consecutively before data-download to the hard disk was necessary.

Flame images were acquired at a frame rate of 9 kHz in the swirl plane, *i.e.* at 1° CA separation between frames at 1500 RPM. The combustion process was visualised by imaging the natural light flame chemiluminescence on a cycle-by-cycle basis for 100 consecutive cycles. Simultaneous acquisition of in-cylinder pressure data allowed mass fraction data to be directly compared to the imaging data on a cycle-by-cycle basis in order to extract information about specific flame growth characteristics, like the evolution of an ‘equivalent’ flame radius and the flame’s centroid trajectory.

RESULTS AND DISCUSSION

Spray Development in Quiescent Environment

Figure 3 shows a direct comparison between the spray development of gasoline and E85 at conditions of 1.0 bar and 20 °C, as imaged from the ‘side’ and ‘end’ views from first fuel seen out of the injector at 333 μs ASOI up to the end of injection at 1888 μs ASOI. Figures 4 and 5 illustrate spray images of gasoline and E85, respectively, at fixed exposure timing, namely 777 μs ASOI, for pressures 0.5 bar and 1.0 bar, with the injector temperature increasing gradually from 20 °C to 120 °C. The sprays are shown for both side and end views of imaging. *Iso*-octane was only imaged through the side view as it did not form an integral part of the

work, but was included as a model single-component fuel with fixed boiling point (99 °C at 1.0 bar) for comparison with the multi-components fuels (Figure 6).

Although difficult to present in the hardcopy format of Figure 3, the analysis of high-speed video imaging revealed significant transient characteristics in the injection process. Over an injection duration of 1.5 ms (corresponding to 13.5 °CA at 1500 RPM), the injection event could be described by two or three different stages. First, spray plume formation straight out of the nozzle when the flow field is still forming inside the nozzle and gradually stabilises (see first 3 rows of images in Figure 3, *i.e.* 333–555 μ s ASOI). Second, a pseudo steady-state stage, where the spray is stable with defined cone angles, but noticeable periodicity within the inner core spray (typically 666–888 μ s ASOI). In the final stage the spray undergoes significant distortion starting from the near-nozzle region, setting up large scale oscillations which put the spray into crossflow (999–1888 μ s ASOI). The effects can be seen in the images (*e.g.* at 1222 μ s ASOI) as ‘curved’ plumes close to the nozzle and areas of darker spray which may indicate either regions of droplet clustering or ligaments. The effect was repeatable from shot-to-shot and was seen with both E85 and gasoline at all operating conditions.

Effects of Fuel Properties and Conditions

Observation of the injection process in Figure 3 further revealed that under certain conditions – most noticeable at 20 °C fuel temperature and 1.0 bar or 0.5 bar gas pressure – there were some fundamental differences in the way E85 and gasoline sprays disintegrated outside the nozzle. The macroscopic characteristics were clearer from the ‘end’ view which allowed the six individual plumes to be seen (in contrast to the side view which shows three plume pairs). The first frame where fuel was imaged showed that the gasoline spray was substantially more developed than the E85 spray, with nearly twice its penetration (as quantified in the penetration graphs shown later). Despite this, two frames later the penetrations of both fuels are very similar and at 555 μ s ASOI in the end view (or after 777 μ s ASOI in the side view), the penetration of E85 has overtaken that of gasoline. At higher fuel temperatures this trend nearly disappears (Figure 4 and 5); the same effect was seen in the spray images from the optical engine as will be discussed in the relevant section later. This clear trend has also been noticed in other recent measurements by some of the current authors using pure alcohol fuels (ethanol and butanol) with a similar injector of different nozzle-hole geometry [10]. Interestingly, analysis of spray images under similar operating conditions from a different study [11] with E85 and an 8-hole injector appear to show similar trends of smaller early penetration for E85 compared to indolene, although this detail was not mentioned by the authors of that study. The finding therefore appears to suggest that this is certainly not a hardware-related issue but one that originates from differences in the liquid transport properties of alcohol fuels compared to the hydrocarbons typically found in gasoline. For example, E85 has a viscosity close to that of pure ethanol, which at 20 °C is more than double, on average, when compared to the main gasoline hydrocarbon components such as pentanes, octanes and xylenes (fuel properties from Yaws, 2003); the viscosity decreases rapidly with increasing temperature however, and at 80–100 °C the difference is only ~40% higher for pure ethanol. The much lower absolute values of viscosity at high temperatures are predicted to reduce the impact of this fluid property on initial

spray development and the sprays can be seen to behave more similarly straight out of the nozzle than at 20 °C. Such strong trends support hypotheses involving fluid properties in explaining this subtle, yet consistent, difference in the initial stage of spray formation immediately after needle lift.

For all fuels, the spray geometry is clearly affected by changes in fuel temperature, especially at the low gas pressure of 0.5 bar. The effect of high temperature is seen to change the spray pattern from clearly identifiable and directed spray plume pairs (side view) to a gradual merging of the three plume pairs at the highest injector temperature, eventually producing a 'single' plume along the injector axis as the outer plumes are drawn together. This is the typical 'collapsed' spray form which occurs due to 'flash-boiling'; as the fuel exits the nozzle it is rapidly depressurised to a pressure sufficiently below its saturated vapour pressure and the vapour generated is drawn towards the central low pressure regions, overcoming the radial momentum of the injected droplets. However, it is clear that pure 'flashing' does not occur and other mechanisms are involved with respect to transient heat and mass transfer phenomena. Results also showed that with *iso*-octane there was no such collapse at these conditions and the three plume pairs remain clearly defined even at 120 °C. As the ambient pressure was increased from 0.5 bar to 1.0 bar, this effect gradually disappeared due to the increase in the liquid fuel boiling point and the spray plumes regained their original directionality.

The geometry of the E85 sprays was clearly less sensitive to temperature than gasoline; although E85 was more sensitive than *iso*-octane. These characteristics were particularly noticeable in the range 20–80 °C and for the end view which shows the six individual spray plumes (see Figures 3, 4 and 5). It is clear that at 20 °C the plumes for E85 were noticeably thinner and more defined than those of gasoline and it can be said that the break-up 'efficiency' was clearly much better for gasoline at these low temperatures; it is expected that an 'intact' core would be longer for E85 because evidence of large scale plume distortion was more difficult to find than for gasoline. These differences were maintained as the liquid temperature was increased to 50 °C; at 80 °C however, the E85 spray plumes lost some of their previous definition and penetrated less with the overall spray appearance resembling that of gasoline. At this temperature and 1.0 bar, the E85 spray resembles gasoline at 20 °C at 0.5 bar or to some extent 50 °C at 1.0 bar, exhibiting better spray break-up and presumably smaller liquid core lengths (this can be seen from the absence of a dark 'spine'). This 80 °C is a critical temperature for E85 because of its distillation characteristics which tend towards a near-single boiling point around 78 °C; see Figure 1. It is also particularly clear from the spray development images in Figure 3 that E85 sprays are 'pointy' from the outset and that they retain this feature for the full duration of the injection event.

