IMAGING AND HEAT FLUX MEASUREMENTS OF WALL IMPINGING SPRAYS OF HYDROCARBONS AND ALCOHOLS IN A DIRECT-INJECTION SPARK-IGNITION ENGINE

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ABSTRACT

The latest generation of fuel systems for direct-injection spark-ignition engines uses injection nozzles that accommodate a number of holes with various angles in order to offer flexibility in in-cylinder fuel targeting over a range of engine operating conditions. However, the high-injection pressures that are needed for efficient fuel atomisation can lead to deteriorating effects with regards to engine exhaust emissions (*e.g.* unburned hydrocarbons and particulates) from liquid fuel impingement onto the piston and liner walls. Eliminating such deteriorating effects requires fundamental understanding of in-cylinder spray development processes, taking also into account the diversity of future commercial fuels that can contain significant quantities of bio-components with very different chemical and physical properties to those of typical liquid hydrocarbons. This paper presents high-speed imaging results of spray impingement onto the liner of a direct-injection spark-ignition engine, as well as crank-angle resolved wall heat flux measurements at the observed locations of fuel impingement for detailed characterisation of levels and timing of impingement. The tests were performed in a running engine at 1500 RPM primarily at low load (0.5 bar intake pressure) using 20, 50 and 90 °C engine temperatures. Gasoline, *iso*-Octane, Butanol, Ethanol and a blend of 10% Ethanol with 90% Gasoline (E10) were used to encompass a range of current and future fuel components for spark-ignition engines. The collected data were analysed to extract mean and standard deviation statistics of spray images and heat flux signals. The results were also interpreted with reference to physical properties and evaporation rates predicted by a single droplet model for all fuels tested.

NOMENCLATURE

$a-h$	Coefficients
I	Current
k	Thermal Conductivity
P	Pressure
q''	Heat Flux
R	Resistance
R_0	Base Resistance at Temperature T_0
R_0	Initial Radius of Fuel Droplet
T	Temperature
T_A	Ambient Temperature
T_{drop}	Fuel Droplet Temperature
T_f	Engine Temperature
T_0	Base Temperature
v	Droplet Velocity
V	Voltage

ABBREVIATIONS

ADC	Analogue to Digital Converter
ASOI	After Start Of Injection
ATDC	After intake Top Dead Centre
BDC	Bottom Dead Centre
CA	Crank Angle
COV	Coefficient Of Variation
DISI	Direct Injection Spark Ignition
HC	unburned Hydro-Carbons
HFS	Heat Flux Signal
PIV	Particle Image Velocimetry
RMS	Root Mean Square
RPM	Revolutions Per Minute
RTS	Resistance Temperature Sensor
SOI	Start of Injection
UNIFAC	UNiversal quasichemical Functional group Activity Coefficient

INTRODUCTION

BACKGROUND

The high injection pressures associated with modern Direct Injection Spark Ignition (DISI) engine fuelling systems provide very good atomisation but also result in increased spray momentum that makes the fuel droplets travel quickly across the combustion chamber and impinge onto the piston and liner surfaces. Wall film fuel dynamics and evaporation are important considerations in the optimisation of modern engine systems because they can have a detrimental effect on the emissions of unburned Hydro-Carbons (HC) and particulate matter from ‘pool’ fires. Therefore, designers of modern engine geometries and injection systems need to make sure that liquid fuel wall impingement is avoided. However, basic experiments and modelling of fuel sprays impinging on heated surfaces, *e.g.* by Lindgren and Denbratt [1, 2], have demonstrated the inability of numerical models to capture reliably the behaviour of droplet impingement due to lack of understanding of those generic processes in engines. More to the point, there are only very few reports in the literature on the study and measurement of liquid films inside DISI engines and it is important to note that even those have concentrated more on reporting the diagnostic techniques that were used rather than on providing a holistic description of the processes involved and discussing potential strategies for optimisation of engine performance. The next section presents an overview of those studies to highlight the current state of knowledge.

Liquid Fuel Wall Impingement in Spark-Ignition Engines

Alger *et al.* [3] used an optical engine to image liquid film evaporation off the piston of an engine. The engine was primarily fuelled with propane and a directional injector probe was used to inject liquid fuel directly onto the piston of the engine (Gasoline, *iso*-Octane and *n*-Pentane). The methodology eliminated two main sources of HC emissions, namely flame extinction and low in-cylinder temperatures for post-combustion oxidation and, in turn, allowed focus on piston wetting as a source of HC emissions. The engine was run idling at 750 RPM and at low load at 1500 RPM. Mie-scattering images showed the liquid exiting the injector probe as a stream and directly impacting the piston top. Schlieren imaging was used to show the fuel evaporating off the piston top surface late in the expansion stroke and during the exhaust stroke. Combining HC emissions data from earlier work [4] with images it was shown that the evaporation of liquid fuel from in-cylinder surfaces was a slow process, much of which occurred after the in-cylinder gas temperature had dropped below that required for post-flame oxidation. HC emissions of *iso*-Octane were found to be lower than those of *n*-Pentane; Gasoline’s HC emissions were similar to *n*-Pentane’s, despite *n*-Pentane’s higher mass diffusivity that should have promoted lower HC emissions. This was an indicator that volatility played a different role in cases of evaporating films compared to when such fuels are present as droplets in air and evaporate faster due to their higher vapour pressures. The observations were linked to the Leidenfrost phenomena (*i.e.* when an insulating layer of fuel vapour between the liquid fuel and the wall reduces the amount of thermal conduction and slows the heat transfer rate) as the piston surface temperatures rose more than ~60 °C higher than the fuel’s boiling point.

Drake *et al.* [5] developed a high-speed refractive-index-matching imaging technique for time and space-resolved measurements of fuel-film mass on the quartz piston window of an optical DISI engine, operating over a range of fully warmed-up stratified-charge conditions with both a pressure-swirl atomiser and a multi-hole injector. Measured fuel-film mass was found to be a small percentage of the total fuel injected with the pressure-swirl

injector (maximum of ~1% with Gasoline and ~0.1% with *iso*-Octane). Switching to the multi-hole injector reduced the maximum wall film mass by a factor of 10, decreasing smoke emissions to negligible levels and reducing HC emissions significantly. Vaporisation of fuel films from the piston was relatively slow; films persisted past 60° Crank Angle (CA) After intake Top Dead Centre (ATDC).

Camacho and Hall [6] used a piezoelectric sensor to measure the mass of fuel that impacted on the piston top during injection in an optical DISI engine. Gasoline was delivered by a swirl injector with 60° cone-angle located vertically and aimed at the centre of the piston. The engine was motored at 600 RPM with the intake pressure set at 0.69 bar. The injection pressure used was quite low (5 bar) and fuel was injected over a period of 11° CA. Total fuel masses injected were between 16 and 58 mg for lean and rich fuelling equivalence ratio of 0.55–2.0. Due to temperature limitations, all fuel-film mass measurements were taken under motored conditions. The measurements showed that the fuel mass deposited on the sensor surface was small, typically less than 2 mg, which corresponded to a film height of 32 μm . The fuel mass followed a general trend increasing in value and reaching a maximum for injection timings when the piston approached TDC. The maximum measured mass for stoichiometric operation was 2.3 mg at TDC and corresponded to 6.3% of the fuel injected. This percentage was very large considering the ratio of the sensor and piston areas, suggesting that the majority of the injected fuel became a liquid fuel film. Near Bottom Dead Centre (BDC) the mass of the fuel film did not reach values higher than 0.5 mg (8 μm in height). Strong air bulk flow pattern dependence was found in the measured mass deposition when the intake valves were open. The average evaporation rate of the fuel film mass was 3.3 $\text{kg/m}^2\text{s}$, and was nearly the same for all injection timings. The average measurement uncertainty was calculated as ± 0.56 mg for stoichiometric operation and, hence, the sensor could not provide good quantitative measurements for very small fuel film masses.

Park and Ghandi [7] measured fuel film temperature and thickness on the piston crown of a DISI engine equipped with a swirl injector (52 bar rail pressure) for wall-guided stratified operation, under both motored and fired conditions (600 and 1200 RPM) using a fibre-based laser-induced fluorescence method wherein a single fibre delivered the excitation light and collected the fluorescence. The overall equivalence ratio used was 0.3 and the amount of *iso*-Octane injected was 12 mg. The fibres were installed in the piston crown of a Bowditch-type optical engine and exited via the mirror passage. The fuel used for the fuel film temperature measurement was a 4% 2,3-hexanedione solution in *iso*-Octane. The measured fuel film temperature was found to closely follow the piston crown metal temperature, which was measured with a thermocouple. The fuel film thickness was found to be quite small however, less than 10 μm , for both motored and fired conditions performed at the same piston temperature.

Cho and Min [8] carried out a two-dimensional visualization of a liquid fuel film on a quartz cylinder liner in a spark-ignition engine rig. The visualization method was based on laser-induced fluorescence and total reflection. After calibration, the wall-wetting fuel mass on the cylinder liner was measured for different valve lifts and port-injected fuel masses using Gasoline. The air velocity at the intake port was kept constant at 14 m/s. The maximum fuel film thickness on the liner was about 45 μm for a valve lift of 8 mm and an injection duration of 10 ms. At 8 mm valve lift, the mass fractions of the wall-wetting fuel were ~60% and ~55% with injection durations of 10 ms

and 5 ms respectively while about 13% of the injected fuel at 2 mm valve lift was wetted onto the quartz cylinder liner for both 10 ms and 5 ms injection durations.

