Optical investigations of the sprays generated by gasoline multi-hole injectors under novel operating conditions

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Optical Investigations of the Sprays Generated by Gasoline Multi-Hole Injectors under Novel Operating Conditions

by

Andrew Wood

Doctoral Thesis
Submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of Loughborough University
March 2014
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Abstract

Political, environmental and marketing factors mean there is a global requirement to produce vehicles with improved fuel economy and reduced emissions. This thesis shows that the gasoline direct injection (GDI) engine will continue to form a significant portion of the automotive propulsion market in the short to medium term. However, to reach future targets continuous development and optimisation of these engines is essential. The introduction to this thesis discusses the role some of the key aspects of GDI engine design have on overall engine efficiency. The fuel spray is shown to be a key contributor to this, as it is a primary driver in the fuel/air mixing process, and therefore intrinsically linked to the combustion efficiency.

Multi-hole injectors are a cost effective method of introducing fuel into the cylinder for GDI applications and the characteristics of these injectors are discussed. These injectors are solenoid actuated, with the solenoid lifting a single needle allowing high pressure fuel to pass through a number of small orifices into the engine cylinder. The behaviour of these injectors and their sprays under certain injection conditions is as of yet not fully understood and the work of this thesis will examine two of these conditions.

The first of these is the use of split injection strategies, which has become a possibility due to the improved response times of the latest generation of solenoid driven injectors. Splitting the fuel mass into two or more separate injection events has the potential to cause a reduction in spray penetration, which in turn should correlate with a reduction in the levels of hydrocarbon emissions which occur due to wall and piston fuel impacts and the subsequent pool fires they cause.

The second topic is the behaviour of multi-hole GDI injector sprays under flash boiling conditions. During certain stages of the GDI engine operating map the combination of low in-cylinder pressure at the time of injection and heated fuel due to high cylinder head temperatures can result in flash boiling of fuel.

The split injection study uses data acquired with two different generations of Continental multi-hole injectors. It is shown that the behaviour of the injector, in terms of needle opening, is not always consistent between subsequent injections of a multiple injection strategy. If the injector has a needle bounce after closing it is imperative to account for this on all subsequent injections as it has been shown to interfere with the repeatable opening of the injector. The results show that with a short enough dwell time, the gap between subsequent injections, the second injection is entrained by the wake of the first injection and therefore propagates downstream at a higher rate. However, there is still a definite reduction in penetration with split injection strategies in comparison to a single injection with the same total fuel mass delivery, with the reduction being between 5 and 10% for the pulse width and dwell time configurations investigated.

At the end of an injection there are eddy currents remaining in the solenoid. With short dwell times these eddy currents have not dissipated by the time the second injection begins. This leads to a quicker injector opening transient and therefore, for the same electronic injection duration, it was
shown to produce a higher total fuel mass delivery. At the start of any injection there are many large droplets produced in the spray tip. During the needle opening phase the needle seat gap has a throttling effect on the fuel delivery to the nozzle holes, thus reducing the fuel pressure within the nozzle holes. However, due to the faster opening of the needle for the second and subsequent injections with split injection strategies, the number of these large droplets occurring is shown to be greatly reduced. Near nozzle macro-imaging confirmed that with a short dwell time large droplets were not produced upon fuel exit from the nozzle holes. Instead, a steady state spray was achieved in a significantly shorter time than for a comparable single injection.

For the purpose of investigating flash boiling sprays a new spray chamber was designed, manufactured which featured optical windows orientated at 110° to each other. This allowed phase Doppler investigations of flashing sprays to be undertaken. This experimental analysis complimented an imaging study conducted in an already existing spray chamber which featured windows at 90° and 180° to each other. A number of interesting conclusions were drawn from this analysis. The overall shape of the spray was shown to change radically under superheated conditions. The original spray streams were shown to disintegrate within the first 10mm downstream of the nozzle tip and form a number of different spray footprints which varied with fuel temperature and chamber air pressure. Among the geometric alterations which occurred after the break-down of the standard spray structure were interstitial streams, i.e. in the gaps between the original spray streams, and significant levels of spray collapsing. Interestingly points with equal superheat did not necessarily display identical spray shapes, most notably in the early phases of injection. The key role played by droplet entrainment into the formation of these new spray footprints was confirmed through phase Doppler analysis. As the droplets produced by flash boiling are very small, they have low momentum levels and therefore a high likelihood of entrainment. The combination of pressure gradients within the spray region and the large number of small droplets is believed to promote fuel entrainment.

Of particular interest in this study was the potential for flash boiling to produce a spray with a greatly reduced mean drop size than the equivalent non-superheated spray. The PDA data confirmed this was the case, with $D_{10}$ and $D_{32}$ values reduced by 32.5% and 42% respectively for a superheat degree of 36°C in comparison to a non-superheated case. The results presented in this thesis provide some interesting comparisons to published data. These show that with a 90° cone angle injector the superheat degree required for flare flashing, which is the complete degradation of the spray shape, is significantly higher than for 60° cone angle injectors, such as the injector characterised in this report. This confirms the importance of entrainment and stream to stream interaction in enacting the spray shape changes. A second test fuel, iso-octane, was compared to n-heptane which revealed similar levels of flash boiling for a similar superheat degree. This investigation by no means represents a complete study of flash boiling sprays, but shows some areas in which further research would be of great interest, most notably further exploration of the role of entrainment on the changing spray morphology under flash boiling conditions.
List of Publications


Acknowledgements

For helping this all come together over the past 3 years there are many people to thank and for many reasons.

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### Nomenclature

#### Acronyms

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>AESOI</td>
<td>After electronic start of injection</td>
</tr>
<tr>
<td>A/F or AFR</td>
<td>Air to fuel ratio</td>
</tr>
<tr>
<td>ASOI</td>
<td>After start of injection</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake specific fuel consumption</td>
</tr>
<tr>
<td>bTDC</td>
<td>Before top dead centre</td>
</tr>
<tr>
<td>CA</td>
<td>Crank angles</td>
</tr>
<tr>
<td>CAD</td>
<td>Crank angle degrees</td>
</tr>
<tr>
<td>CARS</td>
<td>Coherent anti-stokes Raman scattering</td>
</tr>
<tr>
<td>CCD</td>
<td>Charge coupled device</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>CNG</td>
<td>Compressed natural gas</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon monoxide</td>
</tr>
<tr>
<td>CO₂</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>CTL</td>
<td>Coal to liquid</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>FSN</td>
<td>Filter smoke number</td>
</tr>
<tr>
<td>GDI</td>
<td>Gasoline direct injection</td>
</tr>
<tr>
<td>GTL</td>
<td>Gas to liquid</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>ICE</td>
<td>Internal combustion engine</td>
</tr>
<tr>
<td>IRES</td>
<td>Infrared extinction/scattering</td>
</tr>
<tr>
<td>L/D</td>
<td>Length to diameter ratio</td>
</tr>
<tr>
<td>LAS</td>
<td>Laser absorption spectrometry</td>
</tr>
<tr>
<td>LD/LDM</td>
<td>Laser diffraction meter</td>
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<tr>
<td>LDA</td>
<td>Laser Doppler anemometry</td>
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<tr>
<td>LIF</td>
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<td>Laser induced incandescence</td>
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<td>Liquefied petroleum gas</td>
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<td>Laser sheet drop sizing</td>
</tr>
<tr>
<td>LSD</td>
<td>Laser sheet drop sizing</td>
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<tr>
<td>NOₓ</td>
<td>Nitrous oxides</td>
</tr>
<tr>
<td>OECD</td>
<td>Organisation for Economic Co-operation and Development</td>
</tr>
<tr>
<td>PCCI</td>
<td>Premixed charge compression ignition</td>
</tr>
<tr>
<td>PDA</td>
<td>Phase Doppler Anemometry</td>
</tr>
<tr>
<td>PEM</td>
<td>Polymer electrolyte membrane</td>
</tr>
<tr>
<td>PFI</td>
<td>Port fuel injection</td>
</tr>
<tr>
<td>PIV</td>
<td>Particle image velocimetry</td>
</tr>
<tr>
<td>PLI(E)F</td>
<td>Planar laser induced (exciplex) fluorescence</td>
</tr>
<tr>
<td>PM</td>
<td>Particulate matter</td>
</tr>
<tr>
<td>ppm</td>
<td>Parts per million</td>
</tr>
<tr>
<td>SD</td>
<td>Superheat degree</td>
</tr>
<tr>
<td>SDI</td>
<td>Solenoid direct injection</td>
</tr>
<tr>
<td>SLIPI</td>
<td>Structured laser illumination for planar imaging</td>
</tr>
</tbody>
</table>
### Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>Speed of sound in fluid</td>
<td>m/s</td>
</tr>
<tr>
<td>A&lt;sub&gt;t&lt;/sub&gt;</td>
<td>Rate tube cross sectional area</td>
<td>mm&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>C&lt;sub&gt;D&lt;/sub&gt;</td>
<td>Coefficient of discharge</td>
<td></td>
</tr>
<tr>
<td>C&lt;sub&gt;p&lt;/sub&gt;</td>
<td>Specific heat</td>
<td>J/gK</td>
</tr>
<tr>
<td>d&lt;sub&gt;0&lt;/sub&gt;</td>
<td>Hole diameter</td>
<td>mm</td>
</tr>
<tr>
<td>D&lt;sub&gt;10&lt;/sub&gt;</td>
<td>Arithmetic mean diameter</td>
<td>µm</td>
</tr>
<tr>
<td>D&lt;sub&gt;32&lt;/sub&gt;</td>
<td>Sauter mean diameter</td>
<td>µm</td>
</tr>
<tr>
<td>D&lt;sub&gt;v50&lt;/sub&gt;</td>
<td>Droplet diameter at which 50% of the spray volume is contained in droplets with a smaller diameter</td>
<td>µm</td>
</tr>
<tr>
<td>D&lt;sub&gt;v90&lt;/sub&gt;</td>
<td>Droplet diameter at which 90% of the spray volume is contained in droplets with a smaller diameter</td>
<td>µm</td>
</tr>
<tr>
<td>ΔG&lt;sub&gt;c&lt;/sub&gt;</td>
<td>Gibbs energy</td>
<td>kJ/mol</td>
</tr>
<tr>
<td>h&lt;sub&gt;fg&lt;/sub&gt;</td>
<td>Latent heat of vaporization</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>J</td>
<td>Rate of bubble growth</td>
<td></td>
</tr>
<tr>
<td>Ja</td>
<td>Jakob number</td>
<td></td>
</tr>
<tr>
<td>J&lt;sub&gt;0&lt;/sub&gt;</td>
<td>Initial constant for bubble growth</td>
<td></td>
</tr>
<tr>
<td>k&lt;sub&gt;B&lt;/sub&gt;</td>
<td>Boltzman constant</td>
<td>W/m&lt;sup&gt;2&lt;/sup&gt;K&lt;sup&gt;4&lt;/sup&gt;</td>
</tr>
<tr>
<td>L</td>
<td>Length</td>
<td>m</td>
</tr>
<tr>
<td>L&lt;sub&gt;b&lt;/sub&gt;</td>
<td>Break-up length</td>
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</tr>
<tr>
<td>Oh</td>
<td>Ohnesorge number</td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
<td>bar</td>
</tr>
<tr>
<td>P&lt;sub&gt;a&lt;/sub&gt;</td>
<td>Ambient pressure</td>
<td>bar</td>
</tr>
<tr>
<td>P&lt;sub&gt;a&lt;/sub&gt;/P&lt;sub&gt;s&lt;/sub&gt;</td>
<td>Ambient over saturation pressure</td>
<td></td>
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<tr>
<td>P&lt;sub&gt;f&lt;/sub&gt;</td>
<td>Fuel pressure</td>
<td>bar</td>
</tr>
<tr>
<td>P&lt;sub&gt;inj&lt;/sub&gt;</td>
<td>Injection pressure</td>
<td>bar</td>
</tr>
<tr>
<td>δP</td>
<td>Initial turbulence</td>
<td>kPa</td>
</tr>
<tr>
<td>q</td>
<td>Volume flow rate</td>
<td>mm&lt;sup&gt;3&lt;/sup&gt;/s</td>
</tr>
<tr>
<td>R</td>
<td>Radial distance</td>
<td>mm</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
<td></td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>°C or K</td>
</tr>
<tr>
<td>ΔT</td>
<td>Change in temperature</td>
<td>°C or K</td>
</tr>
<tr>
<td>t</td>
<td>Time</td>
<td>s</td>
</tr>
<tr>
<td>V</td>
<td>Volume</td>
<td>m&lt;sup&gt;3&lt;/sup&gt;</td>
</tr>
<tr>
<td>V</td>
<td>Velocity</td>
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<tr>
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<td>Units</td>
</tr>
<tr>
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<td>----------------------------------------------</td>
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<tr>
<td>$V_i$</td>
<td>Initial velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$We$</td>
<td>Weber number</td>
<td></td>
</tr>
<tr>
<td>$x$</td>
<td>Radial distance from injector axis</td>
<td>mm</td>
</tr>
<tr>
<td>$z$</td>
<td>Axial distance downstream of injector</td>
<td>mm</td>
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<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
<th>Units</th>
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</thead>
<tbody>
<tr>
<td>$\delta_0$</td>
<td>Initial axisymmetric turbulence</td>
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</tr>
<tr>
<td>$\varepsilon$</td>
<td>Void fraction</td>
<td></td>
</tr>
<tr>
<td>$\theta$</td>
<td>Spray angle</td>
<td>$^\circ$</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Air/fuel equivalence ration</td>
<td></td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic viscosity</td>
<td>Ns/m$^2$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>Kg/m$^3$</td>
</tr>
<tr>
<td>$\rho_l$</td>
<td>Density of liquid phase</td>
<td>Kg/m$^3$</td>
</tr>
<tr>
<td>$\rho_v$</td>
<td>Density of vapour phase</td>
<td>Kg/m$^3$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Surface tension</td>
<td>N/m</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Scattering angle</td>
<td>$^\circ$</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Fuel/air equivalence ratio</td>
<td></td>
</tr>
</tbody>
</table>
1 Introduction

There are currently a number of vehicle propulsion methods available and the first half of this chapter will introduce the current status and future prospects of these methods. The second half will focus on the gasoline direct injection (GDI) engine and explain the key parameters to the efficiency of these engines.

Current trends in automotive design are towards vehicles with increased fuel economy and reduced exhaust emissions (Heisler 1995), (Heywood 1988), (King 2007), (Pischinger 2006). There are many factors combining to force these changes. Firstly there are new laws which manufacturers must adhere to. These are in the form of increasingly stringent emissions legislations. The Euro III, Euro IV, Euro V and Euro VI regulations for carbon monoxide (CO), particulate matter (PM), total hydrocarbons (THC), non-methane hydrocarbons (NMHC) and nitrous oxides (NOx) are shown in Figure 1-1 (EU 2014). It can be seen here that the levels of these allowable emissions greatly reduced even over the last 10 years and further reductions are planned for the future. The most difficult of these emissions regulations to meet are the HC and PM regulations, the PM regulations being introduced only from Euro V onwards. It must also be noted that future emissions regulations may also include other harmful gases such as ozone and specific hydrocarbon pollutants, such as aldehydes (Gaffney 2009).

![Emissions legislations from 1992 onwards](image)

Governments are also keen to reduce the overall consumption of fossil fuels as there is only a limited accessible quantity of these fossil fuels remaining and consumption is continually increasing, as shown in Figure 1-2. Customers also desire vehicles with reduced fuel consumption and emissions for a number of reasons, such as road taxing and congestion.
charging based on vehicle emissions, but are often not willing to have any reduction in the performance of their vehicle.

![World Oil Consumption 1965-2012](image)

**Figure 1-2 - Global oil consumption (BP 2012)**

Many customers also desire vehicles with reduced emissions due to personal concerns about the issue of harmful gases in the environment leading to global warming. In a report by Nippon fuels, it is estimated that in Japan the transport sector accounts for 20.1% of CO₂ emissions (Yoshida 2011). There is little doubt the burning of carbon based fuels in many applications is leading to a change in the earth’s atmosphere, as shown by the increase in the worldwide levels of CO₂ in Figure 1-3, however, many sceptics remain unconvinced that this will lead to, or is currently leading to, climate change.

![Rising carbon dioxide levels](image)

**Figure 1-3 - Rising carbon dioxide levels (Betts 2013)**

The emissions issue is one which will only grow unless precautionary steps are taken as the number of vehicles worldwide is rising with increased wealth and development, particularly in Asia (Fulton 2011). The plot in Figure 1-4 shows this increase in passenger vehicles and the increase projected due to development, as particularly seen in China.
1.1 Current State of Various Vehicle Propulsion Methods

Although this work will discuss gasoline engine development in depth, it is essential to put the gasoline engine in context compared to alternative propulsion methods, both in terms of the current state and expected future developments.

1.1.1 Gasoline Engines

Gasoline fuelled internal combustion engines have dominated the global market in vehicle propulsion since the inception of the motor vehicle (Heywood 1988), (Zhao 1999). It is highly likely that the gasoline engine will continue to play a key role in vehicle propulsion for the foreseeable future for a number of reasons (Taylor 2008). Most importantly the gasoline engine uses fuel with a high power density which is ideal for vehicle propulsion. There is a large gasoline infrastructure over the whole globe, and a large majority of the public know and trust the gasoline engine (Pischinger 2006). The manufacturing cost for gasoline engines is relatively low and the manufacturing infrastructure is already in place (Cuenca 1999).

Wagner (Wagner 2013) analysed the potential advancements in gasoline engines and remarked that increased use of engine sensors and control will allow engine operation in areas which were previously considered unstable, thus facilitating further improvements in engine efficiency.

1.1.2 Diesel Engines

Like gasoline engines, diesel powered vehicles have been around since the inception of the motor vehicle. Traditionally diesel engines have been used in large vehicles, which require the high torque output afforded by the diesel engine. Originally the diesel engine was not common place in the passenger vehicle market as the initial production cost of diesel engine components is higher than that of a gasoline engine. But in recent times, in Europe in particular, the number of passenger cars powered by diesel engines has grown such that diesel engines have now overtaken gasoline engines as the most common propulsion method for passenger cars in Europe (Robertson 2006).
Diesel engine cars will remain commonplace, and it is expected in the near future will continue to grow in market share, until such a time that the infrastructure, expertise and public desire for alternatively powered vehicles can meet that of petrol and diesel powered vehicles.

### 1.1.3 Alternative Fuels

One alternative propulsion method which can make use of the extensive infrastructure, expertise and public perception already in place is the increased use of alternative and potentially renewable fuels for vehicle propulsion. Compressed natural gas (CNG) and liquefied petroleum gas (LPG) have long been available alongside gasoline and diesel fuels; however the potential to power vehicles with a whole new brand of fuels is an area of much promise.

Many alcohols can be utilised as fuel and in most countries, the gasoline fuel contains a certain amount of ethanol by law; 5% in the UK, 15% in the US and 85% in Brazil (Lagercrantz 2006), (Allen 2011). Brazil has long utilised its vast acreage of fertile land to grow crops to produce bio-ethanol (Pearson 2007). One key advantage of bio-ethanol fuel is that its production, crop growth, actually absorbs CO₂ which is produced through the burning of carbon based fuels, so the overall well-to-wheel CO₂ emissions of bio-ethanol fuel can potentially be negative. This is shown in Figure 1-5 (Kaji 2004).

![Japanese 10-15 test cycle](image)

**Figure 1-5 - Well to wheel emissions improvements from bio-fuels (Kaji 2004)**

However, there is the worry that because the potential monetary gains for farmers to grow biofuels vastly outweighs that of growing alternative crops, this could lead to food shortages and deforestation (Yoshida 2011). Many researchers, though, predict that if properly managed, it can be the unused land which is transferred for use in bio-fuel production, as shown in Figure 1-6 (Hoffman 2009). In addition, an alternative to crop based bio-fuels, is bio-fuels grown from algae, which do not compete with food crops for fertile land.
A further concern associated with alternative fuels is a new blend of engine pollutants for which catalytic converters are not optimised (Heywood 1988) and (Ramadhas 2011).

Hydrogen can be used as a fuel in internal combustion engines. This, however, does require certain alterations to the engine, particularly the fuel injectors due to the different density of hydrogen as opposed to gasoline, so the engine can only accept a single fuel. Hydrogen fuelled internal combustion engines (ICE’s) do, however, have a number of issues such as the effect of hydrogen on the engine lubricant and the requirement for physically larger fuel injectors due to the lower density of hydrogen (Verhelst 2011). As a consequence it is believed the potential benefits of hydrogen engines will not match those of hydrogen fuel cells, so little investment has been put into their development (Wallner 2009). However, it is possible that hydrogen engine vehicles will be used as a “bridge” to hydrogen fuel cell vehicles (Sawyer 2003), once the hydrogen infrastructure is in place, but the fuel cell vehicle has not yet become commonplace.

Shell believes that the bio-fuel market will continue to grow and eventually dominate the field of vehicle propulsion in the near term (Allen 2011). Expansion of the refuelling network is necessary to accommodate these new blends of fuels (Achtnicht 2012).

### 1.1.4 Battery Electric

The concept of battery electric vehicles is not new, but only in recent times with advancements in battery technology has this become a reality in the automotive sector. The battery electric motor uses a large stack of batteries combined with an electric motor which drives the wheels.

A limiting factor in these vehicles is the range of the battery pack before it is necessary to recharge it. Traditionally vehicles with battery powered motors have been designed purely for
users who will only be driving short journeys, such as to work and back, which allows time for
the battery to be recharged. However, while it takes only minutes to fill a fuel tank with
gasoline or diesel due to its higher energy density, it can take hours to recharge the battery pack
to its fully charged state (Jain 2011). With the recent advancements in battery technology, it is
feasible that the range of these vehicles can reach similar levels to those achieved by internal
combustion engine vehicles (Pearre 2011). Although these vehicles may have a high range when
new, the battery life, like in any rechargeable battery, will diminish over time at a rate
depending on the charge cycle (Neubauer 2012). Although the battery technologies available
have improved rapidly in recent years, the cost of these technologies is still high, of the order of
10 times that of an ICE, (Offer 2010), owing to the fact that many of the materials used in their
manufacture are rare.

The driving emissions of these vehicles are negligible, but the true emissions are dependent on
how the electricity which is used to charge the batteries is generated. In the UK over 75% of
electricity is produced through the burning of fossil fuels (Ma 2012), meaning the well to wheel
emissions are comparable to those for an ICE powered vehicle.

1.1.5 Fuel Cells

Fuel cell vehicles combine a fuel cell which produces electricity with an electric motor for
propulsion. The fuel cells in vehicles are most commonly Proton Exchange Membrane, also
known as Polymer Electrolyte Membrane, (PEM) fuel cells and combine hydrogen with oxygen
to form water and create electricity.

As the only product of the fuel cell is water, like battery power, this is a seemingly low emissions
solution. However, hydrogen is rare in the pure form which is required for these fuel cells and
must be produced by using electricity to split either methane or water. The source of this
electricity determines the well to wheel emissions of this type of propulsion system. Currently
95% of hydrogen is produced through the use of fossil fuels (Allen 2011).

The manufacture of the components of fuel cells is relatively expensive (Offer 2010) and
somewhat inhibitive to customers without major government incentives for purchase of these
vehicles.

Another issue which must be overcome if fuel cell vehicles are to become a major force in
passenger transport is that the infrastructure for running these vehicles is not currently in place
(Ogden 2011) and if this infrastructure is to be put in place it arguably must initially be
government funded (Allen 2011). Although there are a number of hydrogen refuelling stations
around the UK, mainly for research purposes, the first was not opened to the public until 2011
(H2O News 2011). It is believed that in limited markets hydrogen fuelled vehicles can be part of
a profitable market in the short term, but only with significant government investment (Yoshida
2011).
1.1.6 Hybridisation

Hybrid vehicles have been given a lot of press in recent years (Choi 2010) and due to this have attracted heavy investment. A hybrid vehicle refers to any vehicle where a primary mover, for example a gasoline engine, is supplemented by an alternative power source, such as a battery powered motor. Even with an engine and motor there are a number of configurations of hybrid vehicle.