The differences in spray shape between different fuels and conditions were quantified from the calculation of the 'umbrella' spray cone angle, *i.e.* the overall external cone angle as measured between 2 and 22 mm from the injector tip. This demonstrates the extent of convergence of all spray plumes, as shown in Figures 7 and 8. At 1.0 bar gas pressure the cone angle initially increases with temperature due to better atomisation which 'swells' the spray plumes. For gasoline the effect of collapse can be seen from 100 °C onwards in contrast to

E85 where this characteristic is seen only from 120 °C fuel temperature. At 0.5 bar gas pressure the same trend is seen but the decrease in cone angle happens even earlier, at 80 °C for gasoline and between 100 °C and 120 °C for E85. The single component *iso*-octane shows a more consistent transitional behaviour at both 0.5 bar and 1.0 bar gas pressures as the boiling temperature is approached.

Break-up Regimes and Non-Dimensional Numbers

The atomisation process can be described by the liquid Weber number $We_L = \rho_L u^2 d / \sigma$, the Reynolds number $Re = \rho_L u d / \mu_L$ and the Ohnesorge number $Oh = We_L^{0.5} / Re$ which incorporates all the main fluid properties. Calculation of the Weber and Ohnesorge numbers for a multi-component fuel is not trivial, particularly at varied temperature/pressure conditions necessary for constructive analysis. Therefore, it was decided to select specific single components to aid the discussion and provide some interpretation of the results. The reference single-component fuels were chosen to be *n*-pentane, *iso*-octane and *o*-xylene (Figure 1). Additionally, pure ethanol was considered for comparison. The non-dimensional parameters were calculated for these single components using a nozzle diameter of 0.3 mm and a nominal jet velocity through the nozzle of 60 m/s estimated from the images. All fuel property data were obtained from [12]. Analysis of the non-dimensional numbers for ethanol and the equivalent light, medium and heavy gasoline fractions revealed that there were differences in the parameter space that could explain the different behaviour of E85.

Figures 9 and 10 present the calculated Re and We_L numbers, respectively, for the single components selected and for temperatures from 280 K (*i.e.* 7 °C) up to 458 K (*i.e.* 185 °C). The Ohnesorge [13] diagram is shown in Figure 11 for the same components, with the Reitz and Bracco [14] break-up regimes superimposed. At 20 °C (293 K) ethanol had $Oh \sim 1.6 \times 10^{-2}$, but gasoline's components (*iso*-octane, *n*-pentane and *o*-xylene) had Oh numbers lower than about half that of ethanol. At higher temperatures, Oh converged to similar values for all fuels, typically $\sim 0.5 \times 10^{-2}$ at 120 °C (393 K). Together, these figures confirmed that with the fuels and experimental conditions used in this study, the jet break-up regime is generally in the atomisation regime. The values calculated for the coldest temperature (7 °C, 280 K) though are close to the boundaries of the second wind induced regime for most fuels. For ethanol, in particular, at cold conditions, the calculated values have crossed into the second wind induced regime and this is quite interesting in its own right considering the implications for engine cold-start conditions. In Figure 10, the Weber numbers for ethanol were found to be $We_L = 35,000$ at 20 °C and $We_L = 50,000$ at 80 °C, compared to *iso*-octane's $We_L \sim 40,000$ and $We_L \sim 55,000$ at the respective temperatures. However, all the values fell within the same atomisation regime, typically defined as $We_L > 30,000$ by Sallam *et al.* [15].

Transition criteria based on Re or We_L numbers are not easy to find in the literature for similar jet and operating conditions, particularly with real fuels. Some experiments with similar magnitudes of Re and We_L to those of the current study were performed by Suh and Lee [16] on two-dimensional nozzles with length to width ratios (L/W) of 1.8 and 2.7 using Diesel fuel but critically under turbulent to cavitating transition conditions. In the current study from the external dimensions of the injector and other studies using similar hardware, *e.g.* [17, 18], L/W was estimated to be in the range 2.0–2.5. Transition criteria were established in

[16] for turbulent to cavitating conditions defined by $Re > 17,000$ – $19,000$ and $We_L > 35,000$ – $39,000$ and this is the only such study that was found where transitions from turbulent to cavitating flows were a) imaged, b) comparable in macroscopic appearance to those in the current study and c) matched Re and We_L in regions where the current authors are also reporting clear macroscopic differences in the break-up mechanism, albeit between two fuels at ‘exact’ initial states. The results suggest that it may be possible to relate the thin/compact spray features observed for E85 at 20 °C to the lower Re of E85 relative to gasoline’s components at 20 °C. This is predicted to fall below the $Re \sim 19,000$ transition point for cavitation reported in [16]. Indeed, ethanol at 20 °C has a $Re \sim 13,000$, compared to $Re \sim 25,000$ for *iso*-octane in Figure 9, placing *iso*-octane clearly in the cavitating regime, based on Re . A similar trend was seen for We_L (Figure 10).

The importance of aerodynamic effects on primary break-up has been a topic of debate. Whereas initial work by Miesse [19] implied that the gas phase Weber number $We_G = \rho_G u^2 d / \sigma$ could be used as a criterion for delimiting the second wind-induced to atomisation regimes ($We_G > 40$), the consensus at this time is that they are not the dominant mechanism in primary atomisation, *e.g.* see [20]. Aerodynamic interactions can contribute to the droplet formation at the liquid interface (via droplet pinching resulting from high energy small eddies) and droplet transport away from the interface (via large-scale recirculating vortices), particularly for $\rho_L / \rho_G < 500$ [21], *i.e.* typically applicable to gas pressures in excess of 1.5 bar for most fuels. The values of We_G at 0.5 bar for 20–80 °C are in the range 25–40 for ethanol and 40–50 for gasoline’s medium fraction, *iso*-octane, whilst at 1.0 bar We_G is in the range 50–75 for ethanol and 60–90 for *iso*-octane. Using the definition of Faeth *et al.* [22] for the secondary droplet break-up regimes which were calculated on the basis of We_G , transitions in the relevant regions occur at $We_G = 13$ (bag break-up), $We_G = 35$ (multimode break-up) and $We_G = 80$ (shear break-up). The differences in We_G between the fuels at 0.5 and 1.0 bar are perhaps manifested by the observed differences in the spray images when considering the limits of $We_G = 35$ and $We_G = 80$.