Simpler methods of measuring fuel impingement on in-cylinder surfaces have involved using heat flux diagnostics. Shayler *et al.* [9, 10] used surface mounted heat flux sensors in the intake port of a port-injected Gasoline engine to investigate the heat transfer between the wall film and the surrounding gas phase. This allowed a simple impingement detection method for use in a running engine and various sensors were placed around the port surface to investigate possible impingement locations. The signals showed peaks in the heat flux when impingement occurred initially some time after fuel injection and then a second peak later in the engine cycle during the intake stroke as a result of a period of rapid evaporation which was associated with backflow into the intake ports. It was found that at cold engine conditions, the highest levels of heat flux were recorded during this period, suggesting that little fuel evaporation took place initially on impact with the port walls. In another study Shayler *et al.* [11] used *iso*-Octane to compare the impingement heat flux signature with that of Gasoline. It was found that at fully warm conditions the signal for *iso*-Octane was higher due to vapour pressure differences and that it disappeared more rapidly than for Gasoline due to the heavy fractions of Gasoline remaining on the sensor for longer. At cold conditions, the signals for the two fuels were found to be very similar.

Bauer *et al.* [12] also performed heat flux measurements in the intake port of a spark-ignition engine under firing conditions. They used square probes with 10 mm sides and 0.5 mm thickness. Measurements were limited however, as very thin films could not flow over the sensor and only fuel targeting locations could be investigated. In similar methodology to that of Shayler *et al.* [9–11], heat flux signals for ‘dry’ conditions (using propane) and ‘wet’ conditions (using liquid fuel) were obtained and the relative difference was used to calculate the net heat transfer. This value was considered to be the sensible and latent heat required to heat and vaporise the impinging fuel. When the predicted values were compared to experimental results a reasonable agreement was reported, apart from cases at high wall temperatures where the combination of other heat transfer modes made estimation more difficult due to many unknown parameters.

Stevens and Steeper [13] reported fuel impingement imaging in a DISI engine with a centrally mounted swirl injector. Gasoline LIF based on UV excitation allowed visualisation of impingement. Intermediate and late injections produced films that persisted through the compression stroke and produced clear pool fires; films created by early injection evaporated during the expansion stroke and did not ignite.

Split-Injection Strategies for Spark-Ignition Engines

Various flexible injection strategies have begun to emerge in order to reduce the spray momentum, whilst maintaining the atomisation advantages of high injection pressures. Specifically, splitting the injection process into smaller multiple injections within the same cycle can offer benefits over certain operating regimes of the engine. Studies of split injections with multi-hole injectors for DISI engines have reported improved air entrainment, mixing and evaporation under late injection strategies for stratified operation, *e.g.* [14, 15]. It needs to be noted though that the main characteristic of multi-hole injectors is that they produce very robust spray plumes consisting of individual vapour branches rather than forming a complex cloud structure. [16]. This is mainly true under stratified conditions with high gas in-cylinder pressures but with injection strategies for homogeneous mixture preparation, *i.e.* spraying into low in-cylinder pressure during the intake stroke, these

vapour branches become less stable and more prone to flow-field interactions, especially under fully-warm engine conditions that lead to alterations in the nominal spray pattern from flash boiling mechanisms [17–19]. Stach *et al.* [20] reported improvements in wall wetting with early split homogeneous injection strategies using multi-hole injectors, but their studies were limited to non-collapsed spray conditions.

Fuel Properties

No published work was found on in-cylinder fuel impingement in latest design DISI engines using multi-hole injectors with single and split injection strategies under homogeneous operation and at conditions of non-collapsed and collapsed spray conditions with fuels of various types for direct comparison. More to the point, when considering that predicted future fuel stocks will include a significant bio-derived component to promote sustainability and reduce CO₂ emissions, understanding the effect of new bio-components on spray and impingement processes in DISI engines that are very sensitive to fuel properties is an essential part of the challenge to adopt widely more sustainable fuels. Gasoline already contains 5% Ethanol (E5) in some countries but its use will have only limited impact on CO₂ emissions. Therefore, some markets demand much higher proportions (E85 or E100). However, not all components on the vehicle which are compatible with hydrocarbons would necessarily be compatible with blends containing large amounts of Ethanol and it will take some time for such issues to be resolved. At the other end, the vapour pressure of blends like E10 is higher than the vapour pressure of either Ethanol or Gasoline because the mixture deviates from the ideal solution behaviour (Raoult's law) due to the intermolecular forces. This deviation significantly alters the distillation curve of the blend in comparison to that of pure Gasoline [21, 22]. Other alkyl alcohols, such as Butanol, have also been suggested as possible Gasoline components. Having twice as many carbon atoms as Ethanol, Butanol is more hydrocarbon-like in its properties; however Butanol lags behind Ethanol in terms of commercial production.

PRESENT CONTRIBUTION

There are no experimental data which clarify the role that the transport properties of current and future fuels have on spray development and wall impingement with modern injection strategies over a realistic range of DISI engine operating conditions. The main objective of the current work was to compare the in-cylinder spray development process and wall impingement with a range of hydrocarbons and alcohols at nominally same running conditions. Within the scopes of the current work it was necessary to investigate the extent of fuel impingement and heat flux at key in-cylinder locations under motoring and firing engine conditions. This was done by using a fully optical piston that allowed imaging of the in-cylinder spray formation process up to and including direct impingement onto the liner walls. Heat flux sensors were also installed flush into the liner in order to quantify the level and timing of spray impingement. The application of the heat flux sensors technique was developed from a perspective that the measurements would allow certain insights into the spray development process without the recourse to optical diagnostics that are more challenging to apply cost-effectively and routinely, especially if any quantification of the underlying processes is sought.

EXPERIMENTAL APPARATUS AND PROCEDURE

RESEARCH ENGINE

A single-cylinder optical engine was used to investigate the in-cylinder spray behaviour and wall impingement under realistic operating conditions. More details about the engine can be found in previous publications by the

current authors on in-cylinder spray dynamics and combustion with various fuels [23–25]. The influence of engine temperature was observed by using engine head coolant temperatures of 20 °C, 50 °C and 90 °C. Enough heat-soak time (30–40 min) was allowed at 90 °C so that the engine head and liner components acquired enough thermal inertia to simulate fully warmed-up conditions. Under such conditions, the engine head temperature was also considered to be a measure of the temperature of the fuel (T_f) that was sitting in the injector that was installed centrally in the engine's head. The engine was run at 1500 RPM primarily under low load motoring or firing conditions (0.5 bar intake pressure); some measurements with the engine motoring at full load (1.0 bar intake pressure) were also acquired for comparison.

INJECTOR

A multi-hole injector designed for vertical installation in close spacing arrangement with the spark plug of a DISI engine 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. Plumes #1 and #6 had a 58.5° inclination with respect to the vertical axis and 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 geometry, nozzle-hole angles and spray formation in a quiescent environment and in a running DISI engine can be found in previous studies by the current authors [17, 18, 23, 24].

FUELS

Five fuels were investigated: a typical commercial grade Gasoline (RON95), *iso*-Octane, Butanol, Ethanol and E10 (blend of 10% Ethanol with 90% Gasoline). A standard commercial grade European Gasoline contains several hundred hydrocarbons, typically about 25–30% C5 or lower, 30–40% C6–C8 and the remainder C9–C10 hydrocarbon chains. *Iso*-Octane is a single component of gasoline with boiling point temperature of 99 °C at atmospheric pressure; Butanol boils at 117°C while Ethanol boils at 78.4°C. Gasoline's distillation curve, along with the boiling points of the single component fuels are shown in **Fig. 2**. The boiling points of *n*-Pentane and *o*-Xylene are also shown in **Fig. 2** for reference purposes and in order to highlight two high and low volatility hydrocarbon components in Gasoline, respectively. E10 was also tested because the vapour pressure of E10 is higher than the vapour pressure of either Ethanol or Gasoline and its distillation curve in **Fig. 2** reflects clearly this effect. More information about the physical properties of the fuels that were tested can be found in earlier publications by the current authors [19, 24].

SPRAY IMAGING

The research engine provided a variety of optical access planes into the combustion chamber. For the purposes of the current work, the 'standard' aluminium piston crown with the quartz optical insert was replaced by a full-bore optical crown made of Perspex® in one piece. The 'standard' optical crown and the Perspex® crown are both shown in **Fig. 3** for direct comparison. The Perspex® crown connected to the Bowditch piston using the same thread as the 'standard' metal/quartz crown and was therefore 'plug-n-play' with the engine set-up. This configuration extended the 65 mm diameter optical access of the metal/quartz crown to 89 mm in diameter, *i.e.* the full bore diameter. Additionally, the Perspex® crown was designed with a 'fish eye' lens configuration (by keeping the top of the crown flat and 'contouring' the inside/lower part of crown) to allow full clear optical access up to the liner walls via a 45° mirror housed under the crown 'inside' the Bowditch.