Full hybrids can be split into two distinct groups: series and parallel hybrids. In a series hybrid the engine is connected to a generator which charges a battery pack. The battery power is used to drive a motor, which in turn powers the car. In parallel hybrids both the motor and engine are connected to the drivetrain. Most hybrids use regenerative breaking, where the kinetic energy of the vehicle is converted to electrical energy through the motor to charge the battery pack.

Mild hybrids use a single motor which replaces the starter motor and alternator and allows the engine to switch off when the vehicle is coasting, braking or stopped. Hence, they are useful for city driving where the engine is often idling in a conventional ICE car. This technology is often referred to as stop-start technology.

At present hybrid cars are still more expensive than conventional cars, due to the extra parts (motors and batteries). However they have successfully perforated the market, as they offer improved economy while driving without the drawbacks of reduced practicality of the alternative techniques such as fuel cells and electric vehicles (Lampton 2012). The reduction in emissions and fuel consumption available from hybridisation is of the order of 10-30% (Toyota 2012) and the implementation of hybrid systems into vehicle powertrains is likely to continue to grow (Jain 2011).

Figure 1-7 - Life cycle emissions of hybrid vehicles (Toyota 2012)
### 1.2 Comparisons and Outlook

#### Table 1-1 - Summary of Vehicle Propulsion Systems

<table>
<thead>
<tr>
<th>Pros</th>
<th>Cons</th>
</tr>
</thead>
</table>
| **Gasoline** | • Infrastructure, expertise and manufacturing already in place  
• Drivers know and trust the technology  
• Only a limited amount of oil remains  
• Production of a range of exhaust emissions |
| **Diesel** | • Infrastructure, expertise and manufacturing already in place  
• Drivers know and trust the technology  
• Only a limited amount of oil remains  
• Production of a range of exhaust emissions |
| **Battery** | • If electricity is produced from renewable sources the well to wheel carbon impact is negligible  
• No poisonous emissions  
• Current vehicles have a shorter range than engine powered vehicles, and recharging takes hours not minutes  
• Pedestrian safety concerns with removal of engine noise  
• Lack of engine noise may not appeal to some drivers  
• Extra costs to motorists |
| **Fuel Cell** | • If hydrogen is produced from renewable sources, the well to wheel carbon impact is negligible  
• No poisonous emissions  
• Hydrogen infrastructure requires implementation  
• Pedestrian safety concerns with removal of engine noise  
• Lack of engine noise may not appeal to some drivers  
• Extra costs to motorists |
| **Hybrid** | • Easy to implement with current technologies  
• Fuel and emissions savings  
• Driving experience is similar to traditional power methods  
• It is anticipated that future vehicles will predominantly show some form of hybridisation  
• Battery packs and motors are expensive  
• Extra cost to motorists |
| **Alternative fuels** | • Driving experience like that of gasoline or diesel  
• Changes to engines to implement these fuels are little or in most cases none  
• If produced from crops the well to wheel carbon footprint is negative  
• Production uses fertile land which could otherwise be used to grow food crops  
• Little or no improvement in tank to wheel emissions  
• Most filling stations only stock gasoline and diesel |

As can be seen from Table 1-1, although the potential long term gains of fuel cell vehicles may outweigh those of all other technologies, in the short to medium term the development of current vehicles to accept different cultivated fuels combined with the addition of hybridisation represents the next major step. However, to the consumer the cost and practicality of this is, in
a lot of cases not worth the potential gains, so for the next few years (10-40 depending on government funding and interference) the gasoline and diesel fuelled engines will continue to dominate the market place.

Many researchers have conducted reviews on the fuel consumption and emissions of various methods of vehicle propulsion.

A review into energy usage up to 2050 by the International Energy Agency projects that alternative propulsion methods will proliferate the market, Figure 1-8, but even in their most ambitious estimations gasoline engines will still be a major player in the worldwide automotive market (Fulton 2011) and in some estimations the usage of the gasoline engine is actually greater in 2050, than 2007.

This same review also looks at the cost of different hybrid and electric vehicle combinations and concluded that at the current time whilst the costs of alternative vehicles vastly outweigh the financial gains of improved economy to the consumer. But with technological improvements the increased vehicle cost could be offset by the reductions in fuel consumption. Based on life-cycle analysis some researchers have determined the cost to the customer and damage to the environment to be very similar between the ICE and hydrogen fuel cell options (both with hybridisation) (Schafer 2006).

These conclusions agree with those of Toyota, a global leader in hybrid powertrains (Tanaka 2011). Toyota see the long term future to be gradual implementation of hybrid and hybrid electric technologies, with fuel cell vehicles being the long term goal. However, they see that in the immediate future gasoline and diesel engines will dominate the market place, and short term advancements will be mainly in the broadening of fuel sources for these already developed engine types. This is because the benefits (in many senses) of the alternative technologies are so slight there is little desire for radical change (Ellinger 2001).
Fuel producers ExxonMobil reviewed the cost per unit mass of reduction in CO₂ emissions and found the cost of all potential technologies was far greater than for gradual improvements to gasoline engines (Tunison 2011). This explains the reluctance of government and automotive manufacturers to invest heavily in technologies which in the short term will show little reward.

One major stumbling block for alternative propulsion techniques to usurp ICE vehicles from the market is the initial cost to the customer of these techniques. However, it is projected that in the next 15-20 years, the cost of these vehicles may be comparable to the price of ICE vehicles (Weiss 2012). After this period it is anticipated, due to the shortages of fossil fuels, worldwide government pressure will lead to the implementation of the technologies and infrastructure for hydrogen fuel cell vehicles to become the norm (Tollefson 2010).

It is anticipated that the majority of future vehicles will show some form of hybridisation (Eppstein 2011) and in the short term, due to the political backing hybrid vehicles have received, and therefore extra funding, progressively more hybrid vehicles will be produced.

A number of researchers (King 2007), (Jain 2011) concluded that battery electric vehicles will never form a large part of the market, but in some cases could possibly dominate some small areas of the market (Endo 2011).

These reviews support the conclusion that the long term goal of powertrain solutions will revolve around fuel cell vehicles, but in the intervening 50-100 years while the hydrogen infrastructure does not exist there will be gradual changes to current powertrains, most significantly hybridisation and increased use of alternative fuels. However it is unlikely the gasoline and diesel engine will disappear in the near future.

1.3 Gasoline Direct Injection Engines

As discussed in the previous section it is widely anticipated that gasoline engines will continue to play a vital role in the marketplace and further research and development of these engines is essential to reduce their emissions and fuel consumption while maintaining driveability. This section will discuss gasoline direct injection (GDI) engines, which are widely regarded as a key player in the short to medium term, 50 years, in the area of vehicle propulsion.

1.3.1 Operating Principles

Under normal operating conditions a port fuel injection (PFI) engine, as shown in Figure 1-9 (left) (Zhao 1999), would have fuel injected into the intake manifold during the intake cycle to mix with the air charge during the intake and compression strokes before ignition by the spark plug at a time close to top dead centre (TDC). This means that the engine must always be run at close to stoichiometric conditions, otherwise the spark plug would not be able to ignite the air/fuel mixture within the cylinder (Heisler 1995), (Heywood 1988). This is a highly suitable strategy at high load conditions, where engine performance is the key parameter. However, at part and low load conditions, where fuel economy and low emissions are valued over performance, this strategy is not ideal and a method of running the engine in a lean burn mode would be preferable. To control the airflow into the cylinder to ensure close to stoichiometric conditions,
a throttle is used, with which large pumping losses are associated (Zhao 1999). Gasoline direct injection (GDI) engines offer one potential solution to this problem. The GDI engine operates in two different modes, which are summarised in Figure 1-10. The first GDI engine to operate in this two-stage mode was introduced by Mitsubishi in 1998 (Kamura 1998), (Ando 1997).

![Figure 1-9 – Port fuel injection (left) and gasoline direct injection (right) principles](image)

The first is at high load, where the fuel is injected directly into the cylinder during the intake stroke. In this mode the engine is run at close to stoichiometric conditions, such as a traditional PFI engine. Therefore careful design of the intake and fuel injection system is required as the mixing time is short compared to the PFI engine.

![Figure 1-10 - GDI operating modes](image)
The differences, however, come at part and low load operating points. At these operating points the engine can be run with an overall heterogeneous mixture within the combustion chamber. This is achieved by injecting the fuel into the cylinder towards the end of the compression stroke, such that it creates a fuel rich zone around the spark plug, which is easily ignitable, as shown in Figure 1-10 and Figure 1-12. This allows the engine to be run with a reduced level of throttling, which represents a major form of losses in engines. A further advantage of running with little or no throttling is that it allows excess air into the cylinder which increases the probability of the fuel being fully oxidised, therefore increasing the thermal efficiency of the engine and reducing the engine emissions.

Some GDI engines use a third mode which is a combination of the two modes previously detailed. Some fuel is injected into the cylinder during the intake stroke, but still in a highly lean state, and more fuel is injected at the end of the compression stroke (Landenfeld 2004), (Rivera 2010). The latter injected fuel is used to create favourable conditions for ignition at the spark plug, and the rest of the fuel is in a well-mixed state so will undergo highly efficient and fast combustion.

The fuel can be directed towards the spark plug for stratified charge operation in any of 3 ways as shown in Figure 1-12. These are wall guided, where the shape of the piston bowl directs the fuel back towards the spark plug at the time of ignition, air guided, where the in-cylinder charge air directs the fuel towards the spark plug, and spray guided, where the spray is directed such that is passes directly by the spark plug at the time of ignition.

![Figure 1-11 - Advantages of different operating modes for GDI engine](image)

It is essential that an ignitable mixture is formed around the spark plug, as if the mixture is too rich or too lean to combust this will lead to misfires (Peterson 2011). The first generation of GDI
engines used wall-guided systems for stratified charge operation (Kamura 1998), but due to emissions regulations limiting the levels of hydrocarbon emissions this technique is no longer applicable and the majority of modern GDI engines have spray guided systems (Yan 2007).

The advantages of the 2-operating mode GDI engine is summarised in Figure 1-11. This shows that even within the homogeneous operation region the engine operation is still speed and load specific. At moderate load fuel consumption is still the primary requirement, so the engine is run slightly lean with high levels of exhaust gas recirculation (which lowers combustion temperature, thus reducing production of certain pollutants, mainly NO$_x$). However, at higher loads the primary engine requirement is power, therefore the main goal of the combustion system is well mixed air and fuel, and GDI engines are often boosted to increase the amount of fuel and air which can be introduced to the cylinder while maintaining stoichiometry.

The advantages of the GDI engine over the PFI have been summarised in Table 1-2 (Zhao 1999):

<table>
<thead>
<tr>
<th>Table 1-2 - Advantages of GDI over PFI</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fuel Economy – up to 25%</strong></td>
</tr>
<tr>
<td>• Substantially reduced pumping loss</td>
</tr>
<tr>
<td>(if unthrottled during stratified-</td>
</tr>
<tr>
<td>charge mode)</td>
</tr>
<tr>
<td>• Reduced heat loss</td>
</tr>
<tr>
<td>• Higher compression ratios possible</td>
</tr>
<tr>
<td>(due to charge cooling by direct</td>
</tr>
<tr>
<td>injection)</td>
</tr>
<tr>
<td>• Lower octane requirement (due to</td>
</tr>
<tr>
<td>charge cooling)</td>
</tr>
<tr>
<td>• Increased volumetric efficiency</td>
</tr>
<tr>
<td>(due to charge cooling)</td>
</tr>
<tr>
<td>• Fuel can be cut off during</td>
</tr>
<tr>
<td>deceleration</td>
</tr>
<tr>
<td>• Reduced acceleration enrichment</td>
</tr>
<tr>
<td>required</td>
</tr>
<tr>
<td><strong>Driveability</strong></td>
</tr>
<tr>
<td>• Improved transient response</td>
</tr>
<tr>
<td>• Improved cold-start performance</td>
</tr>
<tr>
<td><strong>Air/Fuel Ratio Control</strong></td>
</tr>
<tr>
<td>• Less engine cycles before</td>
</tr>
<tr>
<td>stable combustion achieved</td>
</tr>
<tr>
<td>• Less enrichment required during</td>
</tr>
<tr>
<td>cold start</td>
</tr>
<tr>
<td>• Less acceleration enrichment</td>
</tr>
<tr>
<td><strong>Combustion Stability</strong></td>
</tr>
<tr>
<td>• Extended EGR limits (allows further</td>
</tr>
<tr>
<td>reduction of NO$_x$)</td>
</tr>
<tr>
<td><strong>Emissions</strong></td>
</tr>
<tr>
<td>• Reduced cold start HC</td>
</tr>
<tr>
<td>• Reduced HC peaks during engine</td>
</tr>
<tr>
<td>transients</td>
</tr>
<tr>
<td>• Reduced CO$_2$</td>
</tr>
<tr>
<td><strong>System Optimisation</strong></td>
</tr>
<tr>
<td>• Enhanced potential for system</td>
</tr>
<tr>
<td>optimisation</td>
</tr>
</tbody>
</table>

1.3.1.1 Homogeneous Combustion

A GDI engine operating in homogeneous mode has a relatively long time for fuel/air mixing compared to the stratified charge operation mode. The fuel is injected during the intake stroke, so the mixing time comprises of the rest of the intake stroke plus the compression stroke. Therefore the fuel/air mixing and subsequent combustion are governed by:

1. The ability of the fuel injector to deliver a finely atomised spray which is well dispersed around the cylinder without coming into contact with the cylinder liner and piston crown.
2. By the ability of the cylinder airflow to effectively distribute this fuel around the cylinder to form the most homogeneous mixture possible.

The overall combustion efficiency in homogeneous mode and the emissions levels are usually within allowed limits, but the overall efficiency in terms of power produced per unit mass of fuel is not as high as for stratified operation. Ideal injectors for homogeneous operation will spread the fuel as evenly as possible without impinging on the walls, so wide spray angle injectors are most desirable.

1.3.1.2 Stratified Charge Combustion

Under stratified charge operation the fuel is only injected into the cylinder towards the end of the compression stroke, so the quality of the fuel/air mixing is much lower than that of homogeneous operation. Although in the primary reaction zone there is a local rich mixture, which would be associated with inefficient combustion and high emissions, the excess air in the cylinder provides a secondary reaction zone.

As previously mentioned, modern day GDI engines mainly use spray guided systems for stratified operation. To avoid the formation of fuel rich zones away from the initial reaction zone, which may not undergo efficient combustion due to flame quenching as a result of the burned gases, it is desirable to use an injector with a small spray angle. As there is excess air in the cylinder during stratified charge operation, there is an increase in the formation of NOx pollutants (Sawyer 2003) due to locally high combustion temperatures and as previously mentioned this must be controlled with exhaust gas recirculation (EGR) to manage combustion temperatures.

An engine map which uses a combination of the stratified and homogeneous charge operation modes is without doubt the most efficient method from a fuel consumption point of view, while maintaining vehicle driveability. However, the stratified operation has issues from an emissions point of view due to the presence of local fuel rich zones which must be addressed. It is feared that with the next refinement of the Euro emissions regulations, the percentage of the drive cycle which can be completed in stratified charge mode will potentially have to be reduced due to the level of HC and PM emissions formed, due to wall wetting and locally rich combustion respectively.

![Diagram](image.png)

Figure 1-12 - Classifications of GDI engines in stratified operation
The advantages of the different methods of fuel targeting, which are shown in Figure 1-12, were summarised by (Zhao 1999) as shown in Table 1-3.

<table>
<thead>
<tr>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
</table>
| **Wall guided** | • Highly stable as the fuel guide is a solid object, so relatively non-sensitive to spray characteristics  
• Distance from injector to spark plug along the path fuel takes is long, so atomisation quality is relatively good | • Increased impingement on piston crown, leading to fuel “pools” and therefore high HC emissions  
• Shaped piston is detrimental to desired airflow for homogeneous charge operation  
• Requirements for injector position are different from for homogeneous charge, hence a trade-off is made |
| **Air guided** | • Path of fuel from injector to spark plug is relatively long, ensuring a good level of spray atomisation  
• Reduced wall wetting, hence reduced HC emissions | • Sensitive to the engine flow field, which is highly changeable from cycle to cycle  
• Therefore requires matching between airflow and fuel spray which is a highly dynamic relationship  
• Engine airflow is not consistent over the full range of engine speeds |
| **Spray guided** | • Little variation in spray position at the time of ignition, so allows the highest level of stratification  
• Little dependence on engine airflow  
• Reduced wall wetting, hence reduced HC emissions  
• Ideal injector position is similar to that for homogeneous operation | • Injector and spark plug must be positioned close to one another  
• Distance from injector to spark plug is low, hence large drops and fuel bundles still exist at time of ignition causing high soot emissions  
• Sensitive to spray characteristics  
• Potential for mixture to be too rich local to the spark plug causing misfires |

1.3.2 Fuel Spray

As previously mentioned the fuel spray from a GDI injector is vital to the overall combustion process. The fuel injector has the role of ensuring the fuel is metered precisely and is delivered into the cylinder at the correct position and at the right time, to ensure the fuel will be correctly distributed at spark timing.

An equation was produced (Payri 2011) which characterised the key factors for fuel/air mixing time. The key parameters in the coefficient were fuel and air characteristics along with both injector geometry and spray morphology, showing the essential role of the fuel injector towards efficient combustion.

A number of different types of fuel injector have suitable characteristics for use in GDI engines. The ideal fuel injector will be able to, in the homogeneous mode, produce a well distributed fuel
spray with minimal amounts of wall wetting, and in the stratified charge mode introduce a high volume of fuel into the cylinder in a relatively short space of time towards the end of the compression stroke. The three main types of injector which have been used in GDI engines are: the hollow-cone pressure swirl injector which was the original injector used for GDI engines (Kamura 1998), the multi-hole solenoid injector and the piezo injector.

1.3.2.1 Pressure Swirl Injectors

This was the original injector used for GDI engine applications by Mitsubishi (Kamura 1998). The fuel is introduced into the injector through tangential ports, such that within the injector body the fuel is swirling around the needle (Park 2002). When the needle is opened the fuel exits through a single circular orifice. The spray maintains the swirling motion imparted by the tangential ports and also has an axial flow velocity due to the pressure of the fuel inside the injector forcing it out. This configuration results in a hollow-cone fuel spray, with average droplet diameter of 10-20μm (Jang 1999) (Heather 2007).

1.3.2.2 Multi-hole Injectors

Pressure swirl injectors have been replaced in most applications by multi-hole injectors (Rotondi 2010) mainly due to the repeatability of their spray shape over a range of ambient pressure conditions. Particular focus will be given to these injectors as the work in this report is funded by Continental AG&G with the purpose of investigating the SDI (solenoid direct injection) injectors which they produce for advanced injection strategies. These are relatively simple devices in which high pressure fuel is held inside the injector and when the needle is opened the fuel exits through a number of holes, normally between 5 and 10. The fluid pressure is the key driver to force the fuel through the holes. These holes are usually 100-200μm in diameter. The needle is operated by a solenoid, such that to open the needle a current is supplied to the solenoid which then magnetically attracts the metal needle towards it, thus opening the injector. The needle is shut when the solenoid current is released and a spring forces the needle back to the closed position. This basic internal arrangement can be seen in Figure 1-13 and the resulting spray is shown in Figure 1-14. The average droplet diameter for multi-hole injectors is 5-6μm (Mojtabi 2010).

![Figure 1-13 - Multi-hole injector geometry](image)

1.3.2.3 Piezo Injectors

Piezo controlled injectors work on many of the same principles as solenoid driven injectors, however, the injector is operated by a piezoelectric stack. When a charge is applied to the piezo stack it expands and this causes the injector nozzle to open. When the charge is removed from
the piezo stack it returns to its normal size and the needle is closed by a spring. Piezo injectors in most cases are of the outwardly opening nozzle type, however, it is possible for the piezo stack to lift the needle, rather than push it down, to create an inwardly opening nozzle. The fuel spray from a standard piezo driven outwardly opening nozzle is shown in Figure 1-14. The spray is in a circumferential series of strings, and has a hollow-cone shape. Due to the cost of the piezoelectric cells these injectors are orders of magnitude more expansive than their solenoid driven counterparts so are only used in the luxury car market. Average droplet diameters with outwardly opening pintle piezo driven injectors are 3-4μm (Schmid 2010).

1.3.3 In-Cylinder Airflow

Another essential element to the fuel/air mixing process within an engine operating on the four stroke cycle is the in-cylinder charge air motion (Heywood 1988), (Heisler 1995). During homogeneous operation it is aimed to have the fuel evenly distributed throughout the cylinder prior to combustion. High levels of turbulence are vital to this process. This in-cylinder turbulence is caused through the breakdown of larger bulk flow motions through the turbulent energy cascade. Increased levels of turbulence have been shown to reduce cycle to cycle variation. For stratified operation, turbulence is also essential to ensure a short combustion event and also for the dilution of undesired combustion products.

Tumble and swirl motions are normally induced by the shape of inlet ports. The original GDI engine from Mitsubishi (Kamura 1998) featured vertical intake ports for increased tumble motion and most diesel engines feature either a tangential port or a helical port both of which have been shown to impart swirl motion on the flow.

1.3.3.1 Tumble

Tumble is the bulk flow of air in the vertical plane within the cylinder. This bulk fluid motion is caused as the inlet airflow follows the contours of the cylinder wall. As the piston approaches TDC the increased cylinder pressure causes the tumble motion to break down into progressively smaller turbulent flow structures. Most gasoline engines are designed with high levels of tumble as this force aids the air/fuel mixing process during the compression stroke.
It is assumed that a stronger tumble flow will lead to increased levels of turbulence at the time of combustion. However, this is not always the case. For example late inlet valve closing strategies can reduce the strength of the main tumble flow, but form other smaller flow motions. Hence the turbulent energy at the time of combustion is increased (Matsumoto A. 2011). Although the cycle to cycle variation of tumble flow is relatively low, it can be increased by the addition of a fuel spray directly into the tumble motion, as the spray motion will vary from injection to injection (Goryntsev 2011).

1.3.3.2 Swirl

Swirl is the rotational bulk motion around the cylinder axis. Swirl is often induced by helical or directed ports and similar to tumble flow is essential to the fuel/air mixing process. High levels of swirl are more often seen within diesel engine design as, unlike tumble, it does not break down as the piston approaches TDC. (Kim 2007) investigated the influence of tumble and swirl motion on a GDI fuel spray and found, for the engine configuration tested, the tumble to be highly consistent and the stronger of the two bulk motions. Swirl tends to be of a much lower strength, but show much more variability in its motion. For this reason it is suggested that where air-guiding is used for stratified operation a tumble dominated flow regime is used.

1.3.4 Pollutant Formation

As previously discussed the fuel injection process and the fuel/air mixing process are essential to providing a combustible mixture, but the true assessment of any engine will be the efficiency of the combustion process. The combustion event needs to quickly convert as much of the chemical potential energy stored in the fuel into heat energy to drive the piston down.

Efficient combustion is synonymous with a reduction in the formation of the majority of regulated pollutants. If the burning process were to be 100% efficient this would eliminate the presence of carbon monoxide in the exhaust and hydrocarbon and particulate emissions would be eradicated as all the hydrocarbon fuel would have been converted into CO$_2$ and water. However, combustion efficiency is unrelated to the production of some of the other exhaust emissions.
The emissions based on injection timing, more basically stratified or homogeneous charge operation, are shown in Figure 1-16. The highest emissions of CO, HC and PM all occur in stratified charge mode as they are due to poor mixing and spray/wall impacts. It must be noted that the correlations shown here are for a single 1st generation GDI engine, and the emissions characteristics of a modern GDI engine would be significantly different.