Spray Penetration and Break-up Length

For quantitative comparison between fuels and conditions, the spray-tip penetration was measured from acquired images by automated image processing. The results are shown in Figures 12–17 for all the test fuels at each of the temperature conditions for 0.5 and 1.0 bar gas pressure. The plume tip penetration is presented for the left plume pair (plumes 1, 6) due to the differing absolute lengths of the plume pairs. However, all plume pairs illustrated the same penetration trends in relation to the conditions.

For gasoline at 1.0 bar in Figure 13, *i.e.* representative of high-load conditions (wide-open throttle), a reduction in spray tip penetration was observed with increased temperature. The effect of temperature on spray tip penetration was much smaller at 0.5 bar ambient pressure (Figure 12) showing a lower penetration range between 20 °C and 120 °C than at 1.0 bar gas pressure. Spray plume penetrations were larger at high temperatures and low gas pressures (condition of spray collapse) due to the contraction of the spray plumes in the axis of the injector, which increased the spray downward momentum relative to the non-collapsed condition. For E85 at 1.0 bar gas pressure (Figure 15), the range of penetration was similar to gasoline for

both gas pressures; spray-tip penetrations however were larger in magnitude for E85, although at 0.5 bar and 120 °C they were very similar as a result of both sprays experiencing comparable levels of ‘spray collapse’ (Figure 14). There was also a more gradual distribution in penetration values for E85 with respect to temperature than for gasoline, probably as a result of its bias towards a single chemical component (ethanol) thereby exhibiting a more gradual response to fuel temperature as it approached its boiling point. Figures 16 and 17 show the spray-tip penetration calculated for *iso*-octane. From these, it can be immediately observed that the difference between the extremes of the presented curves is much narrower than for gasoline and E85.

No major attempt was made to calculate break-up lengths from the images of the current study. One of the reasons is that a recent study with a laser induced fluorescence technique that illuminated the liquid jet core through the jet nozzle showed that the liquid core lengths were significantly shorter than any ‘surface length’ imaged by shadowgraphs or Mie-scattering, typically by 30% or more [23]. Nevertheless, an effort was made to link correlations from the literature to observations of the current study. The idea of an ‘intact liquid core’ for engine sprays was initially applied in [24] to characterise the inner structure of Diesel sprays; most subsequent work on engine combustion systems also focused on Diesel injectors. Although some reports have tried to relate features of the spray penetration curves to break-up lengths, usually by analysing the change in gradient seen when aerodynamic drag begins to affect the penetration velocity significantly [25], no major reports with empirical correlations were found that could be applied to DISI systems, especially obtained with fuel as the working fluid instead of water. However, Sallam *et al.* [15] has shown that for turbulent flow at the jet exit for water and ethanol with properties of turbulent primary break-up as discussed above, the non-dimensional break up length L_c/d first increased with We_L and then became independent of We_L at large values at conditions specifically governed by liquid-gas density ratio ρ_L/ρ_G as follows:

$$L_c/d = C_b (\rho_L/\rho_G)^{1/2} \quad (1)$$

where C_b is an empirical parameter that describes the nozzle flow conditions and other effects. In the literature C_b takes values typically in the range 3.3–15 [26]. In particular, Sallam *et al.* [15] identified a value of $C_b=11.0$ for shear-type break-up of a liquid-ethanol jet in stagnant air, as derived from a best-fit correlation and showed break-up lengths of L_c/d in the range 250–300 for the We_L values of interest in the present study. This results in break-up lengths of ~75–90 mm for the present study which are clearly not realistic. Most probably this is due to the presence of cavitation in the nozzle under study, as estimated from experiments with a similar injector in [5, 17, 18]. For pure substances with known fluid properties, the static pressure relative to the vapour pressure within the nozzle hole will define the inception of cavitation. However, for multi-component fuels, the combination of different chemical components and vapour pressures requires another measure to be used which accounts for this complexity, the ‘bubble point’ pressure. The bubble point is indicative of conditions under which lower boiling components *i.e.* those with high vapour pressures, begin to evaporate or ‘bubble’ within a multi-component liquid, thus inducing cavitation. At 20 °C gasoline has a typical bubble pressure of ~0.3 bar [6]. This is much larger compared to ethanol’s or *iso*-octane’s vapour pressure of ~0.1 bar at the same temperature. At 90 °C the difference between gasoline’s bubble point and ethanol’s vapour pressure is lower (percentage-wise) than at 20 °C,

namely 2.4 bar bubble pressure for gasoline and 2.0 bar vapour pressure for ethanol, whilst *iso*-octane's vapour pressure is 0.8 bar. These trends are also reflected in the observed spray images (see side views) where E85 behaves much more like gasoline at high temperatures but it is closer to *iso*-octane at low temperatures (Figures 4–6).

Droplet Sizes

In order to investigate further the link between the spray development and atomisation, droplet sizing was carried out in the chamber for a range of gas pressures and injector temperatures. Figures 18 and 19 show the Sauter Mean Diameter (SMD, $D_{3,2}$) measured 25 mm below the injector tip for plume 2 using gasoline and E85. Droplet sizes were found to decrease with both a reduction in gas pressure (for a given injector temperature) and an increase in injector temperature (for a given gas pressure), ranging from ~22 to 8 μm for gasoline and ~27 to 10 μm for E85. E85 always showed larger droplets than gasoline by ~25–30% with the highest differences found at low temperatures. This may well be due to a smaller degree of cavitation from ethanol's lower vapour pressure, as well as from the higher surface tension and viscosity of ethanol. A different mechanism involved may stem from E85's higher latent heat of vaporisation, cooling locally the gas temperature and reducing the droplet evaporation rates. At higher temperatures, particularly with high levels of superheat for both fuels, the differences became smaller, namely ~15–25% higher droplet sizes were measured for E85. Analysis of We_G indicated that secondary atomisation of E85 fell into a different droplet break-up regime compared to gasoline which predicts larger droplet size distributions. The higher surface tension of ethanol at 20 °C appears to explain why the differences existed mainly at cold conditions since at 90 °C the surface tensions of E85 and gasoline are comparable. In the range 50–120 °C, droplets were larger at 1.0 bar by ~10% than at 0.5 bar for both fuels. At 20 °C, however, the droplets were measured to be smaller at 1.0 bar than at 0.5 bar and this may be associated with the particular nature of the spray development and break-up, *e.g.* very dense spray and presence of large non-spherical ligaments, thus only the smaller spherical droplets were picked up and validated by the instrument.