A high-speed camera was used throughout the present study (Photron[®] APX-RS) to obtain images of spray development. Typically a frame rate of 9 kHz was used (corresponding to 1 image/° CA at 1500 RPM engine speed) with 640×480 pixel resolution. The camera had on board memory of 2.14 GB which allowed continuous acquisition of 6826 greyscale images at 640×480 resolution. Typically 100 cycles of data were acquired in continuous mode (*i.e.* consecutive cycles were imaged) with 60 images/cycle for both motoring and firing conditions. This allowed ‘mapping’ of in-cylinder events over a 60° CA period from SOI, sufficient to visualise both the injection period and post-injection mixing stages. Once data acquisition was complete, images were downloaded from the camera via IEEE 1394 FireWire to a computer system for image storage as 8-bit Tagged Image File Format (TIFF) files. The camera was triggered externally with a TTL pulse of 1ms duration supplied by the AVL[®] ETU system used for synchronization of in-cylinder pressure data acquisition and other instrumentation. All the CA degrees for any imaging presented refer to the falling edge of the triggering pulse, *i.e.* images at 3° CA refer to light acquired between 2° CA and 3° CA for a 1 image/° CA resolution.

The illumination source used in the present work was a New Wave Research[®] Pegasus Nd:YLF laser at 527 nm (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. A cylindrical optics unit allowed formation of a laser sheet 0.5 mm thick so that fuel droplets crossing that plane could be captured through Mie-scattering. The laser firing frequency was synchronised with a pulse train from the camera’s frame rate.

HEAT FLUX MEASUREMENTS

Heat Flux Sensor

The levels of wall heat flux at spray impingement locations were recorded using fast-response heat flux sensors made by Vatel[®] Corporation (model HFM-7), together with Vatel[®] amplifiers (model AMP-6) [26]. This sensor is based on thermocouple technology and essentially joins many thermocouple pairs in series to form a flat plate differential thermopile. This plate actually consists of a ceramic thermal resistance layer. Since the thermocouples are arranged in series, the total output voltage V_t of the thermopile is proportional to the number of thermocouples. The temperature difference across this thermopile allows the heat flux to be determined. The heat flux is proportional to the thermal conductivity k of ceramic material and the temperature difference, but is inversely proportional to thickness of the resistance layer t , *i.e.* $q \propto (k/t)V_t$.

Two types of measurement are made simultaneously with this sensor. The first is a temperature measurement from a Resistance Temperature Sensor (RTS) element, consisting of a thin film deposited in a loop pattern around the outer edge of the sensor face. The second is a Heat Flux Sensor (HFS) measurement from the thermopile that is 6.4 mm in diameter and occupies most of the surface area on the sensor face. This is not equivalent to a thermocouple signal; it acts as a differential source and therefore is not grounded [28]. The fastest sensor response time is of the order 17 μ s if used uncoated as shown in **Fig. 4**.

The RTS measurement is critical to proper heat flux measurement, because the HFS is temperature dependent. The RTS relies on the fact that the thin film resistance changes as a function of temperature. This function is close to linear for most temperatures of interest, although strictly speaking it is better described by a cubic polynomial. The function is characterised for each sensor at Vatel and takes the form:

$$T = a \cdot R^3 + b \cdot R^2 + c \cdot R + d \quad (1)$$

where T is the temperature [$^{\circ}\text{C}$], a, b, c, d are the coefficients of the polynomial, R is the electrical resistance of RTS [Ω]. The resistance of the RTS is related to the voltage output from the amplifier by:

$$R = \frac{V_{RTS}}{I_{RTS} G_{RTS}} + R_0 \quad (2)$$

where V_{RTS} is the voltage output of the RTS amplifier channel [V], which may be positive or negative, I_{RTS} is the excitation current through the RTS used to generate V_{RTS} (equal to 1×10^{-4} A) and G_{RTS} is the amplifier gain for the RTS channel. The resistance value from equation (2) can then be entered into equation (1) to determine the temperature T . Then the heat flux can be calculated from:

$$q'' = \frac{V_{HFS} / G_{HFS}}{g \cdot T + h} \quad (3)$$

where q'' is the heat flux [W/cm^2], V_{HFS} is the amplified voltage signal from the HFS [μV], G_{HFS} is the amplifier gain for the HFS channel and g, h are coefficients for the relationship between sensitivity and temperature.

The amplifier signals were digitised with a 12-bit ADC data acquisition card within a Labview[®] environment and logged with a sampling rate of 45 kHz. The digitisation error was reduced to a minimum by adjusting the voltage range over which the 12-bits were used. Signal amplitude was in the range -6.0 – 4.0 V for high signals and as low as ± 1 V for very small signals. Once logged, the two signals (HFS and RTS) required processing in order to obtain the final heat flux and temperature measurements. To do this, an algorithm was coded in MATLAB[®] to batch process the raw data and to output all the relevant parameters and statistics.

Installation of Sensors and Test Conditions

A short steel cylinder liner ring was designed to house the heat flux sensors and replace the optical quartz-ring available to use with the $\frac{3}{4}$ stroke water-cooled cylinder liner. The advantage of having a removable ring in which to insert the heat flux sensors was that it gave effectively infinite flexibility of sensor location around the cylinder bore (except around the head bolt locations where engine supporting columns were in the way). The steel ring was machined with multiple inserts to cope with simultaneous heat flux sensors mounted flush with the cylinder wall, however to eliminate uncertainties with using different sensors only a single sensor was used throughout the duration of the investigation. A replica steel ring was thus produced with one sensor insert only and this was used throughout. The sensor was secured with a capture nut, which was designed and manufactured at UCL and then shipped to Vatell[®] to be integrated into the HFS micro-connector assembly. This was necessary due to the particular micro soldering requirements of the assembly and subsequent calibration procedure, which could only be carried out by the manufacturer.

The steel ring with the heat flux sensor installed is shown in **Fig. 5**. The main purpose of this arrangement was to allow investigation of the region around where plumes #1 and #6 might be expected to impinge, as a result of their injector nozzle angles and orientation in the combustion chamber (as defined in **Fig. 1**). The placement of the sensors was defined by imaging tests carried out at different planes along the length of the cylinder liner, in order to identify the most severe locations of impingement around the bore; the final plane selected was 17 mm below the head gasket and based on that dimension the steel ring was designed and installed in the engine with the heat flux sensors in place. Initial location sweeps around 140° of the cylinder bore at 10° increments were used to

obtain heat flux measurements for different engine head coolant temperatures and fuel types. The final sensor locations used in the engine test matrix ranged from -20° – $+90^{\circ}$.

The heat flux measurements were carried out for *iso*-Octane, Gasoline, Ethanol, Butanol and E10. The heat flux sensor ring was clamped in place and the cylinder liner operated with the same coolant temperature as the engine head using the same cooling system; all tests were carried out with both the engine head and the cylinder liner temperature controlled simultaneously to 20 °C, 50 °C and 90 °C. Based on previous work by the current authors [23, 24], a ‘single’ injection strategy with a pulse width of 0.8 ms was primarily used for all fuels and for comparative purposes. A ‘split’ injection strategy to break-down the momentum of the spray whilst delivering the same amount of fuel as the single injection) was also applied in order to study subsequent effects on levels of liquid wall impingement and heat flux; specifically, following earlier studies by the current authors, a triple injection strategy with three smaller injection TTL pulsewidths each of $\sim 1/3$ duration of the single injection duration was used [23].

All injection set points were repeated 200 times in order to calculate relevant statistics. The sensor location sweeps were adapted to cover the main impingement locations for the tested operating conditions and ranged from -30° – $+90^{\circ}$ in 10° increments. Signal post-processing was achieved by writing a dedicated algorithm in MATLAB[®]. The code allowed the reading of multiple data files simultaneously and performed a number of operations, from splitting the data into individual injection events to calculating new variables using equations (3)–(8). Statistical analysis was also performed to obtain ensemble average values, RMS and COV (RMS/average) for the full signals as well as individual points of interest *e.g.* the magnitude and time reference of the peak heat fluxes.

RESULTS AND DISCUSSION

SPRAY DYNAMICS AND IMPINGEMENT ON THE CYLINDER LINER

General Characteristics

Spray images on a horizontal plane 17 mm below the head gasket are shown in **Fig. 6** for nine random cycles at 15° CA ASOI (or 95° CA ATDC), as viewed through the fully optical piston crown under motoring engine conditions. The large levels of scattering in the centre of the images are due to the four plumes penetrating through the laser-sheet down into the cylinder (corresponding to plumes #2, #3, #4 and #5) and on the right side of the images, due to plumes #1 and #6 crossing the laser sheet close to or at the cylinder walls. At this timing after the end of injection the plumes were completely detached from the injector and therefore were subject to free interactions with the in-cylinder bulk flows; this was more noticeable for plumes #1 and #6 by their different locations from cycle to cycle. In these images, the thin white line around the circumference of cylinder corresponds to the cylinder liner/head gasket whilst the ‘black ring’ shown next to this corresponds to the $\frac{1}{4}$ cylinder length quartz ring annulus used to feed the laser-sheet into the cylinder. The imaging plane was located just below mid-height in the quartz annulus or ~ 15 mm below the cylinder/head gasket plane. In some cycles, like the one represented by the top-left image of **Fig. 6**, plumes #1 and #6 were nearly symmetrical and the spray tip impinged onto locations of the cylinder walls as would have been expected without the presence of engine flow. The other images show other examples of cyclic variations in the final location of these plumes relative to the cylinder walls.