![Figure 1-16 - Emissions from a GDI engine (Maricq 1999)](image)

### 1.3.4.1 NOₓ

Nitrous oxides NOₓ are produced when the combustion temperature is high enough to split the nitrogen in the excess air, such that it will combine with the oxygen in the air. NOₓ refers to both NO and NO₂. The highest levels of NOₓ are produced in traditional SI engines at close to stoichiometric conditions. As NOₓ is only formed at relatively high temperatures, lean mixtures have fewer problems with NOₓ formation. The formation of NOₓ is a local procedure, which highlights the importance of efficient fuel/air mixing particularly for homogeneous operation.

One key method for control of NOₓ formation is the use of EGR, Exhaust Gas Recirculation. EGR is the addition of exhaust gases into the engine inlet. The exhaust gases contain burned gases, some fuel vapour and air. The primary purpose of the EGR is to control the combustion temperature and therefore reduce the formation of NOₓ pollutants. However, the reduction in combustion temperature also leads to a reduction in combustion rate and combustion stability.
1.3.4.2 CO
Partial reaction of the fuel in a shortage of air and with limited reaction time will lead to the formation of carbon monoxide, CO. Therefore CO production will only occur in significant quantities during homogeneous operation as the secondary reaction zone for stratified charge operation will provide excess air and convert the CO to CO₂. The level of CO at the exhaust exit is greatly reduced in comparison to the level present in the cylinder as the CO is continually being converted to CO₂ in the exhaust system.

1.3.4.3 HC
Hydrocarbon emissions are those of unburned or partially burned fuel. HC emissions covers a broad spectrum of compounds, some harmful and some inert. HC emissions occur where excessively rich areas of fuel are formed. The most significant formations of HC emissions occur where the fuel spray impacts the cylinder walls or piston crown (Soid 2011). This is problematic for GDI engines operating in stratified charge mode, as this inevitably will involve some spray impact onto the piston or cylinder walls.

For the next stage of Euro emissions legislations it is feared that the allowed levels of HC emissions will be so low, such that stratified charge operation will potentially not be feasible (Private Communication 2012). However, even a small level of HC will always be found in the exhaust gas for even 100% air/fuel mixing. This is because fuel will be absorbed into crevices in the cylinder and lubricating oil and not combust, but still released into the exhaust gas (Heywood 1988).

1.3.4.4 Soot, PM
Soot and particulate matter, PM, are formed where combustion takes place, with a shortage of air, such that particles of the hydrocarbon fuel have insufficient oxygen with which to react. For a GDI engine the most likely occurrence of this is for stratified charge operation, where there is a rich mixture in the centre of each spray plume, meaning only a partial burn will take place (Etheridge 2011). Due to the excess air in the cylinder for stratified charge operation, a large number of the soot particles formed are converted to CO₂ in the latter stages of the 4-stroke cycle.

1.4 Summary
It is apparent from the literature that vehicles will continue to be powered by gasoline and diesel engines for the near future and that the gasoline direct injection engine market will continue to grow. There are key areas in which the GDI engine can be made more efficient so as to meet future emissions requirements and one area in which significant improvements are both possible and necessary is the mixture preparation prior to combustion for both homogeneous and stratified charge modes.

Multi-hole solenoid driven injectors appear to be a suitable cost effective method of introducing fuel into the cylinder with the necessary control to ensure efficient and stable combustion. However, future emissions targets necessitate refinement and optimisation and for this a
greater understanding of the injection process under a variety of operating conditions and modes must be developed.

1.5 Industrial Impact of this Work

The work detailed in this thesis is partially funded by, and forms part of, a much larger project run by Continental Automotive AG&G known as the MAGIE project (Shi 2008). The project has financial backing from a number of sources including an EPSRC grant and the French government to conduct in-depth research into future injector technologies for reduced vehicle emissions. The project focuses on the latest multi-hole solenoid driven injectors from Continental. In addition to Loughborough, the project includes a wide variety of academic and industrial partners, mainly based in France and Germany such as CORIA at the University of Rouen, the French Institute of Petroleum research (IFP) and the Institute of Engineering Thermodynamics (LTT) at Erlangen University.

A vast array of research is included within the overall project, both experimental and numerical and the significant database of experimental results obtained is a key enabler for validation and development of current and future atomisation models, which are a vital part of the design process for future multi-hole injectors.

Loughborough’s role in this project is the investigation of injector and spray behaviour under a variety of novel operating conditions. On its own the experimental data is vital for enhanced understanding of the injectors, but the highly accurate phase Doppler data are also important to Continental as the drop size data are an integral part of successful spray models.

1.6 Objectives

It is evident that significant research is required into the GDI engine operation and this work will look at two novel operating conditions, split injections strategies and flash boiling sprays. The work will present a significant literature survey into these areas, before presenting a large volume of research. The main aims of the work are:

- Detailed analysis of the performance of multi-hole GDI injectors operating under split injection strategies
  - Asses the capability of the injectors to repeatedly deliver the correct fuel mass at the correct timing.
  - Determine the extent of the penetration reduction achievable through the use of split injection strategies.
  - Investigate any drop size penalties occurring with the continued opening and closing of an injector. The opening and closing periods are known to produce inefficient atomisation.

- Detailed analysis of the performance of multi-hole GDI injectors operating under flash boiling conditions
  - Design and build a pressure chamber in which PDA data on flashing sprays can be acquired.
- Assess the changing spray behaviour over a range of superheat conditions.
- Repeat previous experiments to confirm the injector under investigation does indeed possess the same spray characteristics as many other tested under flashing conditions.
- Determine the level of drop size reduction in flash boiling sprays.
- Build the knowledge level and suggest possible explanations for the changing morphology of multi-hole sprays under flashing conditions.
2 Spray Atomisation from Multi-Hole Injectors

As shown in the previous chapter the spray produced by the multi-hole injectors is one of contributors to overall engine efficiency and will play an important role in emissions control. This chapter will explain the spray break-up procedure and the factors which contribute to formation of a well atomised spray. The modern multi-hole GDI injector will be described in detail and the factors which affect its performance will be explained.

2.1 Spray Characterisation

To be able to understand injector performance it is essential to characterise the spray produced by the injectors. This is with respect to the general morphology of the spray both temporally and spatially, and the nature of the spray in terms of quality of mixing with the medium into which it is injected and the potential for that mixture to ignite in the desired manner.

2.1.1 General Spray Morphology Characterisation

The global position of the transient spray is of vital importance as it must be dispersed suitably throughout the cylinder, such that a combustible mixture is formed within the combustion chamber, or in the case of stratified mode that a region of combustible mixture is in the vicinity of the spark plug at the time of ignition. However, impingement on the cylinder walls and piston crown must be kept to a minimum, as this leads to formation of pools of fuel which do not fully combust and form unburned hydrocarbon emissions.

A number of basic parameters can be used to describe the general position of a spray.

Firstly, the penetration describes the distance from the nozzle exit to the instantaneous tip of the spray. Most often the axial penetration of an injector is quoted, which is the penetration along the injector axis. The radial penetration is also essential to describing an injector. The speed of the spray tip is described by the penetration rate.

The spray angle is the angle between the opposite extremities of a spray plume from the nozzle exit. For a multi-hole injector each individual spray plume would have an associated individual spray angle. Spray angle is sensitive to the measurement location, as it is not constant in the plume direction due to entrainment.

The spray axis is the centre of a particular spray plume. For regular pattern nozzles, this nominally should be the same as the axis of the nozzle hole, but due to cavitation changing the apparent nozzle angle (Mahklouf 2012) and the pressure drop in the centre of the plumes (van Romunde 2007) this is not actually the case.

The cone angle is the angle between the spray axis for two opposite spray plumes from a multi-hole injector (or the opposite sides of a hollow-cone spray). Many multi-hole injectors do not have an even number of symmetrical holes, so definition of the cone angle is not as simple as in Figure 2-1. Many injectors have an odd number of injection holes (for example 3 holes) such that the angle between the injector axis and each spray axis would be 45°, and this would be
classified as a half cone angle 45° injector, or sometimes still as having a 90° total cone angle even though there are no two plumes separated by such an angle.

Z represents the downstream distance from the injector tip and x the radial distance from the injector axis. Inboard and outboard refer to the side of an individual plume either closer, or further from the spray axis respectively.

**Figure 2-1 - Spray Parameter Definitions**

The spray footprint or patternation is the shape of the spray on a plane perpendicular to the injector axis. For example the spray produced by a piezo injector would have a ring shaped footprint, or wetted area.

**2.1.2 Drop Size Characterisation**

For efficient combustion it is essential that the spray is well atomised. The quality of atomisation is important as it is intrinsically linked to the mixing quality. Atomisation is also of vital importance to combustion as the combustion rate is linked to the ability of the fuel to evaporate and evaporation is linked to drop size (Lefebvre 1989). Therefore knowledge of the drop size is essential to spray characterisation.

There are many methods of drop size characterisation and they are summarised in Table 2-1 (Schick 1997). The bold elements are the drop size characterisations are those which will be heavily utilised throughout this thesis.

For the first 7 diameter methods listed in the table, with the form $D_{xy}$, the diameter is calculated as in Equation 2.1:

$$D_{xy} = \frac{\sum_{i=1}^{n} D^x}{\sum_{i=1}^{n} D^y}$$  

Equation 2.1
Where n is the number of droplets and D is the droplet diameter.

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Name</th>
<th>Description</th>
<th>Common uses</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_{10}$</td>
<td>Arithmetic mean droplet diameter</td>
<td></td>
<td>Calculation of evaporation rates</td>
</tr>
<tr>
<td>$D_{20}$ and $D_{21}$</td>
<td>Surface mean diameter</td>
<td></td>
<td>For surface controlled applications, such as absorption</td>
</tr>
<tr>
<td>$D_{30}$</td>
<td>Volume mean diameter</td>
<td></td>
<td>Best suited for volume controlling applications such as hydrology</td>
</tr>
<tr>
<td>$D_{31}$</td>
<td>Mean evaporative diameter</td>
<td></td>
<td>For evaporation and diffusion studies</td>
</tr>
<tr>
<td>$D_{32}$</td>
<td>Sauter mean diameter (SMD)</td>
<td>Volume to diameter ratio, includes information on both surface area and diameter; hence a low $D_{32}$ is important for a spray</td>
<td>Mass transfer rates for chemical reactions</td>
</tr>
<tr>
<td>$D_{43}$</td>
<td>Herdan diameter</td>
<td></td>
<td>Combustion studies</td>
</tr>
<tr>
<td>$D_{v_{10}}$</td>
<td></td>
<td>The droplet diameter at which 10% of the spray volume is contained in droplets with a smaller diameter</td>
<td></td>
</tr>
<tr>
<td>$D_{v_{50}}$</td>
<td>Volume median diameter</td>
<td>The droplet diameter at which 50% of the spray volume is contained in droplets with a smaller diameter</td>
<td></td>
</tr>
<tr>
<td>$D_{v_{90}}$</td>
<td></td>
<td>The droplet diameter at which 90% of the spray volume is contained in droplets with a smaller diameter</td>
<td>Highlighting of large droplets in a spray</td>
</tr>
</tbody>
</table>

At the combustion temperatures and pressures present in GDI engines, the smaller droplets will readily evaporate and are all fully combusted in a short space of time (Samuel 1994). However, the larger droplets, 30-100µm often do not fully combust due to insufficient reaction time and are the cause of a large portion of the engine emissions. This is partly because they have greater momentum, so are more likely to impinge on the cylinder wall or piston, but also because the reaction rates are not high enough due to their significantly higher mass to surface area ratios, so the fuel is not fully burned (Arters 1999).

For these reasons the $D_{32}$ and $D_{v_{90}}$ are increasingly important characteristics in GDI spray definition. At the present stage of GDI engine development, fuel injectors are being manufactured with emphasis on elimination of “large” drops, rather than reducing the overall droplet mean diameter. However, $D_{10}$ data will additionally be discussed as when using short
time bins to show temporally resolved data the overall number of droplets from which the averages are created is low (100’s as opposed to 1000’s). In this instance $D_{32}$ and $D_{v90}$ can be biased by a single large droplet and give potentially misleading information. One issue with $D_{10}$ information is that within large sample numbers a few large droplets will have little effect on the overall $D_{10}$ value, somewhat hiding the presence of these large drops.

### 2.2 Spray Break-Up and Atomisation

Spray break-up is driven by 2 major mechanisms:

1. Due to the instabilities imparted upon the flow in the nozzle.
2. Due to instabilities as a result of the high level of shear force between the fuel and the surrounding air.

It is essential to atomise the spray to create a disperse region with a large number of droplets, as this increases the overall surface area of the spray, and therefore the evaporation rate and combustibility of the spray. It has been shown that a smaller mean drop size will usually correlate to a reduction in the levels of most emissions product (Rink 1986). The correlation between reduced mean droplet diameter and increased number of droplets and overall surface area is shown in Figure 2-2.

![Figure 2-2 - Correlation between improved break-up and increased surface area (Zhao 1999)](image)
For the elevated ambient temperature conditions that occur in an engine, the drop size and, therefore in part, spray morphology is controlled by the evaporation rate (Zigan 2010 (a)) and (Zigan 2010 (b)). However, evaporation is equally controlled by the spray surface area, thus showing the important role of spray break-up on the formation of a combustible mixture within the engine. Comprehensive understanding of the spray break-up process is necessary for understanding of the processes which affect the combustion behaviour of any engine.

2.2.1 Shear Driven Break-Up

Upon exiting the fuel injector nozzle, the initial stages of the spray are often in the form of a thin liquid jet. The main part of the fuel break-up is due to aerodynamic forces between the high velocity fuel and the surrounding air. It has been shown that these aerodynamic forces are the key driver in jet break-up (Mansour 1994), as in the absence of these forces, a laminar and a turbulent jet exhibit the same break-up characteristics. The shear forces cause the formation of rapidly growing waves within the liquid and as these waves grow to a critical level, the liquid jet disintegrates into a series of fragments and further into a series of ligaments and droplets. The disintegration of a liquid sheet is shown in Figure 2-3.

![Figure 2-3 - Disintegration of a liquid sheet (Dombrowski 1963)](image)

Ohnesorge characterised the break-up process above in three separate periods and developed a dimensionless number (Ohnesorge 1936) to characterise this based on gravitational, inertial, surface tension and viscous forces (Lefebvre 1989). The relationships are shown in Equation 2.2.

\[
Oh = \frac{\mu}{\sqrt{\rho g l}} = \frac{\sqrt{We}}{Re} \sim \frac{viscous \ forces}{\sqrt{inertia \ast surface \ tension}}
\]  

Equation 2.2
We is the Weber number (Weber 1899), (Cohn 1960) and Re is the Reynolds number (Reynolds 1883) as in Equations 2.3 and 2.4:

\[ \text{We} = \frac{\rho v^2 l}{\sigma} \sim \text{interatial forces / surface tension} \]  
\[ \text{Re} = \frac{\rho v l}{\mu} \sim \text{inertial forces / viscous forces} \]

Where \( \mu \) is the dynamic viscosity in \( \text{kg/}(s\cdot \text{m}) \), \( \rho \) is the fluid density in \( \text{kg/m}^3 \), \( \sigma \) is the surface tension in \( \text{N/m} \), \( l \) is the characteristic length scale, usually the droplet or jet diameter in \( \text{m} \) and \( v \) is the velocity in \( \text{m/s} \).

(Lin 1998) determined that the break-up process for jets and droplets is dominated by surface tension forces at high Weber numbers. This causes a reduction in the jet or droplet diameter and therefore reduced Weber number. The break-up process will continue until the Weber number reaches a stable value. The onset of break-up will occur when aerodynamic drag is greater than the surface tension force and the critical Weber number for this is characterised by \( \text{We} = \frac{8}{C_D} \) (Lin 1998). This can be classified according to maximum stable jet or drop size \( D_{\text{max}} \), Equation 2.5, and critical relative velocity \( U_{\text{CRIT}} \), Equation 2.6.

\[ D_{\text{max}} = \frac{8 \sigma}{C_D \rho_A U^2} \]  
\[ U_{\text{CRIT}} = \left( \frac{8 \sigma}{C_D \rho_A D} \right)^{0.5} \]

In which \( C_D \) represents the drag coefficient, \( \sigma \) the surface tension in \( \text{N/m} \) and \( \rho_A \) the air density in \( \text{kg/m}^3 \).

For variations in operating conditions, in terms of fuel and air properties as well as the fuel and air relative motion, it was found there were four basic regimes of jet break-up (Reitz 1978) as summarised in Table 2-2 and Figure 2-4.

Rayleigh break-up is initiated by viscous forces as opposed to aerodynamic forces and effectively causes the jet to form a dripping motion. This occurs for liquid Weber number of above 8 and gas Weber number of below 0.4. Droplet diameters are much larger than the jet and are roughly two times the jet diameter according to (Rayleigh 1878).

First wind-induced break-up is the point at which aerodynamic drag has a greater effect on break-up than the viscous forces. The drag force also accentuates the oscillations generated by the viscous forces. This occurs for gas Weber numbers between 0.4 and 13. The droplets formed have a diameter of the order of the jet diameter.
<table>
<thead>
<tr>
<th>Rayleigh break-up</th>
<th>First wind-induced break-up</th>
<th>Second wind-induced break-up</th>
<th>Atomisation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{We}<em>{\text{gas}} &lt; 0.4$ and $\text{We}</em>{\text{liq}} &gt; 8$</td>
<td>$0.4 &lt; \text{We}_{\text{gas}} &lt; 13$</td>
<td>$13 &lt; \text{We}_{\text{gas}} &lt; 40.3$</td>
<td>$\text{We}_{\text{gas}} &gt; 40.3$</td>
</tr>
</tbody>
</table>

**Figure 2-4 - Ohnesorge break-up diagram (Reitz 1978)**
Second wind-induced break-up occurs when the aerodynamic forces further exceed the surface tension forces due to increased relative motion between the air and jet. The break-up of the jet now occurs much closer to the nozzle and the drops are of significantly smaller diameter than the jet diameter. The gas Weber number for these conditions is between 13 and 40.3.

The final stage of break-up is the atomisation regime, which is characterised by the complete disintegration of the jet. The onset of break-up is on, or very close to, the nozzle exit and leads to the formation of a fine spray of very small droplets. This occurs for gas Weber numbers of over 40.3.

It must be noted that the numbers quoted for each stage of break-up do not take into account any instabilities which may have been already imparted upon the flow inside the nozzle, which explains the fact that further research into this area has shown the same trends, but different transition criteria (Hudman 1994).

If the conditions at nozzle exit for heptane injection from a multi-hole injector are placed on the Oh/Re plot the result is clearly in the atomisation regime. A list of parameters is shown in Table 2-3. This leads to the formation of a cloud of droplets and ligaments very soon after the spray exits the nozzle for GDI injectors.

\[ \text{Oh} = \frac{\mu}{\sqrt{\rho \sigma l}} = 7.37 \times 10^{-3} \]
\[ \text{Re} = \frac{\rho vl}{\mu} = 5.316 \times 10^4 \]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial jet velocity</td>
<td>( v )</td>
<td>150 m/s</td>
</tr>
<tr>
<td>Nozzle diameter</td>
<td>( l )</td>
<td>200 μm</td>
</tr>
<tr>
<td>Fluid surface tension</td>
<td>( \sigma )</td>
<td>20.14 mN/m</td>
</tr>
<tr>
<td>Fluid density</td>
<td>( \rho )</td>
<td>684 kg/m³</td>
</tr>
<tr>
<td>Fluid dynamic viscosity</td>
<td>( \mu )</td>
<td>376 μPa.s</td>
</tr>
</tbody>
</table>

Conversely if the downstream conditions are used the results are as follows, with the assumptions in Table 2-4. This places the downstream conditions in either the 1\textsuperscript{st} or 2\textsuperscript{nd} wind-induced break-up modes.

\[ \text{Oh} = \frac{\mu}{\sqrt{\rho \sigma l}} = 3.3 \times 10^{-2} \]
\[ \text{Re} = \frac{\rho vl}{\mu} = 8.86 \times 10^2 \]
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Droplet velocity</td>
<td>(v)</td>
<td>50 m/s</td>
</tr>
<tr>
<td>Droplet diameter</td>
<td>(l)</td>
<td>10 μm</td>
</tr>
<tr>
<td>Fluid surface tension</td>
<td>(\sigma)</td>
<td>20.14 mN/m</td>
</tr>
<tr>
<td>Fluid density</td>
<td>(\rho)</td>
<td>684 kg/m(^3)</td>
</tr>
<tr>
<td>Fluid dynamic viscosity</td>
<td>(\mu)</td>
<td>376 μPa.s</td>
</tr>
</tbody>
</table>

(Hiroyasu 1985) characterised the jet break-up length as a function of jet velocity, as shown in Figure 2-5. The theory shows a number of jet break-up modes for low jet velocity before transition into the spray region, which can be characterised as similar to the atomisation modes shown by (Reitz 1978) and (Lin 1998). It must be noted the plot only shows the break-up of the continuous liquid jet, and is not a fair representation of the eventual atomisation quality of the jet, but in the spray region a shorter break-up length will correlate to improved atomisation.

![Figure 2-5 - Break-up of a standard diesel liquid jet (Hiroyasu 1985)](image)

Further to this basic correlation, (Hiroyasu 1982) also obtained measurements with an electrical sensor of the liquid break-up length for jets in different ambient pressure conditions, Figure 2-6. The results show the same general trend for each ambient pressure as the jet velocity is increased, but there is a significant reduction in the break-up length for increases in ambient pressure. This is due to the greater shear force imparted upon the spray by the surrounding air, and therefore increased initiation and growth of the surface instability waves which lead to break-up.
Mahoney and Sterling (Mahoney 1978) produced the most complete equation for prediction of the laminar jet length, Equation 2.7. The equation assumes an initially uniform jet. All parameters have their usual meaning with $\delta_0$ referring to the initial axisymmetric disturbance. The final term in this equation is an estimation based on a number of parameters and details can be found in the original paper.

$$L = d.We^{0.5}(1 + 3Oh)\ln\left(\frac{d}{2\delta_0}\right) / f(Oh, We)$$

Equation 2.7
The ability of this equation to predict break-up length is shown in Figure 2-7.

Once the jet has been broken up there will be mixture of droplets, ligaments and vaporised fuel. The liquid fuel continues to be broken down into progressively smaller units. The eventual final drop size is controlled by the secondary atomisation processes (Agrawal 2013). An example of droplet break-up mechanisms for increasing Weber number is shown in Figure 2-8 (Pilch 1987). There is a correlation between increased Weber number and reduced drop size; hence droplet velocity remains a key parameter in the downstream break-up process.

Figure 2-8 - Droplet break-up mechanisms

The break-up of droplets into progressively smaller units has been investigated (Arcoumanis 1996). It has been shown that instabilities on the surface of the droplets initiate the process,
with the frequency of these instabilities dependant on the relative shear force. As these instabilities grow the droplet break-up process continues to form secondary droplets and ligaments.

In the downstream region where a cloud of droplets exist, if the assumptions in Table 2-4 are again used, the Weber number is:

\[ We = \frac{\rho v^2 l}{\sigma} = 854.86 \]

Therefore at this point break-up is still occurring due to the relative motion between the fuel and the surrounding air. This break-up is only aided by the instabilities imparted upon the spray upstream; either in the nozzle or soon after nozzle exit and the transfer of these instabilities from primary to secondary break-up has been proven (Daidzic 1994). It is also noted that during the secondary break-up process the majority of droplets are not spherical (Samenfink 1994) and the droplet deformation process actually leads to the formation of a liquid sheet prior to break-up (Hwang 1996).

An experimental and theoretical study (Berthoumieu 1998) showed that the reality of a spray subject to interaction with surrounding air would not fit into the published models. This study showed break-up to be a continuous process and at various positions within the spray the time and length scales for the break-up process are continually varying.

**2.2.2 Instabilities from Inside the Injector**

It is clear that the fuel sprays from GDI injectors will be of the form of a fully atomised spray, due to the high Reynolds number at nozzle exit. Further break-up will occur downstream as the droplets and ligaments subsequently fracture into a number of smaller droplets, which will eventually vaporise at a rate determined by the fluid properties and the ambient conditions. However, the sharp angles inside the nozzle, the injector internal surface roughness and the sharp angle at nozzle exit mean that the spray exits the nozzle with instability already imparted upon it (Suh 2008). The importance of these instabilities has already been suggested in their interaction with shear driven instabilities and it is clear these instabilities are equally important when considering the overall spray break-up process.