The values of droplet sizes measured in the current work are comparable to values that have been recently reported from different multi-hole injectors at similar operating conditions by other authors. In particular, Marriott *et al.* [11] reported droplet sizes with gasoline and E85 in the range 10–25 μm using an 8-hole injector with injection pressures 110 bar and 200 bar, fuel temperatures 20 °C and 90 °C, as well as gas pressures 0.6 to 2.25 bar. They too found larger droplet sizes for E85. Mitroglou *et al.* [9] measured values for *iso*-octane in the range 10–20 μm during the injection event with 120 and 200 bar injection pressure at 1.0 bar backpressure and ambient temperature; higher values of ~40 μm and 25 μm were measured at the start and end of injection phases though. They also predicted that their injector was cavitating and suggested that these phenomena were mostly responsible for the disintegration of the liquid jet. It should be added that the values reported in the current study are averages over the full injection event and therefore include any transient effects occurring inside the nozzle during this period which affect the spray formation and break-up process. Thus it is encouraging that values were found to be so low on average. On the other hand the combined effect with flash-boiling at 100 °C was shown to result in nearly a 30–40% reduction in the SMD,

with gasoline experiencing the largest droplet size reduction as a result of the low boiling point fractions such as pentanes. These effects will be elaborated further in the next section when spray areas are quantified in the engine over the injection and post injection period.

For completeness, it may be pointed out here that there are some limited empirical correlations in the literature for droplet sizes from multi-hole injectors, incorporating Re and We_L numbers. Laser diffraction droplet sizing on Diesel sprays by Hiroyasu and co-workers [27, 28] gives the following relation for the SMD, based on experiments with 900 bar injection pressure and 30 bar gas pressure:

$$\frac{SMD}{d} = 0.38 Re^{0.25} We_L^{-0.32} \left(\frac{\mu_L}{\mu_G} \right)^{0.37} \left(\frac{\rho_L}{\rho_G} \right)^{-0.47} \quad (3)$$

Values of droplets calculated from this relation for all fuel components and conditions of the current study have shown that these follow the correct trend with temperature and pressure, as well as with fuel type (predicting slightly larger ethanol droplets), but the absolute values are ~50% smaller than those measured in the current study. This means that the correlation perhaps can be applied to DISI mixture formation studies, but certain tuning of the constants would be necessary.

In-Cylinder Effects on Spray Formation and Break-up

The engine sprays were initially imaged under ‘static’ engine conditions to confirm the findings found in the quiescent chamber. Then two injection durations were used under motoring conditions. The smallest (0.8 ms) was employed to keep engine surfaces relatively dry and improve image quality for continuous testing, as well as to allow comparison between the fuels at nominally identical conditions. The second duration was the correct one for stoichiometric firing at 1500 RPM, namely 1.25 ms for gasoline and 1.6 ms for E85 at part load (0.5 bar intake pressure), 2.5 ms for gasoline and 3.5 ms for E85 at full load (1.0 bar intake pressure).

Intake Pressure and Fuel Temperature Effects

Figure 20 shows spray development images on the tumble plane at part load, whilst Figure 21 presents spray development images on the swirl plane at the same condition. On the tumble plane, clear interactions between the spray and the in-cylinder flow can be observed. Spray motion is also affected by direct interactions with cylinder/piston walls as the sprays penetrate into the combustion chamber, ‘pushing’ the spray towards the centre of the cylinder and establishing recirculation zones or vortices close to the piston surface. Spray-valve interactions are also clear as a result of the particular valve-timing used on this engine and act to concentrate some of the spray momentum in a downward direction *i.e.* towards the piston crown. The overall effect of in-cylinder flow on spray development and mixing in Figure 20 can be compared directly to the ‘static’ sprays shown earlier. It is clear from the outset that the spray plumes cannot be picked individually due to strong interactions with the intake flow. These interactions lead to enhanced spray break-up and dispersion compared to the static condition. The flow also improves evaporation and ‘entrains’ the spray within the flow thus keeping it away from direct impingement onto cylinder surfaces. Full-load operation (1.0 bar intake pressure) resulted in shorter penetration, different levels of impingement on the piston crown and improved spray break-up due to stronger in-cylinder flow. At high engine temperature (90

°C) the gasoline spray is seen to ‘collapse’, as also observed in the pressure chamber. There is a partial loss of directionality for all plumes, which are drawn together resulting in a vertical spray momentum ‘chasing’ the downward moving piston. However, the differences between the spray collapse of E85 and gasoline on the tumble plane are small and less obvious than those in the quiescent chamber for equivalent views.

The E85 spray on the swirl plane was observed to collapse to a lesser extent than gasoline. This is visible immediately in the first frame in Figure 21, when fuel is first seen exiting the nozzle and the gasoline plumes are already expanded significantly, in contrast to E85, where the individual plumes are still clearly identifiable. Following this period, the footprint of the spray at the end of injection seems less concentrated below the injector for E85 than for gasoline; the results are thus consistent with observations from the pressure chamber. For 90 °C, as in the tumble views, post-injection evaporation was improved, as observed by quickly reducing light scattering in the spray movies. The spray/valve interactions for the current injection timing and valve phasing resulted in large droplets/ligaments seen shredding off the left intake valve after the end of injection. Tests with the actual stoichiometric firing pulse widths showed that the longer duration necessary for E85 resulted in significantly increased interactions with the intake valves, which had the effect of pushing fuel up the intake manifold as the spray jet ‘ski ramped’ off the back of the valves. This effect was again reduced at high engine coolant temperatures as the spray plumes collapsed towards the centre of the combustion chamber therefore avoiding the intake valve to some extent. These interactions resulted in a clear bias of the initial liquid fuel distribution towards the exhaust side of the combustion chamber which should be kept in mind when analysing the flame images in later sections.