This imaging exercise used several horizontal planes to locate the main regions of wall impingement; it was identified that although the location of impingement changed from cycle-to-cycle, the plane of impingement was quite repeatable and justified the use of sensors positioned on a single horizontal plane. The flow field close to this plane (~5 mm above it or 10 mm below the head gasket) has been investigated in an identical motoring engine (without fuel injection) using Particle Image Velocimetry (PIV) by Jarvis *et al.* [27]; it was clear from that study that the nature of cyclic variations in the flow on this plane was compatible with the changes in the spray plume locations observed in **Fig. 6**. Cyclic variability in the flow field and subsequent interactions with the developing spray were therefore probably able to affect the levels of liquid impingement on the cylinder liner walls and might even be able to change the airborne fuel quantity on a cycle-by-cycle basis. This hypothesis was one of the main drivers of the current investigation as it could have noticeable consequences on the overall air-fuel ratio and potentially contribute to small changes in the local air fuel ratio around the spark plug at ignition timing known to significantly affect early flame growth [28, 29].

In order to study the impingement dynamics in more detail, one of the plumes, #6, was chosen to perform a high-magnification imaging study, as shown in **Fig. 7** (corresponding to about one quadrant of the cylinder bore). In the six instantaneous images shown, the spray tip was seen breaking-up and impacting on the cylinder liner wall for representative cycles. The detail provided by these images was such that vortices of the order of ~5 mm diameter, could clearly be seen interacting with the spray just behind the spray tip and this was observed in the high-speed video consistently as being one of the key mechanisms resulting in the initial detachment and break up of the spray tip from the main body of the spray plume. The spreading of the liquid fuel on the cylinder walls immediately after impact can also be seen on some images, and significant ‘splashing’ of the fuel was observed with re-entrainment of atomised droplets back into the cylinder flow. For the more intact plumes, such as the middle-bottom image, there were be significant oscillations that distorted the plume upstream of the spray tip; these appeared to be have been caused by a combination of classical turbulent instability induced break-up coupled to interactions with vortices; however, the strong interactions with the cylinder flow at small and large scales made decoupling of these mechanisms difficult with the current experimental configuration.

To give a perspective of temporal spray development up to the cylinder walls using this transparent piston, the Mean and RMS spray images are also shown for a few crank-angle timings ASOI in **Fig. 8**. The mean image at 10° CA ASOI clearly show the footprints of the four central plumes (#2–#5) and the two shallower plumes (#1 & #6) penetrating towards the spark plug, deflected by the intake valve flows. At 13° CA ASOI plumes #1 and #6 just come into contact with the light-sheet but have not yet reached the cylinder walls. The average timing of liquid impingement on the cylinder walls was clearly observed to be ~15° CA ASOI (95° CA ATDC) for plume #6 and the spatial envelope of impingement occurred approximately over 1/8 of the cylinder circumference (~45°) as seen on the lower right portion of the images. This is also illustrated on the RMS images as area(s) of highest intensity on the cylinder liner surface.

Multiple Injection Strategy

Injector characterisation by van Romunde and Aleiferis [17] showed that the needle response time and the appearance of first fuel at the nozzle tip was very stable at ~ 300±30 µs 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. Therefore, the maximum number of injections that could be used per cycle at low-load 1500 RPM engine conditions was three (for stoichiometric equivalent fuelling), between 0.25 ms and 0.3 ms each for *iso*-Octane). For initial spray imaging shown, the pulse-widths used were 0.25 ms, 0.28 ms, 0.25 ms for the three injections respectively, with SOI 60°, 70°, 80° CA ATDC; a 10° CA dwell time between injections was used to allow the injector needle to seat properly before receiving another trigger pulse (the reader may refer to [23] for further details and a timing diagram, if needed). Recent imaging studies in the same engine on the tumble plane have shown that the shorter injector pulse widths worked well in controlling spray momentum and plume tip penetration, hence, consequently, reducing significantly the levels of liquid impingement on the piston crown. The reduction in spray momentum allowed the in-cylinder flow to overcome the spray plume penetration and quickly entrain the droplet and ligaments into the bulk flow. Using the full-bore optical piston in the current work, potential reductions in the levels of impingement at the cylinder walls were investigated. Specifically, high magnification imaging with the full-bore optical piston allowed comparison of the single and triple injection strategies, as shown in **Fig. 9**. The single injection event clearly shows plume #5 penetrating vertically into the cylinder, saturating the image on the left hand side. For the plumes directed towards the spark-plug (plume #1 and #6) only plume #6 was visualized as it penetrated towards the laser-sheet and made contact with it close to the cylinder walls on the right hand side of the images. It should be noted that the image shown for 95° CA ATDC in this series was equivalent to those shown in the variability mosaic of **Fig. 6**; however, in **Fig. 9** it was included to provide a temporal description of the penetration and impingement processes. To highlight the significant reductions in the direct wall impingement levels observed using triple injection strategy, a period after the end of injection for the second and third injections was used in **Fig. 9**. No images for the first injection sprays are shown because negligible luminosity was detected from 60°–70° CA ATDC during which this first injection occurred. For the second injection, the left of the images showed some saturation due to the penetration of the central plumes, although the intensity levels were much lower than for single injection due to the smaller amounts of fuel injected, and an intact plume #6 reaching the cylinder walls was not observed in contrast with the single injection strategy. While some luminosity around the vicinity of the cylinder walls was detected for later timings around 83° CA and 95° CA ATDC, the latter corresponding to 15° CA ASOI for the third injection, these were found to be very small levels and while some of these droplets and ligaments *were* eventually carried towards the cylinder walls by the bulk flow, they did not have sufficient momentum on their own to produce direct impingement. Such differences in the levels of impingement for different injection strategies were investigated further using the heat flux sensors described in the remaining sections of this paper.

SPRAY IMPINGEMENT HEAT FLUX IN MOTORING ENGINE

Heat Flux Signal Characteristics

In order to give the reader a visual aid of reference for the locations of spray impingement around the cylinder in relation to sweeps with the heat flux sensor, a typical spray produced with single injection strategy under motoring conditions has been overlaid with a polar grid (with the origin located at the centre of the piston crown window, *i.e.* centre of cylinder bore) in **Fig. 10**. The mask applied used the standard (*i.e.* not full-bore) optical piston crown to show the lower right plume clearly distorted towards the 20°–70° locations on the cylinder

circumference; this was the cylinder region where most impingement was initially observed for *iso*-Octane under cold motoring conditions. A similar image for an injection event into the cylinder of the engine under static conditions (1.0 bar quiescent environment) is also shown in **Fig. 10** for comparison and in order to further clarify the distortion of the spray plumes inside the running engine with reference to the polar co-ordinates.

The heat flux signal characteristics are discussed for a mean signal trace averaged over 120 engine cycles and starting from the intake TDC point in the cycle (**Fig. 11**). The first feature of interest of the signal was at the beginning of the intake stroke of the engine cycle, which occurred due to the negative valve overlap used in this work, resulting in a small vacuum before the opening of the intake valves; this depression was seen to cause a small dip in the heat flux signal as heat was drawn out of the sensor but it recovered quickly back to the zero level when the intake valves opened at $\sim 30^\circ$ CA ATDC. For the dry motoring case (shown in black in **Fig. 11**) the signal during the intake stroke was mostly flat since the air temperature was at the same temperature as the sensor, 20°C ; for the fuel injected case liquid impingement resulted in a negative signal $\sim 95^\circ$ CA ATDC, which then gradually returned to the zero level before intake valve closing, as the impinged fuel film evaporated (and/or ‘washed off’ by the incoming flow) from the sensor face. When the intake valves closed, compression raised the gas temperatures in the cylinder and a positive heat flux was recorded. This peak was seen to occur at TDC corresponding to peak compression temperatures. Before this peak, the signal showed a slight dip, which was caused by the piston top covering the sensor during the last $\frac{1}{4}$ of the engine stroke; it should be noted that a low ring-pack piston was used to avoid the piston rings scrapping past the sensor so the effect was simply due to masking of the sensor face by the piston extended top land. In general, greater charge cooling with the injected case resulted in lower compression heat fluxes; however, the variable nature of heat flux measurement at this point on a cycle-to-cycle basis meant that compression heat fluxes were more difficult to interpret with the current experimental arrangement. For more accurate compression heat flux measurements, the sensors should be placed inside the pent-roof in order to avoid interactions with the piston motion.

After TDC, the heat flux decreased rapidly and reached a minimum at the point where the piston uncovered the sensor face again $\sim 400^\circ$ CA on the graph. The signal was slightly negative at this stage probably due to the fast flows within the small crevice space between the piston and the sensor, but once the sensor was fully uncovered the signal quickly returned to zero. For the injected case, there was a temporary negative dip around 480° CA, which was believed to result from a thin fuel film remaining on the sensor face even after compression. The sudden flow past the sensor when the exhaust valves opened was believed to result in evaporative cooling producing this effect; it should be noted that this characteristic was never observed under motoring conditions without fuel injection. When the exhaust valves opened some of the residual heat from compression caused a sustained rise in the heat flux signal, which stopped only towards the end of the exhaust stroke as the flow velocity past the sensor died down. Sweeping the perimeter of the bore with the sensor at different locations indicated that the main areas of spray impingement for 20°C occurred around the 30° – 60° locations and that rather lower levels occurred for other locations, hence the results were fully compatible with observations obtained from imaging tests, as will be discussed in more detail later.

At full-load conditions (1.0 bar intake pressure), the compression peak heat flux was higher than for low-load due to the higher compression pressures and temperatures; the exhaust heat flux was also very different compared to

low-load. It was believed that the higher in-cylinder temperatures generated with full load resulted in local heating of the sensor during compression, to similar temperatures as those of the gas; as the heat flux was a differential signal, it was plausible that the flow through the exhaust valves was at similar temperature to the sensor and therefore registered zero heat flux during this period. There was also evidence of hotter gas temperatures for full load on the intake stroke part of the signal. However, although there was a similar negative dip to that of low-load conditions, the resulting signal after the valves had opened was very different; at full load the heat flux signal registered positive values up to 1 W/cm^2 .