Over recent years the effects of cavitation on spray break-up have been a common research theme in the sprays community. Cavitation refers to the formation of voids within the fuel injector where the fuel exists in a vapour state due to the reduced pressure in these areas. Cavitation is commonly found either in the nozzle hole or in the sac volume (a small volume between the needle and the nozzle hole opening inside an injector).

The effects of these instabilities can be thought of as the start of jet break-up (DeJuhasz 1931), and the instabilities caused by cavitation have been described as the primary factor in atomisation (Tamaki 1998).
(Bergwerk 1959) showed a key factor in cavitation to be the nozzle internal geometry. Some interesting results, shown in Figure 2-9, show the effect of the needle on the jet break-up under otherwise identical conditions. The images for the cavitating case show a liquid jet for a short distance downstream, before the jet breaks up. This shows the instabilities caused within the nozzle lead to enhanced break-up downstream when the surrounding air has caused an amplification of the instabilities.

![Figure 2-9 - Effect of needle on cavitation (Bergwerk 1959)](image)

Cavitation inside the injector has been shown to be a key factor to all aspects of the spray morphology upon exit of the nozzle (Sou 2007). This paper showed that the formation of ligaments on the spray periphery and the internal nozzle cavitation are interlinked. This happens in two forms: the increase in local turbulence caused directly by the cavitation and the collapse of regions of cavitation which were formed within the nozzle.

An investigation which used an enlarged acrylic nozzle for cavitation visualisation found that increased injection pressure led to an increase in the size of the cavitation bubble formed (Matsumoto M. 2011), and this increase in size of the cavitation bubble led to an increase in the spray angle of the atomised jet.

It has been shown that cavitation in the nozzle hole effectively blocks off a portion of the hole, thus changing the relative hole angle (Mahklouf 2012). It was found that the presence of cavitation causes a reduction in fuel velocity upon nozzle exit and that cavitation had no effect to decrease the drop size. It was even speculated that the absence of cavitation aided the break-up process due to the increased shear from the increased jet/droplet velocity.

Through use of CFD analysis it was shown that the spray break-up was controlled by the shear stress acting between the liquid spray and the surrounding air, the hydrodynamic instabilities at the gas/liquid interface and cavitation formation upstream of the nozzle exit (Ishimoto 2010). This study found that the length of the liquid core is inversely proportional to the formation of microcavitation inside the nozzle. The results in this paper were unable to determine the extent to which cavitation aids break-up as the droplet diameter profiles were not found to show any change based on the level of cavitation.

Further studies have examined the difference in droplet diameter caused by differences in cavitation (Suh 2008). This study used two diesel nozzles of the same diameter but different length, such that one nozzle exhibited more cavitation than the other. The study found that the nozzle which showed higher levels of cavitation, showed a reduction in $D_{32}$ of 1-2%.
The literature suggests all aspects of spray break-up are affected by instabilities imparted upon the fuel both externally and internally of the injector. An array of forces will all combine to form a series of break-up mechanisms to change fuel from a solid liquid mass to a well atomised combination of gas and liquid fuel. The processes are clearly very complicated, particularly due to the interaction of a number of complimentary forces upon fuel break-up, but some general trends can be clearly observed, such as cavitation aiding an earlier break-up process and increased shear force between the fuel and air leading to increased formation of instabilities and therefore increased break-up levels.

2.3 Multi-Hole Injector Spray Characteristics

Due to their low cost and efficient atomisation characteristics, the current injector of choice for the majority of GDI engines is the solenoid driven multi-hole injector (Smith 2011). This section will discuss the development process of this family of injectors and the key attributes for producing a well atomised spray in terms of injector geometry and supply fuel characteristics. Investigations have taken place into the advantages of using piezo driven injectors in place of solenoid injectors and while it is undoubted there is an improvement in atomisation and subsequent combustion (Payri 2011), these gains are small and do not outweigh the extra cost of the piezo injectors.

2.3.1 Injector Characteristics

As previously shown the nature of the spray upon exiting the injector is essential to the overall break-up process. Therefore of critical importance is the internal injector flow, mainly in terms of the formation of cavitation within the nozzle. For these reasons injector geometry has been studied at length on all types of fuel injector (Kushari 2010) and its effect on the spray break-up process is well documented. The nozzles used in this investigation are not of the step-nozzle type as depicted in Figure 2-10.
As previously discussed there are a number of geometric features on multi-hole injectors, Figure 2-10, which can be optimised. Optimisation of these parameters will lead to an optimised spray and the key design features of GDI multi-hole injectors will be discussed in this section.

The start and end of fuel delivery are invariably delayed in relation to the logic pulse which is sent to the driver unit which controls the injector, as in Figure 2-11 (Zhao 1999). When the injector command is sent to the injector the driver unit must convert a logic pulse into a current profile which will open the needle as quickly as possible. In this instance “quickly” refers to the time from the commencement of needle opening to the needle fully open state. This is because with partial needle lift the conditions inside the injector are not optimised for efficient atomisation. The injection rate is also subject to an additional delay due to the hydraulic delays as there is a relaxation time for the pressure conditions to reach a stable state after needle opening. The mechanical section of the needle closing time is due to the time required for the force from the solenoid, which holds the needle open, to be reduced sufficiently once the solenoid is no longer supplied with current, such that it is lower than the spring inside the injector. Needle opening and closing bounces are due to the force with which the needle hits its fully open or fully closed position. These are not desirable, particularly closing bounces, as they lead to fuel leakage in the form of large droplets which stay in the vicinity of the injector and can lead to injector fouling when combined with recirculating flows near the injector tip.

Figure 2-11 - Solenoid operation and relation to needle lift
2.3.1.1 Number of Holes and Hole Orientation

The number of nozzle holes and the orientation of these holes determines the spray distribution, and therefore optimisation of this is very much engine specific. The majority of injectors will feature 5-10 holes and the configurations can vary from a number of holes regularly spaced around the centre, with or without a central hole, to a horseshoe configuration in which a portion of the arc is left empty of fuel to allow air to be entrained into the spray core.

As previously stated the fuel injector must deliver a precisely metered volume of fuel into the cylinder in a short space of time. The fuel must be well distributed for homogenous operation, but directed towards the spark plug for stratified charge operation with a penetration that does not lead to an excessive level of piston or wall wetting. Obviously there are some conflicts in these design criteria, so compromises must be met, and these compromises are likely to evolve with future reductions in allowable emissions levels.

2.3.1.2 Orifice Hole Diameter and Nozzle Length to Diameter Ratio

The dimensions of the nozzle through which the fuel passes prior to entering the combustion chamber is clearly of great importance to the spray breakup process. The hole must be of a large enough diameter as to allow sufficient fuel to enter the cylinder in a short space of time, but small enough that the instabilities which aid breakup and spray dispersion which are caused by cavitation and wall effects are not confined only to the periphery of the spray.

The length to diameter ratio of the nozzle hole has been researched at length (Suh 2008), (Sou 2007), (Suh 2008), (Tamaki 1998). This is also a vital parameter to fully utilise the cavitation phenomenon to aid breakup. As the cavitation level will vary through the length of the nozzle it is essential to have a nozzle length in which the exit conditions are relatively stable. It must be noted that the range of L/D ratios tested by (Hiroyasu 1982) are significantly larger than those found in modern day GDI nozzles.

![Figure 2-12 - Effect of variations in L/D ratio at atmospheric pressure conditions (Hiroyasu 1982)](image-url)
Experiments (Hiroyasu 1982) focussed on varying the L/D ratio of nozzles at various conditions. At ambient pressure conditions, Figure 2-12, there are significant differences in the liquid break-up length for different L/D ratios. It can be seen that there is no clear correlation between L/D and break-up length.

However, at elevated pressure conditions, Figure 2-13, the break-up length curves for all L/D ratios follow the same trend. This suggests that at these increased ambient pressure conditions the jet break-up is dominated by the shear break-up between the fuel and surrounding air, whereas for the atmospheric tests the L/D ratio has an effect as the break-up is due to a combination of instabilities created in the nozzle and the shear break-up.

As the L/D ratio has a significant effect on the cavitation behaviour inside the nozzle, it is highly likely this in turn will be key to the initial spray angle formed. As proven by (Honda 2004), the initial spray angle has a great effect on spray dispersion and penetration, as shown from simulation results in Figure 2-14.

In further research (Honda 2004) the authors showed experimentally, Figure 2-15, that a reduction in L/D ratio led to an increase in spray angle and in turn enhanced spray distribution.
It was assumed the correlation between reduced L/D ratio and increased spray dispersion was due to increased effects of instabilities which were formed inside the nozzle.

![Figure 2-15 - Spray dispersion for different L/D ratios (Honda 2004)](image)

It was again shown with a different set of nozzles (Skogsberg 2005) that a reduction in L/D ratio is linked to an increase in spray angle, as in Figure 2-16.

![Figure 2-16 - Increased spray angle for reduction in L/D ratio (Skogsberg 2005)](image)

Clearly there is a correlation between L/D ratio and spray angle and therefore the overall spray morphology. However, there is no universal correlation between increased L/D ratio and either an increase or decrease in spray angle. The role of L/D ratio is to define the cavitation behaviour inside the injector which will in turn lead to the conditions at nozzle exit which can fall into a number of different modes as defined by (Sou 2013).

The effect of L/D ratio on drop size from the work of (Skogsberg 2005) is shown in Figure 2-17 which shows that for increased nozzle length there is a definite reduction in drop size. One possible explanation for this could be that for this particular nozzle the increased instabilities within the nozzle caused by the short nozzle length actually reduced the level of break-up, suggesting the majority of fuel break up is due to effects occurring outside of the nozzle.
2.3.2 Fuel Pressure Effects

Since the advent of GDI engines the fuel pressure used has been continually rising (Fraidl 1996). Increased fuel pressure aids the break-up process firstly as the speed differential between the fuel spray and the surrounding air will be greater, hence the shear break-up will be increased and secondly as the instability growth rate will be increased. Also of note is that the fuel delivery rate will be increased, thus allowing shorter injections to deliver the required fuel.

It seems logical to assume the greater initial fuel velocity would correspond to a significantly greater penetration, but in practice the overall spray penetration varies little with fuel pressure (Allocca 2013). They report an increase of around 10-20% in spray penetration for an increase in fuel delivery pressure from 50 to 100 bar for a given injected quantity of fuel. They conclude this is because some of the extra initial velocity is cancelled out by the increased shear force, which in turn causes the production of smaller droplets, which lose momentum at a quicker rate. The reduction in drop size has been observed by many researchers utilising different drop sizing techniques (Lucchini 2013) (Hammer 2011) (Sens 2011), and a further finding with increased fuel injection pressure is improved control of fuel dispersion and therefore improved homogeneity in the cylinder (Bruno 2003).

As expected there is a correlation between increased fuel pressure and reduced mean droplet size. Many researchers have shown this to be true and typical results are shown in Figure 2-18 (Abe 2013).
Furthermore to these conclusions on the correlation between fuel pressure and drop size, it was shown this drop size reduction was due to the increased initial fluid velocity at exit of the nozzle orifice which occurs with increased fuel pressure, Figure 2-19 (Abe 2013).
2.3.3 Fuel Properties

With the increasing use of alternative fuels it is essential to understand the effect of these different fuels on the fuel spray development. Differences in the spray development and fuel break-up process between different fuels and fuel blends will be determined by a number of fluid properties such as density, surface tension and viscosity. However, at engine relevant conditions, evaporation is promoted so that the heat of vaporisation and boiling point will inevitably become more dominant.

As a key part of the break-up process for GDI injectors is based on the formation of shear-induced surface waves on the liquid as it exits the injector, the fuel properties are one of the main parameters determining atomisation.

Much research has been undertaken into alternative fuels, but a large portion of this has been looking at the effect of varying fuel mixtures on engine emissions (Tartakovsky 2011), (Thewes 2011) with focus on the fuel chemistry, in terms of how it reacts with the surrounding air. This is vital to understand, but it is also critical to have a thorough understanding of how the different fuel properties affect the spray formation and break-up process.

Through investigations using a range of diesel/bio-diesel blends the effect of viscosity on the fuel spray and the subsequent emissions profile were investigated (Gostic 2010). In this study the cone angle was seen to change with fuel properties. The fuel density is increased with an increasing proportion of bio-diesel in the fuel and a non-linear change in cone angle was observed. This study also showed a rise in emissions which occurred due to small changes in cone angle. It was concluded that the cone angle change leads to formation of high temperature combustion zones in which thermal NOx is formed.

The fuel Weber number (effectively the fuels density/surface tension) is key to the break-up process and fuels with a reduced Weber number will exhibit increased penetration due to the momentum loss being lower as a result of reduced shear force against the ambient air (Zigan 2010 (a)).

The fuel blends can also have an effect on the spray angle. This is believed to be due to the different extent of the nozzle cavitation region for fuels with different kinematic viscosity and surface tension (Zigan 2010 (b)).

For bio-diesel blends it was found that the fuel viscosity and density had a major impact on atomisation and that the break-up was highly temperature dependent (Pandey 2012). The temperature dependence is due to the different volatility and therefore evaporation rate, and varying the proportion of different volatility blends in diesel fuel is shown to have a linear relation to the penetration depth, in tests at relevant engine conditions (Vogel 2010).

Differences can be observed in the break-up for different fuels which relate back to the fuels effect on the formation of cavitation in the nozzle hole. This has been observed for a number of fuels (Zigan 2010 (a)), (Zigan 2011) and the differences in cavitation formation for different fuels can lead to unpredictable flapping sprays at the start of injection (Zigan 2010 (b)).
Through investigation of a wide range of potential diesel fuel blends it was deduced that the break-up in the near field is controlled by the fuel surface tension, density ratio and viscosity, for both primary and secondary break-up, but further downstream the break-up, and therefore droplet size, is linked to the fuel evaporation characteristics (Reddemann 2010).

The penetration curves for a variety of single and multi-component fuels are shown in Figure 2-20 (van Romunde 2009). It is evident that under non-evaporating conditions there is little change in penetration between the different fuel blends, however, under evaporating conditions there were significant differences. Therefore the differences in fuel break-up and dispersion between different fuels will not be constant through the range of engine operation and under evaporating conditions the spray behaviour will be controlled by both the fluid mechanics of spray break-up and the physics of fuel evaporation.

![Figure 2-20 - Spray penetration for various fuel blends under atmospheric temperature and pressure (left) and 120°C and 0.5 bar (right)](image)

2.4 Summary

Multi-hole injectors clearly are capable of efficient fuel atomisation and repeatable spray control and hence are very suitable for GDI engines.

The high pressure injection through a small orifice with sharp angles on the inlet leads to the flow emanating from the nozzle being highly unstable. As this unstable jet penetrates through the air within the cylinder the shear force between the two amplifies the instabilities already present in the fuel and also creates additional instabilities. This leads to a very short atomisation length, in fact with some sprays exhibiting jet break-up prior to nozzle exit.

The influence of the key injection parameters have been studied at length. The number of holes and orientation of the holes is a very engine specific requirement. The nozzle hole length to diameter ratio has been shown to have a great influence on the spray in terms of break-up length and dispersion. A lower value of L/D leads to increased dispersion, but there is no clear relationship between L/D and break-up length scales. Fuel pressure has been shown to improve atomisation due to the fuel exiting the injector at a higher velocity, hence having higher shear force against the surrounding air. An interesting point is also the different spray performances of different fuel blends, particularly under evaporating conditions. In the short term it is likely
the range of fuel blends used in engines will only grow and future fuel injectors must be able to produce repeatable sprays with all types of fuel.

However, there are a number of operating conditions in which the spray behaviour has not been reported and is not fully understood, two of which are:

1. The use of split injection strategies in both homogeneous and stratified charge modes, as well as for a mixed mode
2. The flash boiling of sprays under low pressure and high temperature environments which will occur under heavily throttled conditions or with heated fuel.
3 Novel Developments in GDI Spray Research

Spray behaviour under certain engine operating conditions is not fully understood. This section will detail two such conditions and describe the work previously completed within the engine research field. These two areas will form the basis of the investigations undertaken for the completion of the thesis and the results will be detailed in depth.

3.1 Split Injection Strategies

With increasingly strict emissions regulations, gasoline combustion systems must become ever more efficient. As previously mentioned two of the main parameters of an ideal spray are a penetration which is of such a length that:

1. It penetrates sufficiently into the cylinder without excessive piston and wall wetting
2. A spray which allows the correct level of fuel and air mixing

One key method which shows the potential to deliver these improvements is the use of split or multiple injection strategies.

Split injection strategies involve having two or more short injections with the separation time between the individual injection events being known as the dwell time. Split injection strategies have long been used in diesel engine operation to control the heat release rate (Heywood 1988), (Park 2011) through the use of short pre and post injections.

However, for GDI applications the purpose of split injection strategies is very different. The limits on hydrocarbon (HC) emissions are being reduced and the main sources of these emissions are pool fires caused by piston and wall wetting (Abart 2010). To mitigate against this it is essential to develop strategies to control spray penetration. By splitting the injected fuel mass into a number of separate shorter injection events, the cumulative level of resistance between the fuel and air is increased. This is because after the first injection the subsequent spray tips are subject to resistance from the surrounding air, whereas for a single injection event the spray behind the tip is assisted by the entrainment effect of the fuel which has already passed through that region.

The other advantage of split injection strategies is the potential for improved homogenisation within the cylinder or for locally rich zones to be formed, for example around the spark plug (Rivera 2010). In theory, injection into the cylinder at various points within the main engine tumble could lead to improved fuel/air mixing and therefore optimal homogenisation and this could again be optimised across the engine speed range.

3.1.1 Penetration Reduction

As previously mentioned, there is a requirement for future GDI injectors and injection strategies to reduce spray penetration for the purpose of reducing piston and wall wetting, and therefore reduce the HC emissions caused by pool fires. With current solenoid technologies there is the potential to have a number of injection events during the intake phase of the combustion cycle.
(Oh 2011), (Stach 2007). The cumulative air resistance against a number of separate injection events will inevitably be greater than that against a single injection containing the same fuel mass, and hence lead to a reduction in penetration.

The differences in piston and wall impingement between single and double injection strategies were experimentally characterised (Mittal 2010) and the results are summarised in Figure 3-1 for one injection timing strategy. It can clearly be seen that the wall wetting is significantly reduced, however, due to the longer time span from start to end of injection for split injections, there is increased likelihood of piston wetting due to the fact the piston moves closer to the injector during the second injection.

![Figure 3-1 - Spray impingement through injection cycle for single and double injection (Mittal 2010) (fi\textsubscript{lw} and fi\textsubscript{pt} refer to left wall and piston top impingement respectively)](image)

![Figure 3-2 - Predictions of cylinder wall wetting by spray (Yi 2004)](image)
Predictions for cylinder wall and piston wetting were conducted as part of a larger simulation package (Yi 2004) for a pressure swirl injector and the results, shown in Figure 3-2, clearly predict a considerable improvement for split injection strategies in comparison to single injection strategies which produce the same overall injected mass.

Many aspects of modern injector design were characterised (Stach 2007), one of those being the ability of current injectors to deliver the split injection requirements imagined of future injectors. The study showed a significant reduction in levels of piston impingement with the use of split injection strategies.

![Figure 3-3 - Penetration for conditions of 300K, 5 bar (left) and 500K, 1 bar (right) (Martin 2010 (a))](image)

Among other parameters, the penetration from a piezo injector was studied under split injection strategies (Martin 2010 (a)) and (Martin 2010 (b)). The initial results, Figure 3-3 (left), show a reduction in penetration for all split injection strategies. However, this reduction is not as great for evaporating conditions, Figure 3-3 (right), possibly due to the cooling effect of the first spray and due to evaporation of the first spray.

![Figure 3-4 - Liquid and vapour penetration as a function of dwell time (Li 2003)](image)

A swirl injector was tested under a number of double injection strategies under ambient conditions of 10 bar and 500K (Li 2003). The initial tests focussed on the effect of dwell time and...
the results, in Figure 3-4, show that for all split injection cases there was a reduction in liquid and vapour phase penetration. It was also shown that as dwell time was increased, the reduction in penetration was also increased. The results sets are named in the form S (single injection) or D (double injection), percentage of fuel in 1st injection – length of dwell time (ms) – percentage of fuel in 2nd injection.

The change in spray penetration was also investigated for changes in duration of each of the two injections with the dwell time kept constant. These results, in Figure 3-5, show the strategy of a short 1st injection is unsuitable as the penetration of the liquid phase, which is responsible for pool fires, is far greater than for the other two strategies. From a penetration reduction point of view the other two strategies show a significant reduction in liquid phase penetration, and hence are both suitable in this sense. There is an inevitable cooling of the air by the 1st injection, which will hinder the evaporation of the 2nd injection, and for the 25/75 case the cooling effect has left too little heat energy to successfully evaporate a high enough proportion of the 2nd injection. However, further work by the same authors (Li 2004) describes how the evaporation of the 2nd injection can be enhanced by the entrainment of air by the first injection into the region in which the 2nd spray propagates.

Investigations were undertaken (Parrish 2012) using a multi-hole GDI injector operated with split injection strategies in a vessel held at evaporating temperatures. The vessel conditions were 11 bar and 700K. The study featured changes in dwell time, in terms of crank angle degrees, and changes in injected mass split ratio. The liquid and vapour phases were tracked by Mie and Schlieren imaging respectively. The results show that for all combinations of dwell time and injected mass split there is a reduction in both liquid and vapour penetration. The biggest reduction in vapour penetration is shown with a short/long injection mass split, whereas the biggest decrease in liquid penetration occurs with a long/short injection split. In terms of variations in dwell time, the evaporation in this testing is at such a level that the spray is quickly vaporised and therefore does not force the spray further downstream.
3.1.2 Improved Homogenisation / Improved Control of Fuel/Air Mixing

The spray distribution from a pressure swirl injector at engine conditions was simulated (Yi 2004). The results, in Figure 3-6, show a substantial improvement in spray distribution and a reduction in areas of rich fuel distribution.

An optical engine was operated using split injection strategies (Ishida 2011). Laser induced fluorescence was used to characterise the fuel vapour distribution. A comparison between the vapour distribution for a single and double injection is shown in Figure 3-7, which suggests an overall improvement in mixing for multiple injections.

Split injection strategies have also been proposed as a method for “spray shaping” (Schmid 2010). This is the method by which very short injections are used to introduce fuel to the cylinder with an overall low level of momentum and thus the penetration can be reduced. However, an extension of this is to be able to independently vary many parameters such as injection duration and injection timings to control, rather than reduce, penetration. This is
because at different speed/load points in the engine cycle the optimal spray morphology will be different. This work used a piezo injector capable of very short injections, but it is feasible that future advances in solenoid technology will make this a viable option for multi-hole injectors as well.

Further work by (Martin 2010 (b)) showed the potential for the second spray to be much wider and of lower velocity than the first spray. From a radial position of 15-30mm in Figure 3-8 is evidence of the toroidal vortex. It can be seen for the 2\textsuperscript{nd} injection its location has shifted outwards from the injector axis. This is most likely due to the presence of the fuel and fuel vapour from the 1\textsuperscript{st} injection blocking the original position of the vortex. This can only help to distribute fuel throughout the cylinder as many more vortices will be formed through split injection strategies.

![Figure 3-8](image.png)

Figure 3-8 - PDA measurements at a plane 10mm downstream for a 1.1ms dwell time (Martin 2010 (b))

The main body of work by (Li 2003) and (Li 2004) focussed on determining the suitability of the fuel/air mixture created for stable engine operation. These suitable limits were determined to be a local vapour phase equivalence ratio of between 0.7 and 1.3.

![Figure 3-9](image.png)

Figure 3-9 - Extent of stable combustion region for variations in dwell time (Li 2004)
The results shown, Figure 3-9 and Figure 3-10, are at a time of 5.8ms after start of injection, so well after the end of injection, and represent the equivalence ratio along the spray axis and on a plane at an axial distance of 35mm from the injector tip.