Spray Area Quantification

In order to compare the evaporation process of gasoline and E85 in the engine, spray areas were calculated from the swirl plane images and averaged for 100 cycles, to quantify the projected spray footprint and demonstrate the temporal behaviour of liquid fuel in the cylinder *e.g.* the ‘tails’ after the end of injection indicate rate of fuel evaporation. This was done using the same injection pulsewidth initially (0.8 ms) in order to directly compare the two fuels and then using their correct pulsewidths for stoichiometric firing operation (1.25 ms for gasoline and 1.6 ms for E85). Calculated average spray areas are plotted in Figures 22 and 23. These have been normalised to the piston crown area such that 0.7 on the graph y-axis refers to 70% of the piston crown window area; the ratio of visible window area to the full cylinder bore area is ~0.55, with the window diameter corresponding to 0.73 of full bore diameter, *i.e.* 70% of the piston-crown window area corresponds to ~40% of the engine full bore area. Spray images have been superimposed at specific crank-angle degrees after the start of injection to aid interpretation. These show representative sprays at cold and hot conditions for both fuels at different stages in the injection and post-injection periods. The End of injection (EOI) point is also shown for each case in Figures 22 and 23 (by vertical lines) to put spray timings into context. The RMS of the spray areas was similar for both fuels, with values of less than 5% during injection and of the order 20–25% after the end of injection when the collected Mie scattering was tailing off.

The average spray area for E85 is smaller than for gasoline at 20 °C engine-coolant temperature from the start of injection, reflecting the earlier start of fuel seen for gasoline (which begins to contribute to the area earlier) and the fact that the individual plumes of E85 are more compact and break up slower than those of gasoline. This is also clear from Figure 21 where gasoline is seen earlier at the injector tip and is therefore more developed by 5 °CA ASOI. The rate of rise in spray areas is very similar for both fuels during the injection period for 20 °C coolant temperature, highlighting the small differences in individual plume widths or cone angles. After the end of injection the E85 spray areas quickly decrease, reaching a transition point at 20° CA ASOI where there is a slowdown in the light scattering level detected. This level then tapers off gradually until the background noise levels are reached after 55° CA ASOI. By the end of the injection period gasoline has a higher peak spray area reflecting the more efficient spray break-up which widens the plumes. Immediately after the end of injection the rates of evaporation are comparable but gasoline then also makes a transition to a much lower rate of evaporation at ~22° CA ASOI. The intensity levels of light scattering as the organised spray structure breaks up and droplets become entrained in the engine flow remains higher than for E85 however. This is an interesting observation that can have two explanations. The first relates to the proportion of 'heavy' components in gasoline with high boiling points which will reduce the droplet evaporation rates for gasoline as the light fractions evaporate preferentially from the droplet surface. These smaller droplets that remain will take a longer time to evaporate at low temperatures and will therefore continue to participate in the Mie-scattering process. The second reason relates to spatial variations in the droplet field resulting from the injection event; this will be affected by the geometric break-up process of the spray which for E85 was observed to be less efficient, resulting in higher levels of impingement on the piston crown. If more of the fuel volume injected has not been atomised because of surface impingement, then the final droplet field will be less populated resulting in lower Mie-scattering intensities. The differences seen between E85 and gasoline are likely to be a combination of both these factors.

At 90 °C coolant temperature the change in the macroscopic spray structure as a result of 'spray collapse' dictates that it is not meaningful to compare absolute levels of measured spray areas to the 20 °C condition, during the injection period. Once the spray structure has broken up and only the droplet field remains, then comparisons can be made but the effects of spray collapse on the 3-D droplet field should be carefully assessed before general conclusions are made. For example, it has been shown that the effect of spray collapse is to concentrate the six spray plumes below the central axis of the injector *i.e.* in the centre of the piston window. Any change in the level of spray collapse will thus affect the 'projected' spray area or footprint, since this is measured as a 2-D parameter. This explains why spray areas remain higher for longer for E85 compared to gasoline, since E85 spray collapses to a lesser extent and therefore maintains more of a radial component, which fills the piston window even after the end of injection. The spray areas for gasoline decrease rapidly after the end of injection at a constant rate before slowing down at around 30° CA ASOI and approaching the background level close to zero. E85 shows similar rates of evaporation to gasoline at this temperature, which may be explained by the similar 'mean' boiling points of the two fuels.

When the injection pulsewidths are adjusted to reflect their true stoichiometric values, the longer injection duration needed for E85 stoichiometric operation (1.6 ms compared to 1.25 ms for gasoline) naturally results in larger peak spray areas. The initial spray area at 20 °C can again be seen to be smaller for E85 due to the longer delay in seeing fuel at the injector tip as mentioned earlier; however, by $\sim 7.5^\circ$ CA ASOI they are very close to the gasoline line indicating a marginally faster rate of penetration for E85, a characteristic which was also seen in the quiescent chamber. They remain very similar until the end of the injection of gasoline, the thinner plumes of E85 making up the loss in area with their higher plume penetrations. The tail of the area curve for gasoline is similar to begin with when compared to the shorter injection pulse in Figure 22, suggesting similar rates of evaporation which would be expected; however, the tail tapers off much more gradually which potentially indicates local vapour saturation reducing the evaporation rates. A similar trend is shown for E85. At 90 °C the gasoline spray collapses and concentrates the spray in the centre of the piston window thus reducing the peak areas calculated. However, the higher temperatures are reflected in the much steeper tail of the curve indicating more rapid evaporation. The longer length of the injection event for E85 means that both at 20 and 90 °C the window is completely filled with liquid spray. The evaporation rates are again comparable to those of gasoline and more gradual relative to the shorter injection event, potentially indicating again vapour saturation, or simply being the result of a more populated droplet field. The difference between E85 and gasoline is relatively small at 55° CA ASOI for cold conditions, $\sim 7\%$, but it is larger for fully warm conditions, $\sim 13\%$ higher for E85. This compares quite well with the increase in the fuel pulsewidth used for E85; the discrepancy between cold and hot conditions could be attributed to fuel impingement, given that some fuel is expected to have impinged on the engine surfaces at 20 °C and thus would not contribute to the spray area calculation.

Flame Growth and Combustion Pressure

In order to study the observed spray effects with respect to combustion for the two fuels, an imaging study of in-cylinder flame development was carried out for a batch of 100 consecutive cycles. The engine was run at 1500 RPM with 0.5 bar intake pressure. All firing tests were carried out at the same ignition timing (35 °CA before TDC) in order to compare flame growth under the ‘same’ average flowfield and nominal conditions of pressure and temperature. Such observations must be put in context since the ignition timing could be advanced to improve the performance of E85 by taking advantage of its higher octane number, *i.e.* anti-knock potential, and benefit from ethanol’s higher laminar flame speed at stoichiometry [29–32].