The signals were averaged over a number of consecutive cycles and therefore the average signal also shows characteristics that resulted from previous cycle effects. The main feature arising from these was found during the intake stroke. For the low-load case, this was reflected in the lower heat flux measured with injection, immediately after intake valve opening compared to motoring only conditions $\sim 60^\circ \text{ CA ATDC}$. Due to prior-cycle impingement events and the lack of complete film evaporation, particularly at cold temperature conditions (20°C), the incoming flow past the sensor resulted in some convective cooling which was not seen for motoring conditions without fuel injection. As will be shown later, this effect disappeared at high temperatures. At full load this effect was also observed and was more significant because the ‘residual gas’ temperatures were likely to be higher. Comparison between the impingement peak heat flux at part and full load engine operation also showed that the levels were on average lower for full load by at least 50%; it is likely that this resulted from a contribution of both lower levels of impingement due to the strong flow interactions breaking up the spray plumes more actively during injection and of the lower levels of immediate evaporation from the sensor face upon impingement due to the relative low vapour pressures compared to the ambient pressure. Mean signals superimposed from different sensor locations around the cylinder showed that locations which produced large signals in the intake stroke also produced large negative heat fluxes in the exhaust stroke and this was confirmation that the fuel film produced from the impingement event was not evaporated fully during the compression stroke.

The third graph in **Fig. 11** shows the low-load triple injection heat flux trace. The sensor was able to register clearly the three individual spray impact events on the sensor face. The highest peak was generally the first injection followed by the second and then the third; this was thought to be related to different interactions with the flow field, which affected the levels of spray reaching the walls for each injection event. However the levels of heat flux registered were significantly lower than for single injection strategy which suggested much lower levels of impingement, as observed with the full-bore imaging described earlier. Lower levels of impingement for triple injection were further supported by the absence of a negative dip immediately after EVO which was clearly observed for single injection. In fact, the triple injection signal was very similar to the motoring trace without injection; before IVO there was no evidence of evaporative cooling, suggesting the absence of any fuel film on the sensor face, and in the exhaust stroke, a lower rate of rise in the signal compared to the motoring trace was probably caused by only very small levels of convective cooling affecting the overall heat transfer rate.

Fuel Type and Temperature Effects

The mean heat flux signals measured at the location of maximum impingement for the different fuels are shown in **Fig. 12** for 20°C , 50°C and 90°C fuel temperatures (T_f). These figures were used to compare features of the heat flux signal, concentrating on inlet and exhaust stroke events. For *iso*-Octane, the motoring signal showed that the

impinging fuel on the sensor was to a large extent evaporated during compression, as only small effect from evaporative cooling could be seen on the exhaust stroke part of the signal for most temperature conditions. For Gasoline, only at 20 °C were there signs of liquid still on the sensor face during the exhaust stroke, while E10 was very similar to Gasoline in this region. Ethanol showed the largest heat flux signal in the exhaust stroke due to the most significant levels of evaporative cooling, with the highest values recorded at 20 °C, followed by 50 °C; at 90 °C the sensor temperature was well above the boiling point of Ethanol and so large heat fluxes might have been expected. However, no negative dip occurred for 90 °C on the exhaust stroke and the signal showed surprisingly neutral levels of heat flux in this region compared to the other fuels, indicating that full evaporation of the liquid film had already taken place. Indeed this was also borne out by the zero levels of heat flux recorded early in the intake stroke compared to those recorded at lower temperatures.

These prior-cycle effects resulting from unevaporated fuel films on the sensor also affected the heat flux during the valve overlap period. As the intake valves opened (from ~30° CA ATDC), the incoming flow past the sensor produced significant levels of heat flux through evaporative cooling of the order of 2.5 W/cm² at 20 °C for Ethanol. For Butanol, the 50 °C tests were remarkably similar to Ethanol at 20 °C. Compared to Ethanol the heat flux at low temperature for Butanol was very insensitive to mass impinging however. Under further inspection of the cyclic signals and the sensor surface after tests, it was found that the high viscosity and boiling point of Butanol resulted in the impinging liquid on the sensor forming a thick film that did not evaporate or ‘wash’ off with the engine flow from cycle-to-cycle. This resulted in an effective thermal barrier, which significantly affected the sensitivity of the sensor at this temperature; the behaviour also resulted in no heat flux being produced during the valve overlap period, only low levels registered during injection and a slower signal response time observed in the expansion and exhaust strokes. One final feature of the signal observed for the alcohols only, was the behaviour of the signal after the main impingement heat flux peak. This behaviour was clearly shown for Butanol at 90 °C and was thought to be related to the high boiling point of Butanol coupled with its relatively large latent heat of evaporation. Indeed a similar signal response was also observed for Ethanol at 50 °C. Overall, the signal behaviour of Butanol generally appeared to lag that of Ethanol by 30–40 °C, that is, Butanol’s heat flux signals at 50 °C and 90 °C were quite similar to those of Ethanol at 20 °C and 50 °C respectively.

Single Droplet Evaporation Model

In order to understand more about the heat flux signals and to examine how these were proportional to the ΔT of the impinging liquid and the sensor temperature, a multi-component transient single droplet evaporation model was used to simulate wet-bulb temperatures and droplet evaporation rates for all fuels at different initial states. The model uses a Redlich-Kwong equation of state based on UNIFAC method coefficients for each species identified on the gas chromatograph of each of the fuels. It also solves a transient single-component evaporation model of the type presented by Faeth [30] but represents multi-component behaviour by using property data that varies with fraction evaporated as well as temperature. During evaporation, the property data for the remaining fuel in the droplet is continually re-calculated by linear interpolation from values in lookup tables that give the properties as a function of temperature and fraction evaporated. The evaporation part of the model proceeds by first performing an adiabatic flash calculation to get the vapour pressure of the droplets down to the system pressure. From this point on, the transient heat and mass transfer models are solved simultaneously to follow the

progress of the evaporation. The model assumes good mixing and heat transfer within the liquid droplets so the composition and temperature are uniform. The droplet velocity has no influence on the temperature versus fraction evaporated curves but, like the droplet size, it has a considerable influence on the rate of evaporation and, consequently, the temperature versus time curve; the velocity declines with time due to drag on the droplet and this can be dealt with using a Reynolds number correlation for droplet drag [31].

The model was run with initial droplet sizes of 20–50 μm in diameter evaporating in a convective environment and a constant relative velocity of 50 m/s. Spray tip plume penetration velocities measured by spray image processing, as well as droplet velocities measured by Phase Doppler anemometry, were in the range 40–80 m/s, depending on conditions and fuel type [32], hence a 50 m/s relative velocity was a reasonable compromise. Droplet sizes measured at a distance 25 mm vertically from the injector tip and in the centreline of a single spray plume by phase Doppler were found to range between 10–35 μm in Sauter Mean Diameter (SMD) diameter at different fuel temperatures [32]. The model initial conditions were on the high side in terms of droplet sizes because it was taken into account that droplets would be larger closer to the injection nozzle whose inner diameter was 200 μm and outer diameter 500 μm (to avoid cocking effects; the reader is guided to [19] for a detailed analysis of the nozzle's geometry). However, the exact initial droplet size was not critical with respect to general trends between fuels, which was the main objective of this exercise. The model ambient temperature was maintained at 20 °C and only changes in ambient pressure and initial fuel temperature were investigated.

The model predicted different evaporation histories and surface temperatures for the different fuels; these are shown in **Figs. 13–14** with a time-scale 0–30 ms in order to frame the results within the true time scales available for evaporation during the intake and compression strokes at an engine speed of 1500 RPM (although the simulation did not model the change in ambient air temperature that would occur during compression and it did not adjust the relative velocity to lower values past a timescale relevant to the end of injection). The figures show data which are comparable with the engine injection conditions presented in the current paper, *i.e.* 0.5 bar ambient pressure and initial droplet temperatures of 20 °C and 90 °C and therefore can be related to the evaporation trends observed with spray imaging in the engine primarily during the intake stroke [24].

The data for droplet temperature versus time in all cases, showed that the droplet temperature fell very rapidly at the start of evaporation and that most of the evaporation process within the timescales relevant for DISI engines was dominated by the transient period, particularly for the multi-component fuels, **Figs. 16–17**. The rate of evaporation was very high in the early stages; evaporation was also initially faster with higher initial temperature, however, this effect was reduced by the later stages of evaporation. When the fuel was hot (90 °C), the amount that had to flash off to obtain a stable liquid was greater for the multi-component fuels than for *iso*-Octane and the alcohols. Also, the initial stages of evaporation were faster with the multi-component fuels. The reference behaviour for *iso*-Octane showed that when the droplet vapour pressure was below the ambient pressure, the droplet began from a completely unevaporated state and the initial temperature was equal to the given liquid temperature.