Variations in dwell time, Figure 3-9, seem to have an overall similar effect along the injector axis on the equivalence ratio. For the longer dwell times the equivalence ratio drops below the acceptable limit at a shorter downstream distance, but this is mainly due to a reduction in penetration. Radially from the injector axis the effects of increased dwell time are seen more clearly. There is an increase in the area suitable for combustion for an increase in dwell time, and for the longest dwell time the increase in the radius of the combustible region is of the order of 100% larger than for the single injection case.

For the changes in fuel delivery in each injection, Figure 3-10, it is shown that for the 25/75 case the evaporation reduction reduces the suitable operation range. For the two other cases there is a clear improvement, both in terms of the lack of over-rich regions from 10-50m downstream along the injector axis and an overall widening of the combustible region around the injector axis. The widest suitable region is shown with a 50/50 injected mass split.

![Figure 3-10 - Extent of stable combustion region for variations in injected mass split (Li 2004)](image)

Overall, these results show that a long dwell time is preferable for spatial extension of the stable operating region. This however must be offset against the time available for the injections within the engine operating time. It also appears that a fuel mass split of around 50/50 is most suitable.

Further to the earlier shown results from an optical engine (Ishida 2011) the equivalence ratio in the near vicinity of the spark plug was calculated from the LIF images. The results, in Figure 3-11, show a higher likelihood of near to stoichiometric conditions at the spark plug for the two-stage injection strategy, therefore indicating a more stable combustion event, and a reduction in the possibility of misfires.
3.1.3 Injector Behaviour

Inherent to the operation of GDI injectors is that the timing of the start and end of the injector control signal sent to the injector is not equal to the actual timing of the start and end of injection. This takes on increased importance for split injection strategies as it must be ensured an injection event is over before beginning the next one. Even the most accurate piezo injectors struggle to cope with the short dwell times required for split injections in GDI engines (Satkoski 2012).

It was shown for a particular pressure swirl injector (Lee 2002) that a 0.5ms dwell time, actually equated to a time between the end of one injection and the start of the next of 0.15ms. The parameters of injection delay for the start and end of the injection are very injector specific, as they are mainly a function of the strength of the solenoid and the spring which opposes it. However, there is also a delay from the needle lifting to fuel exiting the orifice, and equally needle closing to the end of fuel exiting the orifice.

Mitroglou et al (Mitroglou 2009) showed that with a multi-hole injector if the dwell time requested was too short there would be a small “pre-spray” between injections. For this modern solenoid driven injector the minimum dwell time possible was 0.5ms. This investigation also showed that on a global scale, the 1\textsuperscript{st} and 2\textsuperscript{nd} injections were generally very similar. However, it is believed that the current range of solenoid driven injectors are capable of delivering the required operation for split injection strategies (Stach 2007).

3.1.4 Behaviour in Early Stages of Injection

One point of note is the behaviour of the fuel and injector at the start of the 2\textsuperscript{nd} and subsequent injections. The fuel injectors are operated by solenoids, and at the end of injection these solenoids discharge the eddy currents remaining in them over a period of a few milliseconds (Wands 1982). However, for split injection strategies the next injection will often happen before the solenoid is fully discharged, and hence it is likely that the opening event of subsequent injections will be different to the 1\textsuperscript{st} injection.
The key role in early spray development played by the needle opening speed was shown (Kostas 2009). These experiments demonstrated a link between needle opening speed and initial spray velocity. Furthermore their work proposed a model of the propagation of the spray tip being influenced by a compressible region in front of the tip. Therefore it is entirely plausible that only small changes in the initial stages of injection can lead to a high degree of variation in the overall injection profile.

Using a 3-D model to simulate the diesel injection process (Salvador 2013) the effect of needle lift on a number of nozzle flow parameters was shown, Figure 3-12. For low needle lift (lower images) there is no vapour formation in the nozzle hole for high back pressure, but as back pressure within the cylinder is increased, thus reducing the pressure drop across the nozzle, vapour which has been formed in the nozzle seat gap is drawn into the nozzle. However, at high needle lift (upper images), vapour is formed on the upper side of the nozzle by geometric cavitation as the pressure drop reaches the critical value for that particular injector.

The results show that for a needle lift of over 75µm the mass flow rate reaches a maximum as the needle seat gap is no longer a limiting factor. However, throughout the valve lift event, the in-nozzle conditions are likely to undergo many changes through the different phases. Therefore, if there are changes in the needle lift profile, it is highly likely the initial spray formation will be very different.

Further to this analysis a similar model was produced (Befrui 2011), which showed the importance of flow separation and local velocity gradients on the formation of cavitation within a fuel injector nozzle. The results from this model were compared to experimental results which further substantiated the importance of the needle position, as even when in a fully open state the needle eccentricity caused variations between the experiment and model.

The correlation between needle lift in a diesel injector and spray formation was shown (Tsunemoto 1999). The variation in the pressure in the sac volume during the needle opening...
phase is shown in Figure 3-13. The sac pressure follows the needle lift profile, but while the needle shows a slow early opening period, before quickly opening, the sac pressure rises at a constant rate.

![Figure 3-13 - Pressure in sac volume as a function of needle lift (Tsunemoto 1999)]

The variability in spray penetration at different needle lifts was also tracked and, as seen in Figure 3-14, for low needle lift the variation is far greater than for high needle lift.

![Figure 3-14 - Penetration variability as a function of needle lift (Tsunemoto 1999)]

### 3.1.5 Engine Operation Implications

In addition to all the previous research on split injection strategies from the more fundamental viewpoint, the key work focuses on overall engine operation and the potential for emissions reduction, particularly hydrocarbon emissions. However, split injection strategies are proposed for use during either or both of the intake and compression strokes (Landenfeld 2004).
Using a single cylinder research engine a number of parameters such as injection timings and number of injections along with exhaust gas recirculation (EGR) were varied (Schmidt 2011). The injector used in this testing was a fast-acting piezoelectric injector, and the testing focussed on multiple injection events at stratified injection to increase the stable operating range, and hence the amount of time spent in this relatively low fuel usage mode. The results show for multiple injection events improved combustion stability and reduced NO\textsubscript{x} can be achieved with the aid of external EGR. The effect of the multiple injections is shown in Figure 3-15, and without EGR there is a clear decrease in NO\textsubscript{x} with an increase in number of injections. There is an inevitable efficiency penalty with the use of EGR, hence it is desired to achieve NO\textsubscript{x} reductions without excessive use of EGR.

Studies were conducted (Oh 2011) inside a single cylinder engine using split injection studies of gasoline fuel with a dwell time of 200\textmu s. The results show an improved combustion event, with increased stability over a wider range of injection timings. The combustion event was shortened and the maximum rate of heat release was increased. The emissions implications of this are a reduction in HC emissions over a wider range of injection timings as shown in Figure 3-16. The results also show a slight increase in NO\textsubscript{x} due to increased combustion temperature.
In a study using split injections from a swirl injector (Li 2004) with a fixed end of injection timing, the levels of NO\textsubscript{x} and smoke were reduced, but the level of HC was actually increased for splitting the injected mass. In this investigation the heat release rate was also higher for a single injection than for split injections. This appears to suggest the combination of split injection strategy with the proposed injection timings was non-optimal and shows the importance of fine tuning these engines.

It was shown that for split injection strategies it is vital to get the correct timing for all injection events (Daniel 2012). With the added unpredictability of injector operation which may come with split injection strategies, possibly even worse with an aged injector, the injection timing will become ever more important. This most likely will necessitate increased levels of engine control.

The opening behaviour of multi-hole injectors operating under split injection strategies was characterised both experimentally and numerically (Costa 2010). The investigation showed that split injection strategies are of little benefit at high engine load conditions, but at medium load conditions with correct choice of injection timings the levels of NO and CO can be reduced. It is also suggested that injection timing can be utilised for controlling engine knocking.

(Ra 2009) showed both experimentally and numerically the ability of split injections to control the combustion event. Injection timing and fuel mass split between the injections were found to be vital to controlling the NO\textsubscript{x} and soot emissions. It was also shown that injections at different times through the intake and compression stroke can be used to control different characteristics of the combustion event such as combustion efficiency and in-cylinder pressure rise.

Under stratified injection conditions engine out emissions were investigated and the work tracked the formation of these emissions during the combustion process (Hemdal 2011). The results comparing equivalent single and double injection strategies showed similar soot formation, but a reduction in engine out emissions showed a continued combustion process. There was no change in NO\textsubscript{x} observed and a decrease in HC emissions which was unaffected by ignition timing showing a large part of the HC emissions were caused by under mixing for single injection strategies.
3.2 Flash Boiling

During certain stages of operation of modern GDI engines the combination of increased fuel temperature and sub-atmospheric cylinder pressure during injection can lead to a phenomenon known as flash boiling. Put simply flash boiling is the rapid formation and growth of bubbles inside droplets, which leads to the shattering of the droplets into many smaller droplets as part of a two-phase flow region.

This chapter will deal with the conditions required for flash boiling to occur before moving onto the physics behind the flash boiling process on a local scale, and finally the effect of flash atomisation on whole spray systems, with particular focus on multi-hole injectors used in GDI applications.

3.2.1 Conditions for Flash Boiling to Occur

As previously mentioned flash boiling occurs with a combination of heated fuel and injection into sub-atmospheric conditions. More specifically the conditions for flash boiling to occur are that the ambient pressure into which fuel is injected is below the saturation pressure of the fuel. The saturation pressure has a relationship with temperature as defined by the Antoine equation (Antoine 1888) and (Thomson 1946):

\[ \log P = A - \frac{B}{t + C} \]  

Equation 3.1

Where A, B and C are the Antoine coefficients for the fluid and P and t are the ambient pressure and fluid temperature respectively. It was determined the Antoine equation to be the most useful and practical estimation of saturation vapour pressure to a very high degree of accuracy (Thomson 1946).

Antoine coefficients are readily available for most fluids (Chemistry WebBook 2014), but an experimental approach to find them is described (Nasirzadeh 2004) which correlates well with predicted values.

The saturation pressure curve for n-heptane is shown in Figure 3-17.
With the increased use of bio-fuels and alcohol fuels there is an increased likelihood of use of fuels with low boiling points (Marriott 2008), hence there is much scope for research into flash boiling sprays.

The difference between the actual conditions and the minimum conditions required for flashing is known as the superheat and can be expressed either in terms of temperature or pressure. The superheat degree is the difference between the liquid fuel temperature and the boiling point of the fuel for given ambient pressure conditions. The superheat can also be expressed in terms of the fluid vapour pressure and the ambient pressure conditions as shown in Equation 3.2.

\[ \Pi = \frac{P_{vapour} - P_{ambient}}{P_{ambient}} \]  

**Equation 3.2**

### 3.2.2 Bubble Nucleation and Growth

As previously mentioned the formation and growth of bubbles inside liquid fuel droplets is fundamental to the flash boiling process. Sher et al (Sher 2008) determined there to be 3 stages of flash boiling:

1. Bubble nucleation
2. Bubble growth
3. Two-phase flow

The first stage of this process is bubble nucleation. Bubble nucleation can materialise in two ways (Sher 2008):

- Homogeneous – bubbles form in a regular pattern within the liquid itself
- Heterogeneous – bubbles form on nucleation sites such as impurities in the fluid or the container walls

As the rate of heterogeneous bubble formation is dependent on the container walls and fluid impurities it is impossible to characterise, but the rate of formation of homogeneous bubbles can be characterised by $J$ as in Equation 3.3 (Schmelzer 2003):

$$J = J_0 e^{\left( \frac{\Delta G_c}{k_BT} \right)}$$

Equation 3.3

In which $J_0$ represents an initial constant for the fluid, $k_B$ the Boltzmann constant, $\Delta G_c$ the Gibbs energy and $T$ the temperature.

In any fuel injector heterogeneous bubble nucleation is inevitable under superheated conditions, from interaction with the nozzle walls and from fuel impurities. It has been shown that the nature of any flash boiling process external of the nozzle is highly dependent on the behaviour inside the nozzle (Park 1994). Hence an improved understanding of the heterogeneous bubble nucleation will be essential to a full understanding of the flash boiling process within fuel injectors. It has also been shown that bubble nucleation is intrinsically linked to the cavitation behaviour inside the nozzle hole of a multi-hole injector (Aleiferis 2010).

Upon formation of the bubbles, pressure fluctuations in the fluid will cause the bubbles to grow or contract. (Plesset 1977) described the stages of bubble growth:

1. At the initial stages, after the equilibrium is disturbed, the bubble growth rate is very low and is restrained by the surface tension
2. If the superheat is sufficiently large the growth rate eventually reaches the limit
3. The rate of vapour inflow, which is proportional to the square of the bubble radius, then becomes so large as to produce a substantial cooling of the surrounding liquid. Both inertial and thermal effects limit the growth rate in this stage
4. The growth rate then decreases, which reduces the importance of inertial effects until eventually the bubble is so large that the only limiting factor is the rate at which heat can be supplied to the bubble wall to effect phase change

The rate of bubble growth is related to the phase change between the liquid and vapour phases (Plesset 1954), (Saini 1975). The Jakob number, $Ja$, given in Equation 3.4, describes the ratio of sensible to latent heat absorbed during phase change and therefore is a key parameter in describing flashing sprays.

$$Ja = \frac{\rho_l C_p \Delta T}{\rho_v h_{fg}}$$

Equation 3.4

In which $C_p$ represents specific heat, $\Delta T$ superheat degree, $\rho$ the fluid density, with $l$ and $v$ referring to the liquid and vapour states and $h_{fg}$ the latent heat of vaporisation.
As the Jakob number approaches zero the bubbles created will remain the same size as the latent heat required for phase change is diverging. As the Jakob number is increased from zero there is a corresponding increase in the occurrence of phase change, and hence the bubbles will grow or contract.

It has been shown that over the duration of bubble growth there was a three stage process (Lee 1996). The initial stage is inertia controlled, with the bubble radius increasing in a close to linear fashion with time. The final stage is heat diffusion controlled, in which the bubble radius increases with the square root of time. There is an intermediate stage in which both the inertial and heat diffusion limits will take effect. This work builds on initial work by (Mikic 1970).

The bubble growth process was analysed by Kawano et al (Kawano 2004) (Kawano 2006) based on the following assumptions and is summarised in Figure 3-18:

1. The temperature and pressure inside bubbles are uniform and temperature must be identical to the temperature of the liquid fuel;
2. Bubbles grow spherically;
3. The phase change from liquid to vapour occurs continuously due to the growth process of cavitation bubbles inside the nozzle orifice and fuel droplets;
4. Marangoni convection in the liquid increases coalescence frequency among the growing bubbles (van Stralen 1979). However, the coalescence due to the Marangoni effect is not considered in this model.

As the bubbles grow through phase change within the droplet the ratio of bubble to liquid fuel increases. A characteristic, the void fraction, $\varepsilon$, describes the ratio of bubble to liquid within the droplet, as in Equation 3.5.

$$\varepsilon = \frac{V_{\text{bubble}}}{V_{\text{bubble}} + V_{\text{liquid}}}$$  \hspace{1cm} \text{Equation 3.5}

Once bubble growth has caused the void fraction to reach a critical value, such that bubbles are interacting with one another, the droplet will shatter and form a cloud of many small droplets. (Suma 1977) determined the value of $\varepsilon$ which correlates to this break-up procedure to be 0.51-0.53 for jet fuel. The model of Kawano et al (Kawano 2004) specifies that the number of

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure3-18.png}
\caption{Stages of bubble growth (I. S. Kawano 2006)}
\end{figure}
droplets after shattering will be twice the number of bubbles prior to it. After this process has been completed a mixture of liquid droplets surrounded by vapour will exist. The momentum of the original parent drop will be distributed among the secondary droplets.

3.2.3 Flash Boiling of Sprays

It has been shown that on a local scale the flash boiling process will greatly affect the atomisation process, however, it is essential to develop an understanding of the effects on the more global morphology and on the development of sprays.

Much of the experimental work discussed in this chapter has been conducted in spray chambers which have been used to simulate the pressure and temperature condition range found in engine cylinders and inlet manifolds. (Stansfield 2007) compared fuel sprays in pressure chambers with those in an optical engine and the results showed good agreement.

Spray models will also be discussed as they are becoming increasingly more capable of capturing the transient nature of flash boiling sprays. Such a model is from Chang and Lee who developed a parcel based model approach, which analyses the spray by splitting it into small bundles for each iteration, and predicted the overall spray parameters with a relatively high degree of accuracy (Chang 2005). This model shows the ability to model flashing sprays with a vast reduction in computational cost in comparison to most models simulating flashing conditions.

3.2.3.1 Flash Boiling of Jets

Many researchers have examined the effect of flash boiling on jet like sprays. A large amount of research in this area focuses on the emission of liquids which have been in storage under high pressure. If the container is accidentally breached there is a possibility of the fuel which rapidly exits the container undergoing flash boiling. Although the application is very different, the physics behind the spray break-up remains, hence these papers will be discussed.

Backlit photography was used to show that flashing jets contained an inner intact core surrounded by a diverging fine spray (Reitz 1990). The imaging results from Reitz are summarised in Figure 3-19, which shows images acquired with both Mie-imaging (a) and backlit shadowgraphy (b). The backlit images show effectively the intact core but not the surrounding mist of small liquid droplets and vapour due to appropriate selection of exposure time. It is seen that for the flashing case (right) the diverging angle is greatly increased and the intact liquid length greatly reduced in comparison to the non-flashing case (left). The results suggest that droplets are emitted from the unbroken liquid jet, starting from the nozzle exit. This theory is supported by the findings that the intact length of the core decreases with an increase in the liquid temperature. One disadvantage of the flash boiling process is highlighted in this work, that as conditions close to flash boiling are reached, the formation of air and vapour bubbles within the nozzle will increase, therefore decreasing the effective nozzle cross-sectional area and the fuel flow rate possible. (Sher 2008) also summarised the reduction in discharge coefficient of the nozzle associated with an increasing in level of flashing.
Cleary, Witlox and co-workers examined flashing jets from an experimental viewpoint (Cleary 2007) and these results led towards the formulation and implementation of drop size correlations (Witlox 2007). The experimental results show that an increase in superheat leads to an increase in disturbances within the spray during the transition from non-superheated, where mechanical effects are the primary cause of spray break-up to fully-flashing, where the bubble growth and burst procedure is the principal cause of spray break-up. The results agree with the transition criteria into the full destructive flashing suggested in (Kitamura 1986) based on both Jakob number, Ja, and vapour Weber number, We_v, as shown in Figure 3-20. The data points shown on the graph are from both Kitamura and also (Brown 1962).

The effect of degree of superheat on the SMD of jet sprays was investigated by Witlox, Kay and co-workers (Witlox 2010) and (Kay 2010). The results obtained on a variety of sprays, using the phase Doppler technique, showed a three stage process, as shown in Figure 3-21. For low and negative superheat, there is a small decrease in SMD for increased superheat. For moderate...
superheat degree, in the transition regime, there is a significant reduction in SMD for an increase in superheating. At the point the spray reaches a fully flashing state, further increase in superheat leads to a less substantial decrease in SMD. The decrease continues up until a point, controlled by the thermodynamic and physical properties of the fluid, after which additional superheat leads to no further decrease in SMD.

![Figure 3-21 - Three stages of drop size for variations in superheat degree (Witlox 2010) and (Kay 2010)](image)

Miyatake et al (Miyatake 2001) determined the flashing efficiency of a steam generating system, which was the ratio of steam generation, to the maximum possible generation for infinite superheat, as in Equation 3.6.

\[
\eta = \frac{1}{1 + A(T_{\text{sup}} - B)} \quad \text{Equation 3.6}
\]

A and B represent fluid specific constants, and \(\eta\) and \(T_{\text{sup}}\) representing the flashing efficiency and superheat degree respectively. The results show that for a given fluid, the flashing efficiency is independent of the mean velocity of the working fluid, the equilibrium temperature, the length of the nozzle and the nozzle diameter. For the jets used in this study, the point at which the jet shattered and proved unable to maintain its profile correlated with a flattening of the flashing efficiency curve and occurred at superheat degrees of the order of 10K, but it must be noted that these results do not cover a wide range of operating conditions.

A low pressure liquid jet was examined (Zeigerson-Katz 1996) and the results showed that under flashing conditions the overall spray SMD was independent of orifice dimensions for the conducted experiments, showing the flash boiling to be the dominant factor in the spray break-up under these conditions.

Superheated jet sprays have also been investigated using the PDA technique (Zhou 2012). Axial and radial velocity measurements were also analysed along a number of downstream planes.
and it was revealed that when velocity is normalised by maximum velocity and distance from
the injector axis normalised by axial distance downstream, the same profile is observed for all
distances downstream. This suggests a near uniform jet expansion process. It must be noted
that the closest measurement plane to the injector was 50mm downstream, and hence the near
nozzle expansion was not investigated. It is expected this will be significantly different.

A model for a flashing jet emitted from a water coolant system was developed (Razzaghi 1989).
This work focused on drop size estimations after the break-up of droplets by the bubble
nucleation and growth was complete. It was shown that an increase in both discharge pressure
and temperature led to a decrease in the droplet diameter. The model suggests the duration of
the bubble growth period to be in the order of 20-1000µs with increased superheat leading to a
decrease in the duration.

Polanco et al (Polanco 2010) conducted an extensive review of flashing jets, which drew
together a large volume of work, both experimental and theoretical and the conclusions
discussed were shown across the work of a number of researchers. It was shown there was a
strong sensitivity in the jet formation to small changes in superheat condition, in terms of both
pressure and temperature in addition to a dependency on the dimensions of the exit orifice.
This last point is not in agreement with the findings of other researchers, and therefore must be
true for some orifices and not true for others. The jet angle is strongly affected by the strength
with which the fuel flashes, with higher degrees of flashing leading to greater fuel dispersion
and greater jet angles. The jet angle is also influenced by the droplet concentration and the level
of turbulence within the jet. The minimum temperature distance is defined as the point at
which the boiling and bubble nucleation has ceased, and further atomisation is controlled by
aerodynamic processes. The temperature profile shows a decrease until this point followed by a
recovery to the downstream ambient temperature.

The main conclusions drawn from the literature on flashing jet sprays are:

- Flash boiling leads to increased break-up of the liquid spray
- A wider dispersed spray is formed with a shorter intact core
- The vaporisation rate has been increased
- At moderate superheat, a partial flashing stage exists
- In partial flashing stage there is a linear decrease in spray SMD with increasing superheat
degree
- At high superheat the spray loses its jet like morphology
- SMD is much reduced at full jet degradation stage
- SMD appears to be driven mainly by superheat degree
- Bubble growth duration was characterised and shown to be related to superheat
- Modellers have successfully captured the interaction of flashing and mechanical break-
up mechanisms to build predictive models
3.2.3.2 Flash Boiling of High Pressure Hollow Cone Sprays

Original GDI engines featured pressure swirl injectors, hence they have been investigated under flashing conditions and this research will prove valuable in the understanding of the flash boiling spray from multi-hole injectors.

A pressure swirl injector was investigated under superheated conditions using the planar laser induced exciplex fluorescence technique (PLIEF), to track both the liquid and vapour phases of the spray (Schmitz 2002). The vapour and liquid distribution for iso-octane for non-superheated and superheated cases is shown in Figure 3-22. The results are consistent for all fuels when superheated, and show that the vapour phase collapses to form a gas jet and a denser region on the spray axis.

![Figure 3-22 - Differences in spray morphology for pressure swirl injector under non-flashing (left) and flashing (right) conditions (Schmitz 2002)](image)

The investigation also tracked the spray tip axial penetration and revealed that, for each fuel when superheated conditions were met, there was a significant increase in penetration, as in Figure 3-23. This is related to the spray collapse and entrainment of the vapour phase.