Time-resolved images of flame growth are shown in Figure 24 for both E85 and gasoline and for several crank angle degrees After Ignition Timing (°CA AIT) with engine coolant temperature 20 °C and 90 °C (head and liner). The flames are presented with optimised contrast levels such that they are more easily seen on printed paper. However, they were enhanced by the same amounts to allow comparisons of luminosity across fuels for each crank angle set shown. Some clear differences can be seen in terms of presence of bright spots for gasoline flames, highlighting potential droplet burning in diffusion mode. There are also noticeable differences in the actual texture and average luminosity of the flames for E85 compared to gasoline for which no single cause can be attributed. E85 shows a more opaque appearance, with less

contrast within the flame and less evident ‘spotty’ burning. Certainly no reports have been found in the literature that describe visual characteristics of natural light chemiluminescence of ethanol-based versus gasoline turbulent premixed flames in other optical engines or combustion ‘bombs’ and therefore it is difficult to propose a strong hypothesis for these observations. Other parallel studies by some of the current authors with different injector geometries [10, 33] have shown that similar visual characteristics were seen for pure ethanol and other higher C-chain alcohols such as butanol. Current work in the same engine, as yet unpublished, has also revealed that methane flames have similar attributes in the same engine under the same operating conditions. The link to carbon chain length is one possible mechanism as is the type and levels of particulate matter generated during combustion. Further study of the visual appearance of the flames is needed, including spectroscopy and Laser-sheet imaging to look into the local flame structure.

Combustion performance was very similar for gasoline and E85 at hot engine conditions as shown by the average in-cylinder pressure traces and mass fraction burned in Figure 25 and 26, respectively. Slower combustion development for E85, especially at 20 °C, indicated the potential effect of increased charge cooling and the necessity of advancing ignition timing to match gasoline performance. Specifically, the pressure traces of Figure 25 revealed that at ignition timing the average in-cylinder pressure for E85 was lower by ~0.2 bar, most probably as a result of lower in-cylinder temperatures from ethanol’s greater cooling capacity. However, after ignition the pressure rise was faster than that of gasoline. The phasing of combustion defined by the location of peak pressure was very similar but the peak pressure was lower by ~2 bar for E85. The poor atomisation and higher levels of fuel impingement observed visually with E85 may have also contributed to a leaner mixture at the spark plug and thus to a slower flame growth. At 90 °C both fuels produced very similar pressure and MFB curves with comparable levels of cycle-to-cycle variation in peak pressure. A recent study of direct injection of gasoline and E85 with a different multi-hole injector in a different engine has shown higher peak pressures and faster burning rates with E85 in comparison to gasoline at optimised ignition timings for the two fuels [34]. E85 has also been faster than gasoline with a side pressure-swirl atomiser in an optical engine of similar geometry to that used in the present study [33]. Pure ethanol has also been marginally faster than gasoline in the same engine of the current study and at similar conditions but with different multi-hole injector geometry [10]. This shows that there is great sensitivity of ethanol’s combustion development to the injection system and mixture formation.

In order to assess the average time-resolved rate of flame growth and the direction of flame propagation for each fuel, flame areas were obtained on a cycle-by-cycle basis via binarisation of each flame image. Since no image intensifier was used for the current study, flame chemiluminescence intensities were quite low during the initial phase of combustion (0–15° CA AIT). Extensive analysis of thresholding techniques revealed that image processing with changing threshold values to reflect changes in the mean flame intensity was not very reliable for automated analysis, mainly due to random localised areas of image saturation (from ‘droplet’ burning as discussed earlier) which elevated the automatically identified threshold value to levels above the ‘true’ mean and did not reproduce the flame outline accurately. However, as a result of the low levels of background noise, *i.e.* the good contrast between the flames and the background, it was possible to enhance

the images to improve contrast. This process also resulted in reducing the sensitivity of the intensity value used in the thresholding operation which effectively standardised the processing methodology for all flames. The methodology was visually checked ‘on-line’ for each flame to ensure that no overestimation of the flame areas was occurring during the whole period of flame growth. It also needs to be noted that typically flames were ‘clipped’ by the boundaries of the optical piston crown from $\sim 25^\circ$ CA AIT onwards (Figure 24). The following discussion should therefore bear in mind that a degree of masking will be introduced in the quantification of the flames which get convected towards the exhaust-side cylinder walls by the tumble.

The growth in flame size is plotted for all fuels in terms of an equivalent radius in Figure 27, calculated from the flame areas of all individual cycles on the basis of a circle with the same area to that of each flame. The flame radius is used here as an additional metric to the MFB since it can reveal details about flame growth phenomena immediately after ignition, a period which is not resolved well by thermodynamically derived MFB data. It should be pointed out that the maximum flame radius shown corresponds to the edge of the optical piston crown which was 65 mm in diameter. In order to provide a link between Figure 27 and the MFB traces shown in Figure 26, the flame radius was used to estimate the burned gas volume on the basis of an equivalent sphere. For a flame radius of 15 mm, which corresponded on average to $20\text{--}25^\circ$ CA AIT, the mass fraction burned was calculated from the volume fraction to be of the order 5–10%. This is in agreement with Figure 26 and highlights that flame radii in Figure 27 demonstrate phenomena very early in the cycle.