When the initial vapour pressure exceeded the ambient pressure, a flash calculation was invoked in the model and the initial condition was then a partially evaporated one at which the vapour pressure had just fallen to ambient. This behaviour was exhibited at 0.5 bar for 90 °C for *iso*-Octane. The impact of the initial flash evaporation and

transient stages for *iso*-Octane increased with initial droplet temperature, and was inversely proportional with ambient pressure. In the most extreme scenario (high temperatures, low gas pressures), the near-steady droplet temperature was reached by the time about 45% of the fuel had evaporated (**Figs. 16–17**) However, after this stage, the initial droplet temperature had no further influence on the final droplet temperature. The data for Ethanol showed similar features as for *iso*-Octane and their final steady droplet temperatures were similar, but for Ethanol was such temperature reached when less than 25% of the liquid had evaporated. The data for Butanol also behaved in the same way but the depression of the wet-bulb temperature below ambient was much smaller. The final steady temperature was also reached with significantly less fuel evaporated (<10%). The data for Gasoline showed that the flash boiling and initial transient regimes were similar in general character to those for *iso*-Octane but the presence of more volatile species in the Gasoline meant that the initial flash boiling stage occurred down to lower droplet temperatures and thus the initial wet-bulb (quasi-steady period) temperature was lower. Once the quasi-steady period was reached however, the temperature became independent of the initial droplet temperature and the wet-bulb temperature rose as the evaporation proceeded due to the progressive depletion of high volatility components in the Gasoline. By the end of the evaporation period there was very little difference between the wet bulb and ambient temperatures. The data for E10 was very similar to that of Gasoline but showed that less fuel had evaporated in reaching the wet-bulb temperature, ~25% versus ~30% for Gasoline.

The droplet evaporation data showed that it took between 25–30 ms to achieve 90% evaporation for *iso*-Octane depending on initial droplet temperature. Ethanol evaporated considerably more slowly than *iso*-Octane and required between 0.07–0.08 seconds for 90% evaporation. Although not shown here, the effects of initial droplet temperature were less accentuated for Ethanol because of the higher latent heat relative to specific heat and consequently, there was clearer separation of data at 0.5 and 1.0 bar. The data for Butanol showed that evaporation was the slowest in the time scales relevant for direct injection and that the influence of initial temperature was small so that the time for evaporation depended principally on the ambient pressure. The time for 90% evaporation was ~0.35 s at 0.5 bar and ~0.63 s at 1.0 bar. For Gasoline the influence of initial temperature was significant in the earlier stages of evaporation but smaller in the later stages. The time taken for 90 % evaporation was ~0.042 s at 0.5 bar and ~0.075 s at 1.0 bar, which was significantly slower than for *iso*-Octane. This was because of the influence of the less volatile components (~20%) shown by the reduction in evaporation rate in the later stages of evaporation. However, scrutiny of the early stages showed that evaporation was faster with Gasoline up to about 50% evaporated. The data for E10 were again very similar to those of Gasoline but indicated that E10 evaporated more gradually.

Sensor Location Sweeps

The ensemble averaged peak heat flux values and timings of impingement ASOI for full location sweeps around the cylinder bore are shown in **Fig. 15**. The results clearly indicated that the nominal spray symmetry had been destroyed by the flow. Specifically, plume #1, directed towards the intake valves, registered significantly lower levels of heat flux relative to plume #6 and overall absolute levels were much lower across the range of sensor locations indicating significantly lower levels of impingement over this region. Indeed, spray imaging showed that if the injection timing coincided with the strong valve jet at ~60–80° CA ATDC, the flow momentum was sufficiently strong to overcome the spray plume momentum and plume #1 in particular was significantly distorted

and broken up by this interaction, resulting in less spray arriving at the cylinder walls in the range -30° to $+10^\circ$ sensor locations. The fuel that arrived at the walls had been shifted in space in the direction of the intake flow, *i.e.* whereas tests within the quiescent environment of the engine in static mode the peak heat flux occurred at -20° location [33]; under motoring conditions peaks occurred at the -10° location (the reader may refer again to **Fig. 10** for reference sensor locations and **Fig. 1** for spray plume numbering convention).

Similarly, the intake valve flow deflected plume #6 towards the exhaust side and shifted the main impingement locations from around the 25° – 45° range at quiescent conditions to the 30° – 50° range under motoring conditions. The magnitudes of the heat flux measured either side of the peak values was also higher under motoring conditions, indicating that there was a wider spread of impingement locations on the cylinder liner, whereas for static conditions the locations of heat flux were more concentrated around the plume axis, reflecting the compactness of the plumes under these conditions.

There were also several other interesting features of the results presented in **Fig. 15**. The first related to the cyclic variability of heat flux levels in certain locations. For example, the heat flux for plume #1 (around locations -30° to $+10^\circ$) showed significantly higher levels of COV than for the locations of impingement of plume #6, by nearly two times for some locations. This demonstrated the much greater effect of the valve jet on plume #1 which to some extent also shielded plume #6 from more direct interaction, thus making impingement more repeatable on average for plume #6; this also occurred due to the effect of the plumes' initial trajectories, where plume #1 was directed into the flow while plume #6 was directed in the same direction as the valve flow. The stronger flow interactions for spray plume #1 therefore made impingement levels more sensitive to cyclic variations in the intake flow. The variation in the flow velocity magnitude in the intake runner, during the early period of the intake stroke, was measured using high-speed crank-angle resolved PIV by Justham *et al.* [34] in an identical engine at the same operating point and reported to be relatively small, with values of $\sim 10\%$ of the instantaneous velocity magnitude ~ 30 m/s. These researchers reported that, overall, small cyclic variations existed in the runner flow during the intake phase, primarily represented by small variations in the flow direction. The higher variation in impingement levels on the intake side of the combustion chamber were therefore likely to be influenced by such flow variations over the intake valves, as well as be affected by the randomness of plume break-up from cycle-to-cycle and this was reflected in the impingement patterns. These trends were generally replicated for all fuels, supporting the view that these were flow-field related effects and largely independent of fuel type.

At 90°C engine temperature, the measured changes in the heat flux traces were also able to depict changes in the characteristics of spray formation. For *iso*-Octane, Gasoline and E10, the heat flux no longer exhibited two distinct peaks, rather only a single peak with a maximum at the $+30^\circ$ sensor location. For Gasoline and E10 this change was believed to result from partial collapse of the plumes #1 and #6 due to flash-boiling that caused them to be drawn towards each other as they developed, eventually merging into a single cluster of droplets and ligaments. This clearly affected their directionality towards the centreline of the chamber; with high fuel temperatures increasing the evaporation rates around the spray, only a single central region of droplets reached the wall. For *iso*-Octane, although spray collapse did not occur [25, 26], the break-up and evaporation of the spray was shown to be much greater and faster, which resulted in a similar pattern of impingement. However, even at these low levels of heat flux there were differences in the distribution pattern which indicated different types of

spray development; for example the *iso*-Octane distribution was clearly not symmetrical about the peak location as compared to Gasoline and E10 fuels and this was believed to reflect the fact that *iso*-Octane's plume #1 was completely broken-up and evaporated by the flow so that at the +10° and +20° locations, the heat flux was lower than for these fuels (indeed effectively zero for the +10° location).

The single peak profile of impingement was not observed for Ethanol and Butanol even at the highest temperatures, suggesting that these exhibited more intact and un-collapsed spray plumes. This was not a surprise for Butanol where flash boiling was not expected due to its high boiling point. In contrast, Ethanol has been observed to collapse more than *iso*-Octane in earlier work by the current authors [24]; however Ethanol was also slower to evaporate than *iso*-Octane even at 90 °C in [24], hence the differences in impingement heat flux were believed to reflect both the poorer plume break-up and the slower evaporation of Ethanol (fortified by the droplet evaporation modelling results too).

The high latent heat of Ethanol once again also distorted the real picture when comparing the values of peak heat flux between fuels, because as was already shown, the evaporative cooling of Ethanol contributed significantly to the value of its peak heat flux value. If values were compared directly they suggested that the levels of impingement for Ethanol would be of the order 4 times greater than for *iso*-Octane. This seemed unlikely however, particularly when Butanol produced much lower peak heat flux values than Ethanol but was logically expected to produce higher levels of impingement based on its atomization and evaporation characteristics. Even accounting for *iso*-Octane's and Butanol's lower latent heats of evaporation relative to Ethanol (roughly a third for *iso*-Octane and one half for Butanol) the peak signal for Ethanol would still be roughly twice that of *iso*-Octane, *i.e.* it might be reasonable to speculate that more liquid reached the wall for Ethanol. Compared to Butanol however, by this account Ethanol would have only a marginally higher heat flux than Butanol and, considering the differences in their respective wet-bulb temperatures (higher for Butanol than for Ethanol), the hierarchy of impingement levels was predicted to be Butanol, Ethanol, *iso*-Octane, with Gasoline and E10 producing the lowest levels.

The ability of the heat flux sensor technique to compare quantitative differences in the liquid mass impinged for different fuels was therefore found to be limited, in the current configuration; however, as has been shown, it should be possible with further work to properly calibrate for such differences in the physical transport properties of the fuels, which distorted the heat flux signals away from a directly proportional heat flux *vs.* liquid mass relationship.

The particular heat flux trends discussed for 90 °C were supported by the timings of arrival of liquid at the sensor face, and their cyclic variation, shown in **Fig. 16**. While the timings of liquid impingement and its COV's at 90 °C were very similar for *iso*-Octane, Gasoline and E10, for Ethanol and Butanol there were clear differences, particularly for sensor locations +30°–90°. For the alcohol fuels, the time to impingement was faster and showed less variability than the other fuels, which was indicative of direct liquid impingement, rather than secondary impingement of atomised droplet clusters or splashing from the main impingement event; the latter process clearly took longer to register a heat flux and this was clear from locations +50° onwards for the first three fuels. The faster impingement at sensor locations -20° and -10° for Butanol also indicated more direct impingement in this region due to the more intact plumes and lower levels of break-up which reduced drag and plume travel times.