![Figure 3-23 - Influence of injector temperature on penetration (fuel pressure 80 bar, chamber pressure 0.5 bar, chamber temperature 323K) (Schmitz 2002)](image)
Obokata et al conducted an imaging and PDA investigation on a pressure swirl spray for variations in ambient pressure and temperature (Obokata 2005). The Mie-imaging results reveal a narrower liquid phase of the injection, particularly later in the spray period. This is due to a combination of increased evaporation and spray collapsing. The PDA results show a significant increase in axial velocity for heated fuel and a small decrease in SMD. The PDA results were split into a number of phases of the injection. The phases will not be discussed in detail as it is assumed that they are very injector specific, but it is shown that the difference in the behaviour of superheated and non-superheated sprays is constantly evolving throughout the injection period.

Zuo et al successfully modelled the spray from a pressure swirl injector under a variety of superheated conditions (Zuo 2001). The model was shown to effectively predict the global spray dispersion when applied to in engine conditions. The model also predicted a number of other spray characteristics such as droplet temperature, size and vaporisation. The results showed a significant decrease in SMD with superheating of fuel. The droplet temperature predictions were summarised in Figure 3-24 along with the velocity field, which shows three superheat cases of $\Delta T=0$, 30, 60K. The colour scheme uses red to represent the initial injection temperature to blue which represents a temperature of 300K. These predictions show for the non-superheated case, formation of vortices on the spray tip and a reduction in temperature in these vortices. For the 30K superheat case there was a reduction in penetration and a stronger vortex formation. There was again a reduction in drop temperature in the vortices. For the 60K superheated case there was an extensive reduction in penetration and increase in the strength of the vortices. For this case there was a clear reduction in drop temperature immediately on exit from the injector orifice, due to the temperature gradient between the fuel and the air surrounding the spray and the reduced duration of bubble expansion.

![Figure 3-24 - Modelling of pressure swirl sprays with superheat degree of 0, 30 and 60K (left to right) - left side of images shows equivalence ratio contours, right side shows gas velocity - droplet temperature is signified by colour, with blue as 300K and red as injected fuel temperature (Zuo 2001)](image)

Another model was produced (Zeng 2001), which characterised the spray break-up for a hollow cone spray in terms of both aerodynamic force and as a cause of bubble growth. Traditional aerodynamic break-up models were modified to account for the effects of bubble growth and it was shown that an increase in void fraction lead to a reduction in break-up time. It was also shown that there existed a linear decrease in SMD with an increase in void fraction.
In terms of the effect of superheat degree, an increase in superheat leads to a decrease in the break-up time. As shown in Figure 3-25, the same cases were modelled without the effects of bubble growth aiding the break-up, and it was shown the break-up time was substantially lower for the flash boiling than the aerodynamic break-up, proving the flashing effects represent the primary break-up mechanism for superheated sprays.

![Figure 3-25 - Variation in spray break-up time with superheat (Zeng 2001)](image)

From this break-up the spray SMD was calculated, again with and without the effects of flash boiling and the bubble growth. As shown in Figure 3-26 there is a clear reduction in the SMD with the addition of flash boiling into the break-up model. These results also show little reduction in SMD when superheat degree is increased over 12K, however, there is a definite reduction in break-up time with an increase in superheat.

![Figure 3-26 - Variation in spray SMD with superheat (Zuo 2001)](image)

The model also predicts the liquid and vapour distribution, which shows that for an increase in superheat there is a substantial change in both the liquid and vapour distributions, as shown in Figure 3-27. It is of interest that although there is little change in SMD for superheat degrees of
over 12K, the morphology of the spray is significantly altered, and a superheat degree of between 23.5 and 64.2K is required for complete shattering of the spray.

<table>
<thead>
<tr>
<th>Superheat degree</th>
<th>4.2K</th>
<th>23.5K</th>
<th>64.2K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid phase distribution</td>
<td><img src="image1.png" alt="Image" /></td>
<td><img src="image2.png" alt="Image" /></td>
<td><img src="image3.png" alt="Image" /></td>
</tr>
<tr>
<td>Vapour phase distribution</td>
<td><img src="image4.png" alt="Image" /></td>
<td><img src="image5.png" alt="Image" /></td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
</tbody>
</table>

*Figure 3-27 - Calculated planar spray distribution for spray from high pressure hollow cone spray under various superheat degrees (Zuo 2001)*

The main conclusions drawn from the literature on high pressure hollow cone sprays are:

- As with jet sprays, flashing sprays exhibit increased break-up and shorter break-up times
- Increasing superheat levels lead to progressive widening of the spray angle until the point at which the spray collapses to form a solid cone
- The vapour is drawn into the spray core under flashing and non-flashing conditions
- Axial velocity and therefore spray penetration are increased for flashing conditions
- Drop size is reduced by flash boiling
- Evidence of vortices forming on the edge of the spray tip

### 3.2.3.3 Flash Boiling of Low Pressure Sprays

Until recently port fuel injection (PFI) engines were commonplace, but have since been replaced by GDI engines in many circumstances. However, the initial research into these injectors under flashing conditions will still help to build insight into the application of flash boiling onto spray systems.

Low pressure PFI sprays were analysed under a variety of superheat degrees (Aquino 1998). High speed Mie-images, Figure 3-28, showed a 4-stage process with an increase in superheat. Initially without superheat, the spray shows its normal 4-stream profile. With an increase in superheat the fuel spray develops a milky like appearance. With a further slight increase in superheating the plumes collapse to form a coalesced flow. Finally above a certain degree of
superheat, the wide and misty spray formation is developed due to complete spray break-up due to flash boiling.

<table>
<thead>
<tr>
<th>Standard spray</th>
<th>Milky</th>
<th>Coalesced</th>
<th>Wide and misty</th>
</tr>
</thead>
</table>

Figure 3-28 - 4 stages of superheating of low pressure PFI sprays (Aquino 1998)

The transition criteria between the different spray types are shown in Figure 3-29. The milky like profile is not included as it is merely a transition between the standard and coalesced states. An extra state, the transition state, between the coalesced and misty states is added.

Figure 3-29 - Transition criteria between stages of superheating for low pressure PFI spray (Aquino 1998)

Senda et al tested both n-pentane and n-hexane fuels under flashing conditions with a PFI injector both experimentally (Senda 1992) and numerically (Senda 1994). As the back pressure was reduced to below the saturation vapour pressure, for ambient temperature conditions, there was a clear change in the morphology of the spray. The spray width was increased and further decrease in pressure resulted in an increase in the spray cone angle. For the pressure condition just under the saturation vapour pressure the spray took on a dense, narrow spray, similar to the spray described as coalesced by Aquino et al (Aquino 1998).

The results are summarised in Figure 3-30, which shows that for high superheat degrees there is as expected an improvement in atomisation quality as characterised by a reduction in break-up
length and SMD. This is also associated with an increase in spray angle, and therefore an increase in droplet dispersal. For pressures close to the saturation vapour pressure the opposite is true. There is an increase in the break-up length and SMD. This reduction in the level of atomisation corresponds to a reduction in the spray angle and therefore droplet dispersion.

![Figure 3-30 - Variation in spray parameters with ambient pressure (Senda 1992)](image)

The infrared extinction/scattering (IRES) technique was used to characterise the vapour distribution in a flashing PFI spray from a pintle injector (Adachi 1997). The results show a significant increase in vaporisation rate with a reduction in surrounding ambient pressure and therefore superheat degree and this increased vaporisation leads to improved fuel/air mixing and an overall more homogeneous mixture.

Further researchers have shown for a low pressure swirl spray a decrease in overall spray SMD for an increase in superheat degree (Kim 2012). With an injection pressure of less than 2.5 bar, it was presented that there is actually an increase in the SMD with the addition of superheat, possibly showing some negative effects of void formation in the nozzle. The plot in Figure 3-31 shows, as for the high pressure fluid injection cases, a reduction in SMD associated with an increase in superheat degree. In this case the reduction in SMD is close to linear.
Further researchers investigated PFI injectors both experimentally and numerically under a variety of superheat conditions (Oza 1983). Their results show two distinct regions of flash atomisation. Firstly the spray exits the injector as normal, before the large pressure drop it undergoes causes the onset of flash boiling. However, with a great enough degree of superheat the fuel will begin to flash within the injector nozzle and leads to the formation of a wide dispersed spray. For the external flashing mode, numerical solutions showed a correlation between the intact length before flash boiling occurred and the degree of superheat. This
relationship is summarised in Figure 3-32. The three separate lines are for different levels of initial droplet turbulence upon nozzle exit, δP.

The main conclusions drawn from the literature on low pressure sprays are:

- Superheating of fuel again leads to changes in the spray morphology
- With low superheat the spray goes through a coalesced state, with low spray angle, high break-up length and increased drop size
- High levels of superheat lead to a spray shattering, similar to that seen with jet sprays
- With spray shattering the spray angle is increased, break-up length reduced and SMD reduced with increasing levels of superheating
- A linear relationship was shown between superheat degree and SMD

### 3.2.3.4 Flash Boiling of Multi-hole Sprays

As shown the flash boiling process has a great effect on the morphology of jet sprays, and as multi-hole injectors are effectively a collection of tightly spaced jet streams, it is to be expected that the spray morphology will be altered by similar magnitudes by the flashing process.

Researchers at Shanghai Jiao Tong University have studied an 8-hole, 90° cone angle, GDI injector under flash boiling conditions using a number of experimental techniques. The techniques include planar laser induced exciplex fluorescence (PLIEF) (Zhang 2011 (a)), imaging (Zhang 2012) (Zeng 2012 (a)) (Zeng 2012 (b)), particle image velocimetry (PIV) (Zhang 2011 (b)) (Zhang 2012), laser sheet drop sizing (LSD) (Zeng 2013).

![Figure 3-33 - Variation in liquid dispersion for a multi-hole injector under different ambient pressure and temperature environments (Zeng 2012 (a))](image)

Mie-imaging results, in Figure 3-33 for ethanol fuel, show a clear tendency towards flashing conditions for increased temperature and reduced pressure conditions. It can be seen that for gradual increases in superheat the spray plume width increases, before the plume width
becomes great enough that the sprays merge and spray collapse is initiated. These results agree with those of numerous researchers on the subject (Mojtabi 2011) and (Dahlander 2006).

The results for two alcohol fuels are shown in terms of ambient pressure over saturation pressure, which shares a logarithmic relationship to superheat degree, in Figure 3-34 and Figure 3-35. The results show a consistency for both fuels in the overall spray morphology for different levels of ambient over saturation pressure. It must be noted that for partial spray collapse to occur the value of Pa/Ps must be below around 0.5 and for full spray collapse, or “flare flashing” as it is often referred, the value must be below 0.2.

<table>
<thead>
<tr>
<th>Pa/Ps</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.4</td>
</tr>
<tr>
<td>0.989</td>
</tr>
<tr>
<td>0.71</td>
</tr>
<tr>
<td>0.44</td>
</tr>
<tr>
<td>0.27</td>
</tr>
<tr>
<td>0.13</td>
</tr>
</tbody>
</table>

Figure 3-34 - Variation in spray dispersion of methanol fuel for different levels of ambient over saturation pressure (Zeng 2012 (a))

<table>
<thead>
<tr>
<th>Pa/Ps</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.31</td>
</tr>
<tr>
<td>0.97</td>
</tr>
<tr>
<td>0.69</td>
</tr>
<tr>
<td>0.42</td>
</tr>
<tr>
<td>0.267</td>
</tr>
<tr>
<td>0.15</td>
</tr>
</tbody>
</table>

Figure 3-35 - Variation in spray dispersion of ethanol fuel for different levels of ambient over saturation pressure (Zeng 2012 (a))

The PLIEF technique is used to track the distribution of both liquid and vapour phases of the fuel. The results show that for an increased superheat degree there is an increased level of stream to stream interaction and with sufficient superheat degree the plumes all combine to form a single central plume. The collapsed spray has a much greater penetration than the partially collapsed spray, to around the level seen in the non-superheated spray. It is shown that the vapour phase collapses more severely than the liquid phase. As the superheat degree is increased to a high enough level to induce full spray collapsing the total vapour mass is increased significantly. Under spray collapsing conditions the highest concentration of the vapour phase is concentrated in the spray core in a thin pencil-like structure. The PLIEF results are summarised in Figure 3-36.
From the results the overall vaporisation of the fuel could be determined and the increase in vapour mass which correlates with an increase in superheat degree is shown in Figure 3-37. At the superheat degree at which the spray collapses, the slope of the plot increases, showing that the spray collapse has an influence on the level of flash boiling.
The PIV investigation (Zhang 2011 (b)) initially featured an imaging investigation. These results reveal that with an increasing degree of superheat the total spray morphology changes. The imaging results across 2 planes downstream of the injector are summarised in Figure 3-38 and Figure 3-39:

- With low superheat the individual spray plumes become wider and more dispersed
- Further superheating leads to the individual plumes interacting with each other and forming a solid ring
- With a high enough degree of superheat the spray plumes collapse
- Further downstream under a high degree of superheat the spray forms new spray streams in the gaps between the original streams (these new spray streams are known as interstitial streams)

![Figure 3-37 - Relationship between superheat degree and vapour mass (Zhang 2011 (a))](image-url)
These results show that for this injector and fuel configuration the superheat required to complete spray collapse and formation of interstitial streams is 65°C.

The PIV results show that with an increase in superheat the nature of the air motion surrounding the spray is altered. The PIV results show that the air motion is in agreement with the spray position. For moderate superheat, before the plumes interact the air is entrained into the plumes. With increased superheat, while the fuel forms a ring like structure, the air is entrained equally around the spray core radially outwards. For further increases in superheat the air travels into the gaps between the spray plumes from the spray core, which is the cause of the interstitial stream formation.

Further work by these authors tracked the location of the spray tip vortex (Xu 2013). This showed that under partial collapsing conditions the vortex moves away from the injector axis as the spray travels downstream. However, under full spray collapse, or flare flashing, the vortex travels back towards the spray axis as the injection progresses in time. These results are summarised in Figure 3-40, in which the crossed area represents the vortex core.
Figure 3-40 - Location of vortex core as spray travels downstream (Xu 2013)

Figure 3-41 shows the variation of this vortex core for a set time. This shows that the vortex moves in towards the spray axis for increased superheat and also further downstream. This vortex effectively represents the radial extent of the spray tip.

Figure 3-41 - Vortex core position variation with increasing superheat (Xu 2013)
Laser sheet drop sizing (LSD) results show a clear trend of a reduction in SMD for increased superheat degree. The results also show that the reduction in SMD which is usually associated with increased injection pressure is not present for superheated conditions, demonstrating drop size independency from the injection pressure. It is also shown that there is a reduced sensitivity to injector configurations.

The results of the imaging and PLIEF investigations are summarised in Figure 3-42. The spray behaviour is split into three stages in terms of flash boiling tendency.

As expected the vapour quantity increases with decreasing ambient pressure below the saturation pressure. As the ambient pressure is taken below the point at which flare flashing is initiated the ratio of vapour quantity per unit reduction in pressure is increased.

The distance between the plumes is at a maximum under non-flashing conditions. As ambient pressure is reduced towards the flare flashing initiation point, the distance between the plumes is reduced gradually. By the flare flashing initiation point the spray plume gap is reduced to zero, such that the sprays combine and form a blockage between the outside and inside of the spray cone.

This is related to the spray plume width, which increases as the pressure is reduced down from ambient to the flare initiation point. Below this point the plumes have combined into a single plume.

As the flashing level is increased there is a reduction in spray penetration. This is as a cause of increased dispersion in the individual spray plumes. At the inception of flare flashing the spray penetration begins to increase with increased superheat. This is the point at which the air

![Figure 3-42 - Summary of spray behaviour changes with reduction in ambient pressure (Zeng 2012 (a))](image-url)
Entrainment begins to dominate the fuel motion and resultant spray morphology due to the low pressure region in the spray core and the large number of small droplets and fuel vapour.

Non-dimensional analysis was conducted on all these results along with further results obtained for over atmospheric conditions (Zeng 2012 (a)). The results show that sprays with similar magnitudes of Weber number, Reynolds number and air-to-liquid density ratio have similar characteristics for superheated and non-superheated sprays.

Further analysis of the gasoline fuel dispersion under superheated conditions was conducted by separate authors (Marriott 2008), which showed again the formation of interstitial streams in the gaps between the original spray streams under flashing conditions, Figure 3-43.

This work shows flare flashing and the formation of interstitial streams to occur under a significantly lower superheat degree than the previously detailed work.

Dahlander and Lindgren (Dahlander 2008) showed that for a GDI injector, the distribution of the vapour phase was dependent on injector cone angle. Two otherwise identical injectors were
examined using the LIF technique, one with a 70° cone angle of and the other with an 85° cone angle. It was shown for the wider cone angle injector the vapour phase matched the liquid phase, but for the lower cone angle injector the vapour phase was split between 2 locations. As for the other injector there was a large quantity of vapour within the spray plumes, but there was also a central vapour core, in an area into which no liquid fuel had been dispersed. The results are shown in Figure 3-44, with the 70° injector on the left and the 85° injector on the right.

![Figure 3-44 - Different fuel dispersion for 70° and 85° cone angle injectors (Dahlander 2008)](image)

A further investigation used a number of two-hole nozzles with different diverging angles between the hole orientations to examine the effect of this design feature on the spray formation using the laser absorption scattering (LAS) technique (Sato 2009). The results were obtained at temperature and pressure conditions of 500K and 10bar. The results shown in Figure 3-45 for the three different diverging angle configurations show a clear spray collapse induced by stream to stream interactions. The spray collapse leads to an increase in spray penetration, but in some ways a reduction in the spray dispersion. This reduction in spray dispersion may not be ideal for combustion performance.

![Figure 3-45 - Different levels of spray collapsing for different cone angle, 2-hole injectors (Sato 2009)](image)

Aleiferis et al examined the behaviour of various fuels and fuel blends under superheated conditions (Aleiferis 2013). The investigation features a 6-hole injector and uses high speed
backlit imaging to characterise the fuel sprays. The imaging results show an increase in the level of flashing near the nozzle exit with both an increase in fuel temperature and a decrease in chamber pressure. This is observed for both the start of injection and the steady state injection period, and is most noted for the n-pentane fuel as in Figure 3-46.

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Initial stages of injection</th>
<th>Steady state injection period</th>
</tr>
</thead>
<tbody>
<tr>
<td>20°C</td>
<td>1.0 bar 0.5 bar</td>
<td>1.0 bar 0.5 bar</td>
</tr>
<tr>
<td>50°C</td>
<td>1.0 bar 0.5 bar</td>
<td>1.0 bar 0.5 bar</td>
</tr>
<tr>
<td>90°C</td>
<td>1.0 bar 0.5 bar</td>
<td>1.0 bar 0.5 bar</td>
</tr>
<tr>
<td>120°C</td>
<td>1.0 bar 0.5 bar</td>
<td>1.0 bar 0.5 bar</td>
</tr>
</tbody>
</table>

Figure 3-46 - High-speed, high-magnification spray imaging, n-Pentane (Aleiferis 2013)

The authors also used the phase Doppler technique to characterise the droplet diameter within the spray. The plot in Figure 3-47 shows the $D_{10}$ through the duration of the spray. It can be clearly seen that an increase in both pressure and temperature will lead to a reduction in the mean drop size, as is expected from the theory on flash boiling.

Figure 3-47 - $D_{10}$ drop size for n-pentane fuel (Aleiferis 2013)

A laser diffraction meter was also used in the course of this testing to characterise drop size. It can be seen in Figure 3-48 that the mean drop size measured with the laser diffraction meter (LDM or LD) is significantly lower than that from the PDA technique. This suggests that either the PDA technique is not recording as many small droplets as the LDM technique or the LDM technique is underestimating drop size. Laser diffraction meters can overestimate the numbers of small droplets if the beam is diffracted by a number of droplets, as the large diffraction angle produced by multiple scattering is similar to that produced by diffraction from a single small droplet (Black 1996).
The results shown in Figure 3-49 confirm that the phase Doppler technique employed in this case is indeed not recording a large number of droplets with diameter under 5µm, particularly in the main body of the spray, 1000-2000μs, during which time there are no droplets recorded with diameter under 5µm. Many researchers (Mojtabi 2010), (Skogsberg 2005), (Heldmann 2013) have examined these sprays through various measurement techniques and found there to be a significant number of drops of this size expected from multi-hole nozzles. Therefore, it is concluded that the light energy lost through the thick optical windows led to an input laser power which was too low for measureable light levels to be transmitted by the smallest droplets. It is therefore essential to ensure sufficient laser power is used in any PDA testing where the input and output beams must pass through optical windows.

Figure 3-48 - Comparison between LDM and PDA results (Aleiferis 2013)

Figure 3-49 - Raw droplet data for gasoline, 20°C, 1 bar, showing lack of small droplets measured (Aleiferis 2013)
One important consideration is how a multi-component fuel will behave under superheated conditions, where the different components have different boiling points. Such a model has been produced (Ra 2009). The model reveals that a reduction in ambient pressure or increase in ambient pressure leads to an increase in evaporation rate through the duration of droplet vaporisation. On a single droplet’s evaporation process it is shown that the initial drop temperature affects the early stages of evaporation, but the later evaporation is independent of initial droplet temperature. It can be seen that for the non-superheated case in Figure 3-50 (a) the heavier fuel components travel further downstream quicker than the lighter components. This is expected due to their increased momentum. The effect of superheating is shown in Figure 3-50 (b). It can be seen for the lighter fuel components, the spray penetration is retarded. However, for the heavier fuel components the superheating of the fuel shows little effect on the fuel mass distribution. It should be noted that not all the components of the fuel are superheated, and the lighter fuel components have a lower boiling point at the atmospheric pressure used.

![Figure 3-50 – Dispersion of gasoline components for non-superheated (left) and superheated (right) sprays (Ra 2009)](image)

Negro et al (Negro 2011) modelled a multi-hole GDI injector fuelled with a variety of multi-component fuels. The simulations featured a 1-D model, which was developed for earlier work (Bianchi 2008), for the nozzle flow which focussed on resolving the liquid/vapour equilibrium. The model shows that the primary break-up is initiated by phase change in the nozzle. The output from this 1-D model is shown in Figure 3-51 and shows that increase in fuel superheating results in a decrease in droplet diameter and an increase in bubble diameter.
These results were used as the input for a 3-D model for the external flow which featured a specific model for vaporisation from superheated droplets. The results showed that the vaporisation process is driven by the superheat degree and the latent heat of vaporisation of the mixture. The model showed an increased evaporation rate for gasoline/methanol blends than gasoline/ethanol blends due to the lower saturation temperature of methanol than ethanol.

Another model was produced (Gopalakrishnan 2008) to examine the behaviour of superheated fuel within GDI nozzles. The model built on earlier work (Reitz 1990) which showed a gradual decrease in mass flow rate through the nozzle with an increase in temperature, before suffering an extreme reduction in which the possible mass flow rate drops to a small fraction of its previous value. 2-D simulations reveal that the flash boiling begins well inside the nozzle, usually at the sharp inlet corner and is usually geometrically induced. The simulations also reveal that the altered velocity profile at the nozzle exit leads to the formation of a spray plume with a wider cone angle due to the variation in internal flow for an increase in temperature. 3-D simulations showed asymmetric vapour generation, which was due to geometric effects. These results show flash boiling must be considered along with other in-nozzle effects when considering the overall flow pattern, as the competing mechanisms are complementary to one another.