Specifically, the flame radius evolution in Figure 27 shows that E85 and gasoline have very similar rates of flame growth at hot and cold engine operating conditions, however E85 was always marginally slower. The fact that Figure 26 and Figure 27 show the exact same trends between gasoline and E85, suggests that the mechanism of combustion during the early kernel growth of 10% MFB is carried over to the full duration of the combustion event. The flame growth speed (calculated by the gradient of the flame radius curve using a ‘central differencing’ approach) is shown specifically in Figure 28. In the first $5\text{--}7^\circ$ CA AIT (corresponding to values below 0.5% MFB), the rate of flame growth goes down from high values imposed by the initial expansion of the spark. Once this effect has ended, the flame growth speed increases steadily from values of the order 1–2 m/s for 20°C and 2–3 m/s 90°C to about 10–12 m/s at $20\text{--}25^\circ$ CA AIT as ‘masking’ by the optical bore starts affecting the images (with the cold conditions peaking about 5° CA later). Previous studies of ethanol’s laminar burning at conditions relevant to engines [29–32] have shown values of flame speeds of the order 2 m/s which are in accordance with those quantified in Figure 28. E85 shows generally a lower flame speed than gasoline, starting from levels that are lower by ~ 1 m/s at 5° CA AIT, but at $15\text{--}20^\circ$ CA AIT E85’s speed has approached that of gasoline. The laminar flame speed of ethanol is greater than *iso*-octane’s at stoichiometry [29–32], but the current study’s results may also suggest that the flame speed of the gasoline fuel employed was much larger than *iso*-octane’s, and potentially greater than pure ethanol’s as well, thus the trends observed. However, the trends largely disappear if flame speed is plotted against flame size in Figure 29. E85 shows faster growth for the same size to gasoline for both hot and cold conditions when the flame radius is smaller than ~ 15 mm, leading to different levels of stretch. The reduction in flame growth speed for E85 after ~ 20 mm flame radius relative to gasoline is a result of E85’s faster large-scale ‘convection’

towards the exhaust side up until this point; at 20 mm radius, or $\sim 28^\circ$ CA AIT, some ‘clipping’ of the flame area (as dictated by the optical access) is already occurring for E85, whereas for gasoline this occurs later.

Early kernel growth is generally affected by two main features of the flow-field: large-scale convection by ‘bulk’ motion and small-scale turbulence. In practice, and from the experimentalist’s point of view, it is very difficult to decouple these effects reliably and consistently on a cycle-by-cycle basis; the flame ‘sees’ increasing number of sizes and frequencies of turbulent scales as it grows. The flame centroid displacement can reveal information about the level of interactions with the surrounding flow field and/or potential temperature and concentration gradients experienced by the flame as it grows. The flame centroid was calculated from the images of the current study by taking the geometric centroid of each flame and obtaining the ensemble average of 100 cycles. Figure 30 shows the calculated flame centroid co-ordinates from $3\text{--}60^\circ$ CA AIT for both gasoline and E85 at engine coolant temperatures of 20°C and 90°C . The (0, 0) point represents the spark-plug electrode point just below the platinum tip as superimposed on the graph. The end centroid points are the same for all fuels as the flame fills the whole window and this point is directly equivalent to the centre of the piston crown. There are subtle differences in the average flame motion between E85 and gasoline. The E85 flame centroid trajectory is biased to the right side of the combustion chamber and also shows that E85 flames travel further towards the exhaust side than gasoline flames. The flame trajectories for both fuels are not affected significantly by engine coolant temperature. However, at the cold operating condition it appears that E85 follows a different path to that followed at hot conditions. At high temperatures, the increased levels of evaporation and therefore mixture homogeneity may bias the sensitivity of flame motion toward dependency on the local velocity flow field, over the concentration field. This would explain to some extent the greater similarity of the flame motion for the two fuels at high temperatures. Particle Image Velocimetry (PIV) has shown that the mean flow-field on the swirl plane for this engine is nearly symmetrical between the left and right side of the combustion chamber, with a central channel running through the centre of the combustion chamber from intake to exhaust side [35, 36]. On the assumption of ‘same’ average in-cylinder flowfield at ignition timing for both fuels, different trajectories can be an indication of different fuel and temperature distribution of the unburned mixture.

In order to add a temporal dimension to the trends presented in Figure 30, Figure 31 quantifies the flame centroid displacement along the path of movement in time; Figure 32 shows the same relationship relative to flame size. It is clear that E85 is associated with much larger centroid displacement for the same flames size to gasoline’s throughout its flame growth. A ‘global’ centroid speed is calculated by taking the flame displacement at any crank angle and dividing it by the time elapsed after ignition at that crank angle. As such it probably presents a centroid speed biased towards ‘bulk’ movement and essentially filters the effects of local smaller scale flow effects. This is shown in Figures 33 and 34 where the global centroid speed starts from levels of $6\text{--}8\text{ m/s}$; this is higher for E85 than that for gasoline, by about $3\text{--}4\text{ m/s}$. However, the centroid speed drops to values of the order $2\text{--}4\text{ m/s}$ at about 25° CA AIT before starting to increase again gradually at $\sim 30^\circ$ CA AIT (corresponding to the mean duration of $\theta_{\text{Xb10\%}}$) as the flame starts to move from the exhaust side to the intake side; for E85 the levels are always higher than for gasoline. Flow velocities measured by

PIV just below the spark plug in an identical engine at the same operating point have shown mean bulk flow velocities of 2–4 m/s in the direction from the intake to the exhaust side [36]. The magnitude of the centroid speed shown in Figure 33 (~3 m/s) is therefore consistent with mean flow velocities. The higher centroid speeds of E85 may suggest that the different characteristics and timescales of spray formation, impingement and evaporation with E85 may have affected either the bulk flowfield, or the final concentration field, or indeed both, at the time of ignition, leading to differences in the directionality and speed of flame motion.

By calculating the change in flame centroid displacement with time using a local ‘central differencing’ approach for every crank angle that the flame images were acquired at, a ‘local’ centroid speed is also obtained from Figure 30; this now reveals in Figure 35 an estimate of previously masked effects of smaller scales on flame motion, especially at the early stages of flame growth. The early flame growth phase 0–20° CA AIT is characterised by small changes in the local displacement speeds, with E85 and gasoline showing similar trends, albeit E85 starting from much higher values and dropping to lower values than gasoline at hot conditions. The subsequent increase in centroid displacement speeds after about 25° CA AIT is a result of the flames expanding into the unburned regions on the intake side of the engine; E85 moves across the bore towards the intake side earlier than gasoline, just before 25° CA AIT, with gasoline lagging by ~2–5° CA. Plotting the ‘local’ flame centroid speed versus flame radius in Figure 36 shows that E85 and gasoline flames are subjected to different ‘stresses’ during flame growth. The local flame centroid speed is much smaller for gasoline than that of E85 after ~15 mm in radius, whilst E85 has lower centroid speed for flame radius in the range 5–15 mm. It is worth noting that it was only in the latter size range where the flame growth speed was found to be higher for E85 than for gasoline in Figure 29. Therefore, periods of lower ‘local’ flame centroid displacement speed were associated with faster combustion for both fuels.