These results in conjunction with the heat flux location sweeps demonstrated that a quite detailed picture of the phenomena taking place inside the cylinder could be drawn from the heat flux signal data, if carefully interpreted.

Split Injection Strategies

The results in **Fig. 17** indicate that the triple injection strategy significantly reduced the measured levels of heat flux on the cylinder liner for all fuels. Experimental conditions were otherwise comparable to single injection cases and wet-bulb temperature effects were not considered to have a different effect on the results since a triple injection strategy was conducive to faster evaporation and therefore wet-bulb temperatures were expected to be reached in similar or lower times than for single injection. This left only mass effects as the major factor influencing the measured heat flux. Comparing heat fluxes between injection strategies for the same fuel, the peak values were reduced by ~0.5–1.0 order of magnitude (depending on the temperature condition) for *iso*-Octane, Gasoline and E10, with triple injection; for Ethanol and Butanol the differences were somewhat smaller, with reductions of ~0.3–0.8 for Ethanol and ~0.25–0.5 for Butanol.

The second indication of lower levels of impingement could be seen in the exhaust stroke around 480° CA ATDC. For *iso*-Octane there was not much difference in the signal in this region from single to triple injection, however for Gasoline and E10 the negative heat fluxes produced with single injection were observed to disappear altogether. Similarly for Ethanol, this was only seen at the lowest temperature of 20 °C where clearly some fuel still remained on the sensor face. For Butanol, at 50 °C no real difference in the heat flux value was observed compared to single injection case in this region and it was interesting to note that at this temperature, the impingement heat flux was also very similar for both injection strategies. This suggested that splitting injections to improve spray break-up on its own was insufficient with fuels of relatively high viscosity and boiling points such as Butanol. In such cases only heat addition combined with these strategies could improve the atomisation of such fuels, and this was reflected in the results obtained at 90 °C.

A comparison of the fuel specific effects on heat flux with triple injection also showed that the mean peaks for each injection event occurred within 1° CA for all fuels, highlighting the remarkable repeatability of the injector hardware even with these low pulse widths and the small effect of fuel type on liquid transport velocities to the cylinder walls. Compared to the single injection case shown in **Fig. 12** the evaporation trends based on the heat flux signal returning to equilibrium were the same for the fuels at 20 °C. Peak levels of impingement however differed, with Gasoline producing the lowest average heat flux, E10 and *iso*-Octane producing very similar levels but double the magnitudes of Gasoline and the highest being Ethanol; the levels for Butanol were nearly zero throughout due to the similar sensor and wet-bulb temperatures of impinging liquid.

Although it was not trivial to decouple the competing effects that defined the final heat flux signal, certain insights were possible. For example, Gasoline had the lowest values of heat flux, yet it would be expected to have the highest based on absolute temperature difference between the fuel and the sensor (due to its low wet-bulb temperature), suggesting that it was the much lower levels of mass impingement due to improved break-up and evaporation that produced its low values. E10 on the other hand should have had very similar temperature related effects compared to Gasoline so the differences were more likely to be related to slightly higher mass impingement or due to contribution from the latent heat of the 10% Ethanol. Similarly for *iso*-Octane its heat flux signal relative to Gasoline was less a function of absolute temperature differences and more a function of mass,

based on its wet-bulb temperature. Thus, Butanol, Ethanol, *iso*-Octane, E10 and Gasoline was the hierarchy of mass impingement based on the triple injection data obtained. These predicted trends were also consistent with the spray formation and break-up behaviour observed with the imaging studies.

At 50 °C the heat flux measured for Gasoline was practically zero; E10 and *iso*-Octane once again had very similar values which were lower than those at 20 °C. As seen with the single injection case, Butanol showed a clear negative plateau before the impingement event due to prior cycle fuel film cooling during the intake flow phase. Butanol's heat flux levels at 50 °C were similar to Ethanol's at 20 °C, while Ethanol produced much higher levels than any of the other fuels. For all fuels however, the peaks of all three injections were remarkably similar and suggested that changes in intake flow over the range 60°–100° CA did not significantly impact spray trajectories with the triple injection strategy when temperatures were below 50 °C. These conclusions appeared to be supported by results for 90 °C. This time heat flux was near zero for the multi-component fuels and *iso*-Octane as rapid atomisation quickly entrained the spray into the cylinder bulk flow before impingement could take place. For Ethanol, the second injection was significantly lower than the first or the last injection, which may have been due to transient intake flow effects. Similarly for Butanol, the first injection produced the highest heat flux while the second and third injections produced lower values.

Assessing the location sweeps for triple injection in **Fig. 18**, it was clear that a significant reduction in the absolute levels of heat flux was achieved for all locations. More importantly, there were near zero values of heat flux registered on the intake side of the cylinder locations as the split sprays were completely deflected by the valve jet and entrained into the bulk cylinder flow. The highest impingement levels were also skewed more towards the exhaust side of the cylinder than for single injection as a result of the reduced spray momentum and increased effect of the intake flow 'pushing' the sprays towards the walls. The COV's of the heat flux values also confirmed this, particularly on the intake side of the cylinder (locations -30° to +10°) where both levels of heat flux and COV were very low. The trends were observed for all fuels and even Ethanol's higher sensitivity to heat flux variations due to the effect of its high latent heat of evaporation showed near zero levels of heat flux on the intake side. At 90 °C there near zero levels of heat recorded in all sensor locations and for all fuels except the alcohols. Here the distribution of fuel followed the trends observed for lower temperatures and the differences in the spray plume trajectories between Ethanol and Butanol was clearly demonstrated. Ethanol heat flux levels while high, were skewed towards the exhaust side around sensor locations +50°–70° while Butanol impinged mainly around +40° location. Combined with the higher COV's for Ethanol these results indicated that the spray plume atomisation behaviour of Ethanol was quite different from that of Butanol and that its similar absolute peak values were a function of its high latent heat. Nonetheless, the significant impact of triple split injections in reducing liquid wall impingement was clearly highlighted for this operating condition.

SPRAY IMPINGEMENT HEAT FLUX IN FIRING ENGINE

In order to complete the characterization of the in-cylinder heat flux sensor technique, the engine was fired at low-load and 1500 RPM to investigate the effects of hot residual gases on the impingement levels measured. All the settings for motoring conditions were replicated. The injection pulsewidths however were adjusted to obtain stoichiometry and therefore were marginally higher than the 0.8 ms used for motoring conditions for all fuels.

Comparison of the heat flux signals obtained for firing conditions with those for low-load motoring at 90 °C, in **Fig. 19**, showed that under firing conditions, there was an initially positive heat flux recorded during the intake flow period ~40° CA ATDC for all fuels. This was believed to be the result of the hot residual gases flowing back into the cylinder due to the internal EGR produced by the current valve timing. In the injection period, peak heat fluxes were generally lower than at motoring conditions for the hydrocarbon fuels but higher for the alcohols. For the hydrocarbon fuels, this was believed to represent directly a reduction in liquid mass reaching the cylinder walls. The return to zero heat flux was also faster for firing conditions, as a result of higher local sensor surface temperatures; however, it was interesting to note that once this initial main evaporation had taken place, the heat flux dipped again in the same way that was observed for full load motoring conditions. It was therefore believed that this occurred due to evaporative cooling heat transfer effects from some unevaporated fuel remaining on the sensor face. It was interesting to note that this characteristic was not observed for the alcohol fuels, which showed a signal rapidly returning to thermal equilibrium after impingement. For Ethanol there was effectively no difference in the signal between firing and motoring conditions, except for the absolute peak values, which were marginally higher for firing conditions. While these peaks are not shown in **Fig. 19** for the alcohol fuels in order to retain the same axis scaling as the other fuels, these values can be seen later in graphs of fuel distribution around the cylinder bore (**Fig. 20**) and can be compared to their motoring equivalents (**Fig. 15**). Higher peak heat fluxes for the alcohols indicated that other factors were probably contributing to the heat flux beyond mass effects, such as the high latent heats of evaporation as highlighted in previous sections. Marginally higher sensor face temperatures under firing conditions could also have contributed to a higher temperature difference but this would also have affected the other fuels. For Butanol, fuel film evaporation was indeed shown to be faster for firing than for motoring conditions in **Fig. 19**, suggesting that the cylinder walls temperatures could be higher than expected. Indeed, when the sensor face temperature signal was interrogated during data analysis, it showed that sensor face temperatures were of the order or 10–20 °C higher under firing conditions. These differences would affect the signal profile due to changes in local conditions, *i.e.* the absolute values of heat flux. The differences would change the absolute temperature difference between liquid and sensor, and also affect phenomena such as film evaporation dynamics on the sensor surface. These issues highlighted some of the difficulties in performing such measurements with the accuracy obtained under motoring conditions and indicated that there were features of the experimental configuration that could be improved. Some suggestions for achieving this are provided before the end of this section.

The distribution of fuel around the cylinder for firing conditions is shown in **Fig. 20**. For 20 °C engine coolant temperature conditions, the general pattern of fuel impingement did not change relative to motoring conditions and levels of variability were also similar for all fuels. The magnitudes of the peak heat fluxes were generally higher for firing conditions than for motoring conditions at 20 °C because of hotter sensor temperatures, which increased the relative temperature difference with the impinging fuel. At 90 °C however, as was shown in the previous figure, the heat flux was lower for hydrocarbons and higher for the alcohols. Indeed the distribution for Ethanol can be seen to approach a unimodal function compared to motoring conditions where two distinct regions of impingement were observed. These effects were believed to result from the increased injector and gas temperatures due to firing operation affecting the spray formation and break-up in the same manner as for

gasoline, which experienced spray collapse. A limited set of data was taken for Butanol to check whether a defined heat flux peak was still observed, and indeed it was, indicating that a non-collapsed spray still existed.