Neroorkar and Schmidt produced a model to investigate the effect of flash boiling on the spray formation from multi-hole GDI injectors for a number of test fuels and experimental conditions (Neroorkar 2011). The authors proposed that under flashing conditions there exists a three stage process in the nozzle for increasing superheat. The three stages are: a geometrically driven phase, a second phase with the residence time in the nozzle acting as the limit and a third phase which is vaporisation rate controlled. The model also interestingly showed the potential for the formation of “string flash boiling” which appears similar to string cavitation, as in Figure 3-52.
The main conclusions drawn from the literature of flash boiling of multi-hole injector sprays are:

- Flash boiling greatly affects the spray dispersion
- Break-up time is reduced and vaporisation rates increased
- Small droplets and vapour are more likely to be drawn into the low pressure region in the spray core
- As spray jets interact with one another this creates a blockage between the outside and inside of the spray preventing air passage equalising the pressure difference
- Increased superheat leads to wider spray jets and therefore more interaction
- Interaction can also happen in non-superheated conditions for narrow cone angle injectors
- The level of superheat required for different injector (particularly different cone angles) is vastly different
- Vortices are formed on the edge of the spray tip
- A correlation was shown between ambient pressure/saturation pressure and overall spray dispersion
- As in all previous sprays, flash boiling reduces the mean drop size in the spray
- For moderate spray collapse the penetration is reduced, but for full spray collapse the axial penetration is increased as all the droplets and vapour travel along the injector axis
- Under flashing conditions interstitial streams are formed in the gaps between the original spray streams
- Void formation in the nozzle is often geometrically induced
- Some evidence of “string flashing” was seen in the sac volume

### 3.2.4 Effect on GDI Engines

It is evident that the flash boiling process has a pronounced effect on the sprays from GDI injectors, and therefore will lead to differences in the combustion cycle. On the one hand, the smaller drops, and enhanced atomisation will lead to a shorter and more efficient combustion cycle, but conversely the combustion must take place in an enclosed chamber, under closely controlled conditions and the spray changes caused by flash boiling may not always adhere to the requirements of the engine.
Tests were conducted (Kabasin 2010) in which the total emissions for a GDI engine fuelled with E85 over a 20 second period were captured. The results in Figure 3-53 show a significant reduction in hydrocarbon and carbon monoxide emissions, but on the other hand a significant increase in NO\textsubscript{x} emissions.

![Figure 3-53 - Emissions from 20 seconds of GDI engine running with normal and heated fuel (Kabasin 2010)](image)

Fuel heating was proposed (Piock 2011) as a method for particulate emissions reduction. As is shown in Figure 3-54, an increase in coolant and oil temperature, and therefore fuel temperature, leads to a reduction in particulate number for two different injection timings.

![Figure 3-54 - Potential of heated fuel for particulate number reduction for two different injections timings (Piock 2011)](image)

Brake specific emissions were calculated from a single cylinder GDI engine under normal and heated fuel conditions (de Boer 2013). The results showed a reduction in emissions and a reduction in fuel consumption for a number of engine operating points. The results for hydrocarbon (HC), carbon monoxide (CO), nitrous oxides (NO\textsubscript{x}), brake specific fuel consumption (BSFC), smoke, particulate matter (PM) and particulate number (PN) are presented in Figure 3-55.
For GDI engines the likelihood of flash boiling presents an issue for engine operation from an emissions perspective. The occurrence of under or over mixing of the fuel and air within the cylinder can lead to a rise in emissions levels (Sandquist 2000). GDI engines are optimised in terms of spray mixing for non-flashing conditions so with the change in morphology initiated by the flash boiling process the mixing process is likely to be non-optimal for a flashing spray.

A premixed charge compression ignition (PCCI) engine was fitted with the capability of fuel heating prior to injection (Seko 2001). The results for high load engine operation, Figure 3-56, show a clear trend of decreased CO and HC emissions, most likely related to a shorter, more complete combustion event, however, the downside of this is the higher NOx levels, which are
likely related to an increase in the combustion temperature. The low load results show similar trends, but with even more of a NO\textsubscript{x} penalty at high fuel temperatures.

A number of researchers have investigated the use of heated injectors in PFI engines to improve atomisation prior to the fuel entering the cylinder. These investigations are very useful as the effects of charge motion inside the cylinder on the engine performance are minimised and the effect of the reduced drop size on engine performance can be observed.

Richter et al (Richter 2002) used a heated PFI injector to examine the differences seen with flashing fuel. The investigation revealed that although there is an increase in atomisation for flashing sprays, the increased spray dispersion leads to higher wall wetting and therefore an emissions increase.

Experiments were conducted to assess the performance of a heated PFI injector (Beheshti 2011). The results showed a reduction in the levels of all emissions products over the equivalent non-heated injection. These results were shown to be consistent over a range of injection timings and engine load points. The results showed a coefficient of variability significantly lower than the PFI equivalent, showing the increased spray repeatability due to the superheat playing a major role in the spray break-up procedure. The fuel consumption of the engine was also shown to be reduced; by 6.5% compared to the PFI equivalent and 1.8% compared to an equivalent modern GDI alternative.

Flashing sprays have also been investigated in diesel engines from both a power and emissions point of view. (She 2010) showed that for hot fuel, which is likely to undergo at least partial flashing, there is an increase in specific power and reduction in NO\textsubscript{x}. However, for smoke and CO emissions there is a reduction for moderate temperature increase, followed by an increase for further increased temperatures.

From this research it can be seen that flashing sprays have a wide variety of negative and positive effects on engines and further understanding of these sprays will lead to enhanced exploitation of the numerous benefits.
4 Experimental Techniques and Facilities

Engine diagnostics are essential to continue to increase the level of knowledge on the in-cylinder processes, which is key to further refinement of the internal combustion engine (Fansler 2005), (Hung 2008). Increased knowledge of the fuel spray and air charge motion will give more insight to the air/fuel mixing process. Analysis of engine combustion also aids the process of engine design.

As previously mentioned the fuel injection process is integral to overall engine efficiency. Increased knowledge of injector performance leads to engines which can be further optimised in terms of many factors such as engine geometry, injection timing and injection pressure. A large number of techniques are available for this characterisation procedure. All these techniques are complementary and no single technique provides all the necessary data required, so the amalgamation of data from a number of techniques will be required to characterise injector performance. This chapter will introduce a number of these optical diagnostic techniques with complimentary information about measurement techniques used to characterise engine airflow and combustion. The chapter will also provide in-depth descriptions of the experimental set-ups utilised for the work detailed in this thesis.

4.1 Techniques

4.1.1 Imaging

Fuel spray images are normally obtained through three imaging set-ups: shadowgraphy, Mie-scattering and Schlieren. These techniques will be described and their role within fuel spray imaging discussed.

4.1.1.1 Shadowgraphy

The shadowgraphy technique involves capturing backlit images of the fuel spray. This involves a flash panel and camera orientated as in Figure 4-1.

![Figure 4-1 - Shadowgraphy schematic](image)

Shadowgraphy is a technique used by many researchers all over the world (Mathieu 2010) to investigate GDI sprays. With correct optimisation of the set-up parameters the light will only penetrate through the areas of low optical density, effectively the areas with low fuel density. In areas where the optical density of the spray is too great for the light to pass through the camera will record a reduced light signal.
4.1.1.2 Mie-Imaging

The Mie-scattering technique is based upon elastically scattered photons. These photons originate from a light source placed at an angle to the receiver (camera). The Mie-scattering effect is only present for liquid fuel particles so therefore the photons which reach the camera must all have been deflected by the liquid phase. The illuminating light can be in the form of a full flash panel or a single light sheet. Light sheet illumination is very useful for sprays such as that produced by a pressure swirl injector where a simple back-lit or global Mie-scattered image does not show the hollow nature of the spray core. An example of this is shown in Figure 4-2. The intensity scaling has been inverted; hence the areas of high light reflection appear in black.

An extension of the Mie imaging technique is that of the Structured Laser Illumination for Planar Imaging (SLIPI) which uses a number of different modulated input beams. A combination of the scattered light from these different signals is used to account for the multiply scattered light (Berrocal 2010) such that the final signal is based only on single scattered light, and therefore is a more accurate representation of the spray.

4.1.1.3 Schlieren Imaging

Schlieren imaging is used to observe the gas phase of the injection process. It is essential to understand the evaporation process of sprays as this is essential to the overall combustion efficiency. Images can be obtained in a number of configurations. A knife-edge is utilised so only the refracted light is obtained at the camera sensor. More information about Schlieren imaging is available (Kook 2011).

4.1.2 Phase Doppler Anemometry (PDA)

Phase Doppler anemometry (PDA) is a highly useful tool for fuel spray characterisation as it gives information on both the size and trajectory of individual droplets within the fuel spray (Durst 1997). PDA is a point based flow diagnostic technique, in which information is deduced from the influence each droplet has on a laser beam crossover. A laser, usually an Argon-ion laser, is used to produce laser beams at two wavelengths for 2-D PDA. A series of optical devices
are used to split each of the original two beams into another two beams with matching polarisation and focus these beams onto a measurement zone within the spray. This process creates a measurement zone with a fringe pattern as the peaks and troughs as the polarised light beams interfere as shown in Figure 4-3. As the droplet passes through the measurement volume light is refracted towards the receiver with a typical simplified signal in Figure 4-4 (left). This represents the basic principle of the operation of a Laser Doppler Anemometry system (Durst 1976). The signal which reaches the receiver is in the form of a series of peaks, related to a droplet passing through the light peaks and troughs in the measurement volume and the frequency of these peaks can be used to determine the speed at which the droplet has passed through the measurement zone, as the distance between the fringes in the measurement volume is known. To calculate the droplet diameter a second receiver is required. The phase difference between the signals arriving at each of the receivers is used to calculate droplet diameter as the relative position of each of each receiver is known, as in Figure 4-4 (right).
For studies with gasoline like fuels the receiver is usually placed at a forward scatter angle of 70° as this ensures the reflected light component is zero and the 1st order refracted light is maximised as shown in Figure 4-5 and also to minimise the effects of refractive index changes due to either droplet temperature changes, or for multi-component fuels (Saffman 1990).

![Optics of a droplet](image)

**Figure 4-5 - Optics of a droplet**

### 4.1.3 Other Spray Diagnostic Techniques

This section details a variety of other spray diagnostic techniques which are widely used and could in future be combined with the aforementioned techniques used in the Loughborough University Spray Lab.

#### 4.1.3.1 Laser Induced Fluorescence

Laser induced fluorescence (LIF) is a technique based on inelastically scattered light from the fuel spray, through fluorescence (Miles 2011). A high energy laser pulse is fired towards the spray under investigation and the frequency shifted fluorescence signal is detected. The laser light directed towards the spray can be in the form of a beam to conduct point based measurements, a light sheet to conduct planar measurements or an expanded beam to observe more global phenomena. Band pass filters are essential to record only the inelastically scattered light, and hence block the elastically scattered light from the camera sensor.

As different species present within the fuel spray fluoresce under different input light frequency signals and further produce light of different frequency under fluorescence conditions this technique is useful for discovering the presence and position of different species within the fuel spray, and concentrations can be calculated from the strength of the fluorescence signals. This process can be used to examine the evaporation of fuel sprays as the gas phase and the liquid phase will produce different fluorescence signals.

#### 4.1.3.2 X-Ray Imaging

The X-ray imaging technique reveals much about the morphology of the fuel spray (Liu 2009). While various other imaging set-ups reveal information about the gas and liquid phases and the structure of the liquid core, the X-ray technique can be utilised to observe the break-up mechanisms in the spray through imaging of individual droplets and bubbles. These bubbles
often represent cavitation bubbles formed within the nozzle. The most developed X-ray imaging set-up is that situated at Argonne National Laboratories Advanced Photon Source (Liu 2009). This system is used to produce picosecond light pulses, such that droplets, ligaments and bubbles within the spray are captured at a single point in time.

4.1.3.3 Laser Diffraction Meters
The most common laser diffraction meter (LDM) is the Malvern. The technique works with a laser beam which is split into many known beams. The diffraction of these beams, as recorded by a detector array, contains information about the fuel passing through the measurement volume. The Malvern is often used to gather information on drop size within the fuel spray. The average drop size data acquired by the laser diffraction meter should be comparable to that of a phase Doppler system; however, in dense sprays this is not the case due to multiple scattering effects (Quoc 2011). It is determined that for results to be of acceptable quality to begin to draw conclusions beam transmission must be at least 40%. It must also be noted that while the PDA technique acquires data at a single point over a time span, the Malvern gathers data at a single point in time over a larger area. Comparisons have been conducted between the data acquired with a Malvern meter and a Phase Doppler system (Fdida 2008), (Dodge 1987).

4.1.3.4 Patternators
Patternators work to determine the footprint of a spray on a defined plane.

Mechanical patternators (Bade 2009) use an array of condensers to determine the presence of fuel particles at a point. In each of these condensers there is an electrode pair, which can measure the capacitance of the fluid within the condenser with high temporal resolution and therefore obtain the rate at which the fuel is filling that condenser.

Optical patternators (Ullom 2001) work with a series of laser sheets and optical detector arrays. At a high refresh rate the detectors monitor the light signal. Combination of the signals from the detectors can determine at which points within the planar measurement volume fuel particles, which block transmission of the beams, are present.

4.1.3.5 Raman Scattering Techniques
Raman scattering, similar to LIF, uses the inelastically scattered light signal given out by the particles within the fuel spray (Beyrau 2010). When a particle within the spray is subjected to an intense light energy a small number of the incident photons will, rather than be elastically scattered, be instantaneously inelastically scattered. The time scales for this are significantly shorter than the other inelastically scattered light signals. Analysis of the Raman scattered light can be used to calculate species population within the fuel spray.

4.1.3.6 Ballistic Imaging
Immediately upon exiting the injector a liquid core of fuel is formed (Agrawal 2013). This liquid core is broken up as the fuel travels downstream due to shear break-up (Hiroyasu 1982). Traditional measurement techniques have been unable to probe the nature of this liquid core,
however in recent years the depth of knowledge in this area has increased exponentially (Linne 2010).

Ballistic imaging is basically a shadowgraphy technique, but the time span over which the image is acquired is of order of nanoseconds as opposed to a millisecond. This is achieved through use of a Kerr gate, which opens an aperture for a short space of time. This allows only the ballistic photons, the photons which have either transferred straight through the measurement volume without being scattered or have only been scattered a small number of times, to reach the receiver.

4.1.4 Injection Rate Testing

One of the most informative tests to conduct on a GDI injector is to measure the instantaneous flow rate throughout an injection period. This, along with the injector trigger signal and injector drive current, will give detail on many aspects of the injection as detailed in Table 4-1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
</table>
| Opening delay                 | • Mechanical delay for drive current to charge solenoid and lift needle  
                                 | • Hydraulic delay for fuel to exit the nozzle after the needle has been lifted |
| Opening rate                  | • The rate at which the injection rate grows during the needle lift phase   |
| Maximum injection rate        | • Steady state injection rate reached when needle is fully lifted            |
| Real injection duration       | • The actual time the injector is releasing fuel                            |
| Closing rate                  | • The rate at which the injection rate decreases during the needle closing phase |
| Needle bounce                 | • To assess the existence of, and time relative to end of injection of a needle bounce event |
| Total injected quantity       | • The integral of the injection rate profile is the injected mass           |

4.1.4.1 Bosch Rate Tube

A Bosch rate tube works on the principle that when fuel is discharged from an injector into a tube filled with liquid the pressure rise produced is proportional to the injection rate (Gill 1966). The injection rate is calculated according to Equation 4.1.

\[ q = \frac{P A_t}{a \rho} \quad \text{Equation 4.1} \]

The tube used is filled with whatever fluid is being tested, in this case gasoline or a representative single component test fuel, and must be of a certain length such that the standing pressure waves produced by the injection are not reflected back to the injector during
the injection time, as this would interfere with the measurement procedure. The pressure sensor used is a quartz type transducer, which is situated in a recess just downstream of the nozzle exit. The pressure of the fluid within the tubing can be held anywhere from 1 bar to 100 bar, with higher back pressures leading to a slight reduction in injection rate.

Rate tube testing has proved to be a highly accurate and quick method, of gaining vital information on the injection event (Bower 1991), (Arcoumanis 1993).

4.1.4.2 Akribis

The Akribis system from Loccioni uses injection into a small pressurised chamber (Loccioni 2013). This chamber features a piston which is displaced downwards by the injected fuel and the rate of displacement is linked to the injection rate. Below the piston is another chamber of pressurised nitrogen, which opposes the piston movement. At the end of injection when the piston ceases to move downwards a relief valve is opened to allow the injected fuel to leave the chamber to return to the original back pressure level and the piston position in time to accurately measure the subsequent injection.

4.1.5 Flow Diagnostic Techniques

As previously discussed knowledge on the engine air flow is essential to understanding the combustion process. In recent years the vast majority of optical air flow measurement has utilised the PIV technique, when previously the LDA technique was the most common technique. A number of mechanical methods are also available, but the mechanical techniques involve interrupting the flow, so are less accurate than optical techniques.

4.1.5.1 Particle Image Velocimetry

Particle image velocimetry is a planar laser based flow diagnostic technique, as summarised in Figure 4-6. A double pulsed laser is used to produce a light sheet and a camera usually orientated at 90° to this light sheet records a pair of images of the measurement zone separated by a small time interval. The flow is seeded with small particles which follow the flow. The motion of these particles can be tracked between the two images recorded. Therefore fluid motion can be determined as the temporal image separation is known.
PIV can also be used to calculate air motion in 3 directions. This requires the use of a second camera, with both cameras observing the measurement volume at a different angle. The particles can therefore be tracked as they pass through the thickness of the laser sheet.

Further to 3-component PIV it is also possible to conduct 3-dimensional PIV. This involves using multiple parallel laser sheets. The imaging can be done through either multiple cameras, or a single camera with a second lens which has multiple lenses embedded into it, such that the resulting image has: both information on position in the x-y plane, as in 2-D PIV, but also information on depth.

4.1.5.2 Laser Doppler Anemometry

LDA is based on the same principles as the Phase Doppler technique previously discussed, but only a single receiver is used in the axial plane such that drop size cannot be calculated. Similar to PIV seeding particles are placed within the flow and as these particles pass through the measurement volume they direct the fringe pattern towards the receiver. As the measurement principle is not dependent on all the light signals having been deflected towards the receiver through the same manner there is no restriction on the positioning of the receiver relative to the input laser beams.

4.1.6 Combustion Diagnostic Techniques

As useful as knowledge of the fuel/air mixing process within the cylinder is, the eventual goal is highly efficient combustion.

Chemiluminescence refers to the light energy given out during the combustion process. Basic chemiluminescence images will show the whole zone which is undergoing combustion. However, use of short-pass wavelength filters will allow only the chemiluminescence signal from separate species within the combustion zone to be viewed. The two radicals most often tracked are the CH and OH radicals (Varea 2010).

LIF is also a useful technique for observing in-cylinder combustion and can be used to track the flame front on a planar level (Cessou 2010). Therefore, rather than looking on a global level, as in the chemiluminescence technique previously discussed, it is possible to observe the flame front on a planar level, therefore providing spatial information on the burned region near the spark plug and the unburned region towards the edge of the cylinder. A laser sheet is used to excite the particles formed on the flame front, usually the OH* radical and the fluorescence signal from this particle is recorded.

4.1.7 Summary

It is clear that these techniques are all complementary to each other and all will help build up a complete picture. Figure 4-7 shows the purpose of all the relevant diagnostic techniques and the complementary roles which they play, and also includes others which are not discussed in this work (Leipertz 2007).
4.2 Experimental Equipment

This section aims to give an overview of the available experimental equipment suitable for research in spray diagnostics within the facilities of the Department of Aeronautical and Automotive Engineering at Loughborough University, and will discuss the experimental set-ups used to obtain the results detailed in this thesis.

4.2.1 Imaging

The Loughborough University fuel spray imaging set-up is of the basic shadowgraphy type. A PCO Sensicam 12-bit type camera is used in conjunction with a Fostec flash panel connected to an EG&G xenon flash unit. The timing is controlled primarily by a Stanford Research DG645 digital delay generator. The CamWare program is used to control the interaction of flash and camera timing and for recording purposes. The images are processed using PCO Pictures, and this processing procedure will be discussed in detail in the following section.

4.2.1.1 Image Processing

The images taken with the shadowgraphy system can be processed in a number of ways as shown in Table 4-2.
<table>
<thead>
<tr>
<th><strong>Scale Image</strong></th>
<th><strong>Background image</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Taken for calculation of spray penetration. Nozzle must be in the measurement position as the nozzle tip is used as the origin for the scaling system.</td>
<td>Taken to be subtracted from individual spray images to account for variations in background intensity.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Raw image</strong></th>
<th><strong>Raw image with background subtracted</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Accounts for variations in spatial background light intensity.</td>
<td>Accounts for variations in spatial background light intensity.</td>
</tr>
</tbody>
</table>
4.2.2 High-Zoom Mie-Imaging

For investigation of the spray morphology upon exit of the nozzle, a high-zoom Mie-imaging set-up was used. The same camera as for the shadowgraphy study was used but this time equipped with a 2x converter, extension rings totalling 107mm and a 105mm macro lens. This achieves an image area of ~3x3mm. This relates to around 3μm per pixel. The area directly below the nozzle was imaged and the spray was illuminated using the xenon flash unit, as previous, connected to a fibre optic bundle. As the centre of the spray was associated with excessive levels of multiple scattered light, a plane 0.3mm from the spray axis was imaged. The angle between camera and input light was approximately 15°. This angle was determined experimentally as the ratio of illumination of the large drops to illumination of the small drops was high. The injector was kept in the orientation as shown in Figure 4-8 and Figure 4-9 (left) as this was essential due to the small depth of field of the imaging set-up. The depth of focus was in the order of 0.4mm. An example of the image captured is shown in Figure 4-9 (right).

PCO CCD digital camera with lens system of 2x converter, extension rings totalling 107mm and 105mm macro lens

Xenon flash unit delivered through fibre optic bundle

Figure 4-8 - High-zoom Mie schematic
The images presented from this set-up are all single shot images to highlight the large droplets and ligaments in the spray.

4.2.3 Far-Field Mie-Imaging

Far-field Mie-imaging was also employed for various reasons within the body of this work. The same high resolution camera and xenon flash unit were used. Firstly, a global Mie set-up was used, which refers to an experiment in which light is introduced to the spray in the form of a diffuse light source. In this case a flash panel mounted at an angle of 90° to the camera with a scattering angle of 90°. Secondly, planar Mie images were acquired in which a light sheet was passed through the spray from an input angle of 90° to the camera. These planar Mie images were used for investigation of the spray footprint on a plane at various time steps as the spray passes through the plane upon which the PDA measurement volume sits.

4.2.4 Phase Doppler Anemometry

The phase Doppler technique is used for analysis of fuel sprays produced by automotive injectors. The set-up is of the traditional discrete optic type, thus allowing full control of all beams. The system consists of a Coherent Innova 90C Argon-ion laser combined with a Dantec 57X10 receiver. The Loughborough University Phase Doppler system is shown schematically in Figure 4-10 and the full design details can also be found (Wigley 1999).

The configuration for the 488 and 514 nm laser beam wavelengths at the final focussing lens is: beam diameters of 5 mm, equal beam pair separations of 50 mm, laser powers of 100 and 200 milli-watts per beam with a horizontal polarisation. With a focal length lens of 300 mm this produces coincident measurement volumes of diameters of 37 and 39 microns with fringe spacings of 2.94 and 3.10 microns respectively for the two wavelengths. The 514 nm beam pair
are in the vertical plane to measure the axial droplet velocity and size with the 488 nm beam pair in the orthogonal plane to measure the radial droplet velocity.

The Dantec 57X10 receiver is positioned at a scattering angle of 70 degrees with the polariser set to collect only scattered light in the horizontal plane and an aperture micrometer setting of 0.5 mm. This optical configuration results in an effective measurement volume length of 0.1 mm. In-conjunction with the Dantec processor the transmitter and receiver set up produced a droplet velocity bandwidth of -30 to 110 m/s with a drop size measurement range of up to 100 microns.

The injector under investigation is supported from a gantry incorporating a rotation stage and three precision orthogonal linear traverses to orientate and position the spray in three dimensions relative to the static PDA measurement volume. The horizontal traverse is computer controlled.