SUMMARY AND CONCLUSIONS

Spray and combustion characteristics of E85 and gasoline from a multi-hole injector were investigated using high-speed imaging and droplet sizing in a quiescent chamber and in an optical DISI engine. The effects of fuel temperature (20–120 °C) and gas pressure (0.5 and 1.0 bar) were studied and linked to well-known non-dimensional numbers, along with flame growth and motion speeds. The main findings are:

- E85 spray formation was more insensitive to changes in temperature and pressure when in combination compared to gasoline. Specifically, E85 was relatively insensitive to changes in pressure for a given temperature but showed a gradual decrease in penetration with increasing fuel temperatures. This was particularly evident at low gas pressures and led to E85 spray collapse at more severe combinations of pressure and temperature than gasoline’s.
- E85 spray plumes were thinner and more compact at low temperatures and this was reflected in higher spray penetration. Gasoline’s penetration was more insensitive to changes in temperature.
- Droplet sizes were larger for E85 than gasoline at all conditions. Analysis of the gas phase Weber number indicated that secondary atomisation of E85 fell into a different droplet break-up regime

compared to gasoline which predicts larger droplet size distributions. The higher surface tension of ethanol at 20 °C appears to explain why the differences existed mainly at cold conditions. The lower vapour pressure of ethanol at 20 °C also indicated that in-nozzle cavitation phenomena would be more severe with gasoline, probably also contributing to the smaller droplet sizes measured.

- Estimated liquid core break-up lengths from correlations in the literature were unrealistically long, most probably due to the transient nature of injection and the presence of in-nozzle cavitation phenomena.
- At high values of the liquid Weber number, the Reynolds number and the Ohnesorge numbers were probably better able to differentiate between different atomisation characteristics of fuels. The Oh number for ethanol was more than twice that of gasoline's components at standard temperature and pressure. The results highlight the importance of the fuel viscosity on atomisation and break-up efficiency and its particular relevance in light of the increased use of alcohols in gasoline blends. Furthermore, at cold conditions (7 °C), the calculated values of the Oh vs. Re crossed from the atomisation regime into the second wind induced regime and this has certain implications for engine cold-start conditions.
- In the engine, the sprays showed the same trends to those observed in the chamber. The effect of in-cylinder engine flow on spray formation was observed and showed the spray envelope to be very robust to cyclic variations in the flowfield. However, flowfield interactions were helpful in breaking up the spray and also controlled the levels of fuel impinging on the piston crown.
- In-cylinder wetted footprint spray areas showed that initial evaporation rates were similar for E85 and gasoline. In particular, similar trends were observed for the two fuels with the same injection duration (0.8 ms), but E85 showed larger differences between 20 °C and 90 °C engine coolant temperature than gasoline. In fact, gasoline and E85 exhibited close evaporation characteristics at 90 °C but gasoline was quite slower in evaporating at 20 °C. For injection durations corresponding to stoichiometry for the two fuels, E85 exhibited longer evaporation time at 90 °C than gasoline due to the longer injection duration that was required, but at 20 °C the respective differences between E85 and gasoline were much smaller.
- The E85 flames exhibited lower contrast and a smaller number of bright spots than those seen for gasoline. The latter may be associated with lower levels of soot.
- The poorer atomisation and break-up efficiency indicated from droplet sizing and spray imaging for E85 could have contributed to the lower peak combustion pressures, marginally slower mass fraction burned curves and slower equivalent flame radius growth observed for E85 at 20 °C engine coolant temperature. However, the combustion performance of E85 at 90 °C was very similar to that of gasoline; the greater similarity between E85 and gasoline in terms of non-dimensional break-up parameters, macroscopic spray characteristics and average droplet sizes at 90 °C, all suggested small differences in mixture preparation quality at high engine coolant temperatures which were reflected in the combustion process.
- The laminar flame speed of ethanol is greater than *iso*-octane's at stoichiometry, but the current study's results may also suggest that the flame speed of the gasoline fuel employed was larger than pure *iso*-

octane's, and potentially greater than pure ethanol's as well, thus the trends observed. However, E85 had larger flame growth speed than gasoline for flame radius smaller than 15 mm, suggesting different levels of stretch for the same flame size. Additionally, the average in-cylinder pressure for E85 at ignition timing was lower by ~0.2 bar, probably as a result of lower in-cylinder temperatures from greater charge cooling capacity; this combination could also be associated with slower initial flame kernel growth.

- Average flame-centroid motion paths indicated tendency for flames to grow towards the exhaust side in the early stages of flame-kernel growth for both fuels. However, for E85 the flame centroid path was strongly biased towards one side of the combustion chamber compared to gasoline which showed a more 'circular' flame centroid motion. This suggests that the different characteristics and timescales of spray formation, impingement and mixing observed with E85 may have affected either the bulk flowfield, or the final concentration field, or indeed both, at the time of ignition, leading to differences in the directionality of flame development.
- The mean 'global' flame centroid displacement speed generally agreed with the expected mean flow velocity field around ignition timing (~3 m/s). E85 flames showed longer displacement path from the spark plug, as well as larger 'global' flame centroid displacement speed. However, on a 'local' scale, E85 showed slower centroid displacement speed for flame radii smaller than 15 mm in comparison to gasoline and much larger displacement speed after that size; this was found to correlate with those sizes where E85 had faster flame growth.

Current work is focused on studies of in-nozzle cavitation and transition to flash boiling upon spray formation. The fuel matrix has been expanded by other alcohols with different properties to ethanol, like butanol. E10 is also studied since previous work has shown that when ethanol is added to gasoline in small percentages, it alters significantly the distillation curve of gasoline [37–38]. The combustion study is being expanded by laser-sheet imaging to identify more detailed differences in the flame fronts of those fuels.

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Table 1. Fuel Properties.

Fuel Properties	E85	Gasoline
Density [kg/m ³] (20 °C)	784	735–760
Latent Heat [MJ/kg]	0.850	0.34–0.37
Energy Density [MJ/kg] (25 °C)	28.94	42–45
Boiling Point [°C]	84.8	30–190
Reid Vapour Pressure [bar]	0.536	0.54–0.6
Research Octane Number (RON)	109.1	95
H:C, O:C	2.657, 0.411	1.86–1.92

Table 2. Optical Engine Specifications.

Engine Base Type	Prototype V8 Head
Cycle	4-Stroke
Cylinders	1
Valves	2 Intake, 2 Exhaust
Bore	89.0 mm
Stroke	90.3 mm
Compression Ratio	11.15:1
Valve Timings	IVO 24°, IVC 274°, EVO 476°, EVC 6°