The original aim of the work was concerned with investigating the ability of the heat flux sensors to map the physical in-cylinder events in terms of liquid level distribution and fluid dynamic characteristics of the individual fuel spray plumes on impact with the cylinder walls. These objectives required careful control of engine and sensor temperatures in order to facilitate the decoupling of competing effects, and these conditions were more difficult to achieve under firing conditions. One of the weaknesses of the configuration was the sub-optimal cooling system for the sensor body and steel ring section which housed the sensors. Unfortunately, the space constraints of the optical engine meant that it was not possible to design an integrated cooling system for the steel ring itself, where the sensors were located. The configuration was therefore somewhat compromised for sustained firing operation and was limited to very short runs of maximum 100 cycles at 1500 RPM or ~8 seconds. An improved sensor ring design to allow more controlled and detailed fired investigations should therefore clearly be an objective for future work.

SUMMARY AND CONCLUSIONS

This paper is concerned with the analysis of impingement of fuel sprays onto the cylinder liner of a direct-injection spark-ignition engine. Full-bore imaging using a fully optical Perspex[®] piston equipped with ‘fish-eye’ lens allowed direct visualization of regions of liquid impingement on the cylinder walls and was initially used to define a plane for measurements of wall heat flux. Heat flux sensors were positioned on this plane using a movable ring section that allowed full flexibility of sensor location around the cylinder bore. All tests were performed under motoring and firing engine conditions, for different fuels, injection strategies and engine temperatures. Gasoline, *iso*-Octane, Butanol, Ethanol and E10 fuels were used. The collected data and images were used to calculate mean and standard deviation statistics. The results were also discussed with reference to evaporation rates predicted by a single droplet model. The main conclusions from this work can be summarised as follows:

- The crank-angle resolved images revealed that impingement occurred on the cylinder walls in the form of a broken up spray consisting of a mixture of airborne droplet clusters and larger ligaments originating from the spray leading edge. The nature of the partially atomised sprays meant that these were subject to influence from the in-cylinder bulk and intake valve flows. Variability in the impingement characteristics was therefore linked to variability in both liquid break up and flow interactions on a cycle-to-cycle basis.
- Using split injections to break up the high spray momentum of a single injection was found to be a successful strategy for reducing the spray penetration and thus reducing impingement onto the engine’s liner. This partially decoupled the mixture preparation process from large-scale spray/flow interactions with the intake valve flow, enhanced atomisation and produced faster evaporation.
- Analysis of the signal characteristics indicated that the peak heat flux signal due to impingement was related to the relative liquid impact temperature; tests for all fuels showed that this liquid temperature agreed well with calculated droplet wet bulb temperatures, *i.e.* droplets had reached their steady-state evaporation period by the time they reached the cylinder walls.

- The latent heat of evaporation also influenced the signal produced. While this was not observed to be a major issue for and between fuels with relatively low or similar latent heats, *e.g.* *iso*-Octane and Gasoline, direct heat flux comparisons between these fuels and Ethanol for example, required caution, particularly for conditions conducive to fast evaporation due to the convective cooling associated with film evaporation *e.g.* hot fuel, hot sensor face conditions or high convective velocities around the sensor. Peak heat flux signals from such tests were found to be significantly larger than expected, based on wet-bulb temperatures and characteristics of spray formation obtained through imaging. Such signals were thus not believed to be directly related to liquid mass levels and further work is necessary to decouple these effects fully.
- The heat flux sensor was able to capture the effects of spray/intake flow interactions on the spray development, namely the preferential break-up of plume #1 and the deflection of plume #6 observed initially with direct imaging. This resulted in an asymmetric impingement pattern across the cylinder bore for all fuels. The signals also clearly captured the change in spray plume geometry (*i.e.* ‘spray collapse’) of plumes #1 and #6 for Gasoline, E10 and Ethanol and showed that the plumes of *iso*-Octane produced a very similar impingement pattern due to similarly rapid break-up and atomisation at low pressure, high temperature conditions (0.5 bar inlet pressure, 90 °C coolant temperature).
- The peak signal location was used to calculate the timing of fuel arrival at the cylinder wall and found to be of the order 15–20° CA ASOI depending on impingement location, defining the flight time of liquid from injector to the wall. Changes to the time of fuel arrival reflected differences in the type of impingement taking place around the bore, with faster times produced in locations of direct impingement *i.e.* regions of high liquid impingement probability. Additionally, the difference in timing of peak heat flux between the different fuels was a measure of the differences in rates of spray penetration as a result of different physical properties (mainly viscosity and surface tension), especially under ‘non-collapsed’ spray conditions.
- Tests with triple split injections under motoring conditions produced three smaller individual peak heat flux values that were ~1/3 of the values obtained for single injection as a direct result of the lower levels of impingement. Impingement on the intake side of the cylinder bore was effectively eliminated. Similarly, wide-open throttle operation also produced ~50% lower peak heat flux values compared to part-load conditions (for single injection strategies) as the strong cross flow past the valves was able to deflect plume #1 away from the cylinder walls and increase the break up of plume #6.
- It was possible to determine qualitatively whether fuel films produced from impingement had survived during the compression and expansion strokes, as the convective flow in the intake stroke of the next cycle was of sufficient magnitude to generate evaporative cooling of the remaining fuel film and thus produced a recordable heat flux. Fuel films that survived a full engine cycle were virtually eliminated on average for all fuels using triple injection, except for Ethanol at 20 °C and Butanol up to 50 °C engine temperatures. This contrasted with single injection for which all fuels produced surviving fuel wall films except for 90 °C condition.
- The distribution of fuel around the cylinder for firing conditions under the same engine load (0.5 bar inlet pressure) showed that for 20 °C engine coolant temperature conditions, the general pattern of fuel impingement did not change relative to motoring conditions and levels of variability were also similar for all fuels. The magnitudes of the peak heat fluxes were generally higher for firing conditions than for motoring conditions at 20

°C because of hotter sensor temperatures which increased the relative temperature difference with the impinging fuel.

- Under firing with 90 °C, there was initially positive heat flux during the intake flow period of ~40° CA ATDC for all fuels. This was believed to be the result of hot residual gases flowing back into the cylinder due to internal EGR by the current valve timing. In the injection period, peak heat fluxes were generally lower than at motoring conditions for the hydrocarbons, but higher for the alcohols. For the hydrocarbons, this was believed to represent directly a reduction in liquid mass reaching the cylinder walls. The return to zero heat flux was also faster for firing conditions as a result of higher local sensor surface temperatures; however, it was interesting to note that once this initial main evaporation had taken place, the heat flux dipped again. It was therefore believed that this occurred due to evaporative cooling heat transfer effects from some unevaporated fuel remaining on the sensor face. This characteristic was not observed for the alcohol fuels which showed a signal returning rapidly to thermal equilibrium after impingement.
- The higher peak heat fluxes for the alcohols indicated that other factors were probably contributing to the heat flux beyond mass effects, such as the high latent heats of evaporation. For Butanol, fuel film evaporation was shown to be faster for firing than for motoring conditions. When the sensor face temperature signal was interrogated during data analysis, it showed that sensor face temperatures were of the order or 10–20 °C higher under firing conditions.
- Analysis of the distribution of fuel around the cylinder at firing conditions and 90 °C showed that the distribution for Ethanol approached a unimodal function compared to motoring conditions where two distinct regions of impingement were observed. These effects were believed to result from the increased injector and gas temperatures due to firing operation affecting the spray formation and break-up in the same manner as for gasoline, which experienced spray collapse. For Butanol a defined heat flux peak was still observed, indicating that a non-collapsed spray still existed.

Current work is focused on analysis of impaction of single fuel droplets of known size onto the heat flux sensor's array to gain insights for decoupling the temperature-related and mass-related response of the sensor. Further analysis of heat flux signals is carried out at quiescent conditions (*i.e.* with the engine 'static' at 1.0 bar ambient pressure) and from the engine running at full load (*i.e.* with 1.0 bar intake pressure). The diagnostic technique can be extended to 3-D transient characterisation of the cylinder liner and piston crown surfaces using multiple sensor locations sampled simultaneously.

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Intake

Intake

Exhaust

Exhaust

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Single Injection

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Single Injection Strategy

Split-Injection Strategy

2nd Injection

3rd Injection

91° CA ATDC

75° CA ATDC

85° CA ATDC

93° CA

78° CA

88° CA

95° CA

80° CA

90° CA

98° CA

83° CA

93° CA

100° CA

This space is left intentionally blank

95° CA

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iso-Octane

Gasoline

E10

Ethanol

Butanol

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iso-Octane

Gasoline

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Ethanol

Butanol

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Butanol

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iso-Octane

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Butanol

Ethanol

Fig. 19. Fuel Effect on Motoring and Firing Mean Heat Flux: All Fuels, $T_f = 20$ °C, 0.5 bar Inlet Pressure.

iso-Octane

Gasoline

E10

Ethanol

Butanol

Fig. 20. Peak Heat Flux at Various Peripheral Locations for $T_f = 20\text{ °C}$ & 90 °C , 0.5 bar Inlet Pressure (ensemble averaged values in blue, COV in grey).