### 4.2.5 Rate Tube

A rate tube working on the Bosch principle is used by Loughborough University (Gill 1966). The overall length of the tubing is 6079mm. This was combined with a Kistler 12C pressure transducer located close to the injector tip, which was all mounted in an injector mount which was manufactured in-house. The transducer was connected to a charge amplifier. Due to pressure fluctuations in the near vicinity of the nozzle and the pressure transducer which are not caused by the injected fuel the results from this rate tube are not sufficiently accurate to be analysed quantitatively, but for the analysis presented within this thesis the accuracy of the measurement technique is considered satisfactory (Arcoumanis 1993). A schematic of the rate tube set-up is shown in Figure 4-11.
4.2.6 Atmospheric Rig

The atmospheric rig has been used in conjunction with the shadowgraphy imaging and phase Doppler systems for many years at Loughborough. The atmospheric rig consists of a translation stage for the injector with accurate x, y and z traverses. Fuel is delivered to the injector via a 2-stage pump process, with a low pressure pump operating at 4 bar, followed by a high pressure pump which can operate at up to 250 bar. All investigations detailed in this thesis used a fuel pressure of 100 bar. Pressure was controlled by a pressure regulating valve. The rig is shown in Figure 4-12 along with the shadowgraphy system.
4.2.7 Pressure Chambers

Pressure chambers are used to simulate injection into the conditions experienced under fired engine operating conditions. Various pressure chambers can be run at temperatures up to 1000°C and up to pressures of 100bar, therefore simulating even diesel injection conditions. It is also of interest to simulate low pressure conditions, such as those which can be experienced in gasoline engines operating with high levels of throttling, which can produce flash boiling of the fuel (Nishikori 2011).

Loughborough University are in possession of two High Pressure/High Temperature cells, one of which was designed within the scope of this work. These pressure cells are designed such that fuel can be injected into any set of conditions, in terms of temperature and pressure, to simulate the conditions found within a gasoline engine at the time of injection.

4.2.7.1 90° Window Pressure Chamber

The original pressure chamber can be run at any air pressure from around 0.1-10 bar and chamber temperature 20-100°C. The low pressure cases are designed to represent a gasoline engine running with high levels of throttling or reduced valve lift (Diana 2001) and the high pressure cases is designed to represent injection during stratified charge operation.

Solenoid driven valves are connected to a high pressure nitrogen feed on the inlet and a vacuum pump on the exhaust. The chamber is mounted on a heater plate, which heats the entire cell, and the cell is insulated to avoid excessive heat loss. However, for the results presented in this thesis which were obtained with this cell the injector/fuel heating method which has been developed for the second pressure chamber is used.

This pressure chamber was used at Loughborough for work on a previous PhD project (Mojtabi 2011) to examine the flash boiling phenomenon, and it was considered further investigation of the flash boiling phenomenon would prove vital to the field of engine research. One limitation of the this high pressure cell is that its viewing windows are orientated at 90° and 180° to one another, as seen in Figure 4-13, and as such only backlit and Mie imaging studies have been undertaken on flash boiling sprays.
For the reason of further investigation of GDI sprays over a range of temperatures and pressures, similar to the those occurring in the engine, a pressure chamber with optical access for PDA studies has been designed, constructed and validated in the course of the work in this thesis. The chamber is designed to cover the same operating range as the previous 90° chamber, but has two windows orientated at 110° to each other, hence a 70° scattering angle for light entering one window and exiting the other. One major difference is that this chamber features heating of just the injector and the surrounding injector holder to achieve fuel heating as opposed to heating the whole chamber. This is in order to reduce the testing time at high temperatures, for which the previous set-up could take over an hour to reach 80°C. This improved method of heating could reach a stable operating temperature of 80°C in around 10 minutes. The 70° window chamber and its relationship to the phase Doppler measurement system is shown in Figure 4-14.
Another improvement is that the new chamber has a fully automated LabVIEW control programme. The LabVIEW program has sensors for temperature at 2 points within the system, one on the injector tip and one on the injector holder, and also a sensor for the chamber pressure. The injector holder is heated by two 175W cartridge heaters, and as before the inlet is connected to a pressurised nitrogen supply and the exhaust to a vacuum pump. The control software works in the procedure shown in Figure 4-15 for sub-atmospheric conditions, with the exhaust closing after the inlet to allow negative pressures to be achieved within the chamber. For over-atmospheric conditions the inlet would be kept open longer than the exhaust.

The variables in Table 4-3 can be controlled by the user. A full cycle refers to a purge cycle followed by a series of injections.
### Table 4-3 - Timing control variables for chamber operation

<table>
<thead>
<tr>
<th>Overall</th>
<th>• Number of repeats of cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>• Cylinder pressure</td>
</tr>
<tr>
<td>Inlet</td>
<td>• Opening time</td>
</tr>
<tr>
<td></td>
<td>• Closing time</td>
</tr>
<tr>
<td>Exhaust</td>
<td>• Opening time</td>
</tr>
<tr>
<td></td>
<td>• Closing time</td>
</tr>
<tr>
<td>Injector</td>
<td>• Time from exhaust closed to start of injection cycle</td>
</tr>
<tr>
<td></td>
<td>• Injection frequency</td>
</tr>
<tr>
<td></td>
<td>• Number of injections between purge cycles</td>
</tr>
</tbody>
</table>

The cartridge heaters are controlled on a separate circuit in which the temperatures from the injector tip and the injector housing are monitored, with the injector tip temperature being used to define the injector temperature. This should adequately represent the fuel temperature, as the fuel will be subject to the hot surfaces inside the injector and injector holder for around 50-200s during testing. For commercial reasons internal injector dimensions were not available, hence the large error margins in this number.

Over the course of a full day working with this pressure cell it was found to be necessary to align the laser beams into the measurement volume 2-3 times to ensure accuracy of results and high data rates. However, with careful management of injection duration and the number of injections between each purge cycle the level of fuel within the chamber could be kept low enough that it was all vaporised and drawn out the cylinder during the purge cycle.
5 Results – Split Injection Strategies

Split injection investigations were conducted in two separate studies. Initial investigations used a Continental XL2 injector, which is the previous generation of production multi-hole injector. Further investigations were undertaken with the replacement for this injector, the XL3 injector. As the injector behaviour is very different for each injector, the result sets will be dealt with separately.

5.1 XL2 Investigations

Results were obtained using a Continental XL2 injector through a number of spray characterisation techniques. Shadowgraphy was used for visualisation of the general spray morphology and the early spray development. These results were complimented by PDA results, with focus paid to the spray propagation downstream, especially the differences between the first and second sprays. Further to this, the PDA drop size results were also investigated in depth. Injection rate was characterised using the rate tube method.

The results presented here use a 3-hole injector, as opposed to the normal production 6-hole injector. The 3-hole injector is specifically designed for research purposes to show similar spray streams to the 6-hole variant. The reduced hole number allows improved access for the PDA input beams and transmitted signal and allows visualisation of a single spray, as shown in Figure 5-1. Inevitably the reduced number of holes will lead to some minor changes in the spray behaviour, with this injector being designed to have the same internal flow patterns as its 6-hole equivalent (Private Communication 2010).

This investigation was the first of its type on the injector in question which meant the injector behaviour under multiple injection strategies was of particular interest. Therefore the range of dwell times, which is the time between successive injection events, is relatively short: 350, 400, 500 and 600μs. A pair of 500μs injections was separated by the aforementioned dwell times. In
practical applications there is scope to use up to 4 or 5 injections per engine cycle, however for the scientific purposes of this investigation only two injections were used. Shorter dwell times were not possible as the needle would not close during the shorter injector off period. The longest dwell time tested represents a “normal” second injection, which is one which is similar in nature to an injection 1 as defined by initial testing, i.e. the effects on the second injection from the first injection were suitably small. The data for the shortest dwell time case, 350μs, is not discussed in detail in all results sections for reasons which will become apparent.

5.1.1 Shadowgraphy

The shadowgraphy images presented in this section are for the dwell time cases of 400μs and 600μs. Similar images were also recorded for all other dwell times investigated. The flow structure at the time the second spray tip reaches each of the four axial locations of the PDA measurement volumes is shown in Figure 5-2 and Figure 5-3. The colour coding represents graduations of 12.5% in light intensity. The laser beams, entering from right to left, can be seen in each image as a distortion of the coloured contours. It is clear from the spray images that the tip of the second stream, shown in the red circle, unlike the case for the first injection, cannot be uniquely defined due to the density of droplets in the tail of stream one.

For a dwell time of 400μs a broadening and flattening of the second spray tip can be seen at Z=10 and 20mm from the bulbous shape of the tip, which will be discussed further using the
PDA data. This broad tip becomes less pronounced as it propagates downstream to Z=30 and 40mm, but there is still some evidence of overall spray widening. The broadening of the second spray tip is less pronounced at longer dwell times as can be seen in the images for a dwell time of 600μs. Following behind the spray tip a zone of high concentration fuel can be seen, which represents the near steady state fuel delivery.

Based on the images with the whole of one spray in the focal plane, taken with the injector in the 90° orientation, a small amount of fuel was observed exiting the nozzle between completion of the first spray and the second spray starting for dwell times of 500μs or greater. This fuel stayed local to the nozzle and thus was not seen in the PDA data at Z=10mm. This was attributed to a needle bounce which occurred from t =1175μs to t =1375μs after the start of electronic injection. With a dwell time of 400μs the second injection begins at t =1175μs and hence, coincides with the bounce. With the dwell time 500μs, the second injection begins during the bounce and the fluid from the two events combine to form the injection stream. The combination of the fuel from the needle bounce and the early fuel from injection 2 can be seen in Figure 5-4. The images show the early stages of the second injection for the 600μs dwell time case to be similar to the start of a single injection, whereas for the shorter dwell time cases, in which the needle bounce affects the early stages of the injection, the morphology of the tip of the second spray is significantly different to that of the first spray.
Figure 5-4 - Early spray 2 formation for different dwell times (single spray shown with injector orientation rotated relative to previous images)

From these images it was also observed that the flow angle is different between the first and second sprays. Closer to the injector axis the cone angle is seen to be reduced for the second spray for the cases with a dwell time of 500μs and less. This will be discussed in more detail in relationship to the PDA data.

### 5.1.2 Injector Behaviour

It has been well documented (Mulemane 2004) that the start and end of injection are critical phases of the injection process and it is essential to understand how the use of split injection strategies affects the injector opening and closing phases. Beyond the previously shown possibilities of starting subsequent injections during the needle bounce, there is little change
observed during the closing phase, but during the opening phase there are a number of interesting differences. These will be discussed first in terms of injector drive current, which is linked to needle opening velocity, and injection rate profile, both of which are shown in Figure 5-5 in relation to their trigger signals.

The comparison of injector drive current, based on time after the start of the trigger signal for that injection, in Figure 5-6, shows that for all dwell times investigated the initial time to peak current is reduced for all of the injection 2 cases, in comparison to the injection 1 case. This is due to eddy currents remaining within the solenoid from the first injection reducing the energy, and hence time, required to charge it. Another interesting observation from this plot is that as the dwell time is reduced there is a corresponding reduction in the time to peak current.

![Figure 5-6 - Injector drive current for injection 2 for different dwell times compared to that from injection 1](image)

The equivalent plot for injection rate is shown in Figure 5-7 which shows a significant advancement of the start of injection for the injection 2 events. With a dwell time of 400μs the additional advance is due to both the reduced time to peak current in the injector drive current in combination with the start of injection 2 coinciding with the needle bounce from the end of injection 1, as confirmed by the imaging study. The quicker needle opening is likely to be advantageous for atomisation as it reduces the time spent in the needle opening phase, during which the needle seat gap limits the pressure in the nozzle holes which has negative effects on atomisation quality. However, the increased overall fuel mass delivery will require careful control in engine operation to maintain correct air to fuel ratios. It is expected that throughout the life of the injector, the correlation between dwell time and total fuel mass delivery will
evolve, necessitating advanced control systems to be twinned with any engine using split injection strategies.

![Graph showing instantaneous injection rate for injection 2 for different dwell times compared to that from injection 1.](image)

**Figure 5-7** - Instantaneous injection rate for injection 2 for different dwell times compared to that from injection 1

### 5.1.3 Global Scale PDA Results

The complete PDA data set quantifying the instantaneous droplet velocities and sizes for the two transient sprays with four different dwell times in both space and time represents a massive amount of data. As such, data will be selected to highlight the expected maximum interaction between the two sprays. The first data to be presented is in the plane \(Z=10\text{mm}\) for \(x=10\text{mm}\) where the maximum axial velocities were found to occur for each spray and is denoted as the spray axis.

The discrete axial droplet velocities shown in Figure 5-8 indicate that 25,113 samples were recorded in the time window 0ms, i.e. the electronic start of injection, to 4ms with a time varying mean velocity profile derived from averages based on consecutive 30\(\mu\text{s}\) time bin sectors. Data were acquired for 200 consecutive injection cycles (one injection cycle combines both a 1\text{st} and 2\text{nd} injection) at an injection frequency of 4 cycles per second. The first droplets for spray 1 arrive at the PDA measurement volume at just after 0.5ms with a mean velocity of 68m/s. The broadening of the spray tip, seen for the second spray at \(Z=10\text{mm}\) in the images, Figure 5-2, is shown in Figure 5-8 for both the first spray and the second spray by the wide range of droplet velocities in the early spray region. This is also a consequence of the flow angle increasing slightly, with the maximum velocities being found at \(x=10.5\text{mm}\), while the velocity profile across the spray becomes flatter. A drop size class analysis shows that the low velocity droplets have sizes below 5\(\mu\text{m}\) whereas the droplets with velocities up to 100m/s have larger drop sizes. This...
is due to the smaller droplets having lower momentum, hence being slowed by the ambient air through which it is penetrating, and therefore being more likely to be entrained or deflected.

Between 0.75 and 1.0ms spray 1 ‘recovers’ reaching a near steady mean velocity of approximately 80m/s with peak velocities in excess of 100m/s. After this the velocity decays rapidly due to the needle closing and the sample number recorded increases as the spray density decreases and laser beam transmission increases.

The end of spray 1 is quite abrupt with the impact of spray 2 appearing at 1.30ms with a sharp increase in the velocity, up to 65m/s at 1.38ms. The tip of spray 2 encounters a field of small droplets with D_{10} values of nominally 3μm moving with a mean axial velocity of nearly 25m/s. These are entrained into the early tip of spray 2 contributing to the data in the time period 1.30 to 1.38ms. After this, spray 2 also exhibits a flow angle change and flattening of the velocity profile across the spray stream, similar to spray 1. However, unlike spray 1 the flow angle reduces with the peak axial velocity found at x=8mm. The recovery is fast and the same near steady velocity of 80 m/s as spray 1 is reached at 1.56ms. The flow angle change does vary with dwell time and will be discussed later. The main difference between the two sprays though is that the steady state period for spray 2 is held longer, until approximately 1.95ms, i.e. periods of 0.25 to 0.39ms respectively for spray 1 and spray 2.

The discrete drop sizes are shown in Figure 5-9 with profiles of the arithmetic mean, D_{10}, lower trace and Sauter mean diameters, D_{32}, upper trace, superimposed for the 30μs time bin sectors. Of the 25,113 samples plotted only 1,452, i.e. 5.8%, are greater than 10μm. The sample number distribution closely mirrors the velocity profile where, essentially, the high speed periods for the two sprays produce lower sample counts and a higher probability for larger drop sizes. The
sample number reduction correlates with the denser regions of the sprays and fewer of the smaller size classes are recorded which leads to an overestimation in the $D_{10}$ size estimates. This has been quantified, (Mojtabi 2010), for this injector and PDA system but here, where comparisons between multiple injections are being made with the 2D PDA the use of the $D_{10}$ drop size estimates is justifiable. The $D_{32}$ profile is erratic, however, this can be expected with 30μs time bins, leading to small sample numbers, and a spray that is still undergoing primary breakup. Nevertheless, the trends shown in both the $D_{10}$ and $D_{32}$ profiles are consistent for the two sprays.

Since both the axial and radial droplet velocities were measured, the vector field plots of the sprays as a function of both space and time can be generated. Such plots are shown in Figure 5-10 to highlight the flow fields existing at the interaction of spray 2 with the tail of spray 1 for the dwell times of 400 and 600μs. The central line represents the spray axis, with the vectors to the left representing the inner edge of the spray and those to the right the outer edge.

The vector plots show that for the longer dwell time, 600μs, the spray propagation as it travels downstream is similar for the first and second injection. However, for the 400μs dwell time case the tip of the second spray is wider than the first spray. This shows that the wider spray tip, detected in the near nozzle region from the shadowgraphy imaging, remains wider as it propagates downstream.
Figure 5-10 - Early spray propagation velocity vectors from PDA results

A comparison of the velocity, Figure 5-11, and \( D_{10} \), Figure 5-12, drop size profiles as a function of dwell time for the coordinates \( Z=10\text{mm} \) and \( x=10\text{mm} \) are shown. The velocity profiles for spray 1 demonstrate a very high consistency giving confidence in the comparison of the data from multiple experiments even where only very small differences are found. The first point of note is that the leading edges for spray 2 for the two shortest dwell times, 350 and 400\( \mu \text{s} \), are virtually identical even though the velocity gradient on the leading edges appears to be consistent for all four dwell times. The conclusion here is that the needle response time is limiting a correct reproduction of the requested opening response; hence most of the further analysis omits the data from the 350\( \mu \text{s} \) dwell time case.

The tip velocity of spray 2 demonstrates the reduction in velocity due to the spray flattening for the shortest two dwell times. This flattening is less apparent for the higher two dwell times but the main stream velocity is still consistent at nominally 80\( \text{m/s} \). There appears to be a quick recovery for the tip velocity profile at the 400\( \mu \text{s} \) dwell time which takes the velocity over 80\( \text{m/s} \). It is considered that this is a case where the needle lift is augmented by the small bounce. The fact that the velocity profiles at 10mm downstream are very similar for all dwell times for injection 1 and 2 shows that the nature of the steady state spray is very similar as it exits the injector, so it can be said that any differences in velocity profile observed downstream will be due to aerodynamic effects.
The analysis of $D_{10}$ profiles shows no major differences in the drop size between different dwell times for each injection, however as will be shown later there are some differences which are captured by analysis of $D_{10}$ data.

In the next section the development of the sprays downstream to 40mm for the dwell times of 400 and 600μs are presented in Figure 5-13 and Figure 5-14 for the axial velocity on the spray
axis. As the arrival at each downstream plane is retarded due to the transit time the plots have been time shifted such that the tip of spray 1 arrives at the same time. The key shows that spray 1 tip arrives at Z = 10mm at 0.54ms with the time offsets of 0.15, 0.33 and 0.48ms respectively for Z = 20, 30 and 40mm.

The downstream development of spray 1 is highly consistent for the two dwell times. The initial flow angle change in the interaction of the tip of spray 2 with spray 1 at Z = 10mm for the 400μs is not seen for the downstream measurement locations, hinting at both the combination of the two sprays and some level of entrainment on spray 2.

Spray 2, for the 400μs dwell time, does not develop a steady state profile in time, however, the maximum velocities are slightly greater, by some 5%, than spray 1. For the 600μs dwell time the spray 2 velocities are essentially the same as for spray 1. However, spray 2 develops downstream with a sharper gradient of velocity rise associated with the spray tip combined with a near flat velocity profile for the duration of the steady state injection period, albeit short due to the 500μs injection duration.

The higher maximum velocity in spray 2 for the 400μs dwell time case is attributed to entrainment on the second spray by the wake of the first spray. However, this effect is not observed for the 600μs so it is assumed that the entrainment effect is minimised in this case.

![Figure 5-13 - Velocity profiles for 400μs dwell time](image)
5.1.4 Drop Size Characteristics

The drop size profiles for 400 and 600μs, shown in Figure 5-15 and Figure 5-16 respectively, demonstrate a very high consistency in the values for spray 1 for both dwell times. The drop size on the tip of spray 1 appears to increase dramatically with distance downstream. This is a consequence of primary breakup still taking place over this distance. Most droplets detected at Z = 10 mm will be due to prompt atomisation due to the very high shear gradients between the fluid and its environment and therefore leading to smaller size classes, whereas further downstream many larger droplets are detected which were in the form of large ligaments at Z=10mm and were therefore undetectable. In addition to this, as previously discussed, the largest droplets have the highest momentum and will reach the downstream measurement locations first as they will have the lowest momentum losses due to the surrounding air.

The most notable aspect is that this mean drop size peak is not found for the tip of spray 2. Certainly, significantly more small droplets will be found on the tip of spray 2 due to the interaction with the tail of spray 1 which will be contained in any mean droplet diameter value. Sauter mean diameter calculations also support this finding. Further analysis of the discrete drop size data will determine if there is evidence of large drops occurring at the time of arrival of the tip of spray 2 for all locations downstream.
These two plots only show the differences on the spray axis, however this effect is more pronounced towards the outer edge of the spray. Figure 5-17 shows the D\textsubscript{10} drop size for the same measurement points as in the earlier vector plots for the time that the spray reaches each measurement plane, in which there is a substantial reduction in D\textsubscript{10} towards the spray periphery. From this point the results for spray 2 are all from the 400\textmu s dwell time case, as this presents
the biggest reduction in drop size. The phenomenon is still present for the other investigated cases, but is less evident. What should be noted is that the averages presented for spray 2 are mostly dominated by the small droplets from the tail of spray 1, particularly in the spray periphery; hence it is essential to investigate this phenomenon in more detail.

Figure 5-17 - \(D_{10}\) across spray at all 4 investigated planes

As discussed it appears that the droplets on the tip of spray 2 are smaller than the equivalent droplets from spray 1. But from the data presented up until this point, it would be incorrect to draw this conclusion. Although the PDA technique is highly accurate and has unparalleled repeatability it can produce misleading results without careful analysis. The fully atomised small, slow moving droplets from the tail of spray 1 which are still passing through the measurement zone at the time that the tip of spray 2 arrives create a number of difficulties with the results. Firstly the mean droplet diameters, \(D_{10}\) and \(D_{32}\), so often used to characterise sprays will produce ambiguous results as the smaller droplets from spray 1 affect this value, and secondly the transmission levels of the laser beams are lower due to the greater droplet density so care must be taken with any data presentation method.

The main issue to be addressed is to present the data in a manner which answers the question of whether the droplets in the tip of spray 2 are actually smaller than those in the tip of spray 1 but also one in which the effects of the extra droplets from the tail of spray 1 can be observed and therefore discounted in any comparison. Potential data presentation methods are summarised in Table 5-1.

Table 5-1 - Summary of potential presentation formats (chosen formats in bold)

<table>
<thead>
<tr>
<th>Method</th>
<th>Problems</th>
</tr>
</thead>
<tbody>
<tr>
<td>Any mean diameter calculation</td>
<td>Droplets from the tail of spray 1 will affect the value of mean diameter in the tip of spray 2</td>
</tr>
<tr>
<td>(D_{10})</td>
<td>(D_{10}) values are governed mainly by the many small droplets within the flow so would fail to highlight any minor effects expected due</td>
</tr>
</tbody>
</table>
Method | Problems
---|---
$D_{32}$ | $D_{32}$ values are too susceptible to the influence a small number of large droplets to calculate a time resolved average so would show some false effects. A way to combat this would be to increase the time bin size but this would reduce the temporal resolution
Compare spray 1 tail and spray 2 tip | Only possible with equal transmission, as many more droplets in small drop size classes are seen in spray 1 tail due to higher transmission levels
Calculations of $D_{v90}$, $D_{v50}$ and $D_{v10}$ | Shows some promise but droplet numbers are too low and diameter value presented is susceptible to the few larger droplets, eg. with only 20,000 droplets $D_{v90}$ is usually governed by the largest droplet
Presentation of raw discrete drop size data | Impossible to make conclusions on smaller drop size classes due to high numbers of these droplets
Histograms of drop size classes | Smaller drop size classes in histogram are dominated by droplets from spray 1 tail (simple subtraction of the droplet data from a standard spray tail from the data with a second injection during the same time span is not possible due to different levels of laser beam transmission)

The two methods highlighted in bold in Table 5-1 are the two chosen for further data presentation. Presentation of the raw discrete drop size data will show differences in the number of larger droplets within the spray. This cannot be used to look at the smaller diameter droplets, as the number of the droplets in the lower drop size areas of the diagrams makes it impossible to make any conclusions. It was determined that there were too few droplets with a diameter of over 30μm to make any conclusions about the effect of the split injection strategy on their numbers. For this reason the analysis focuses on the 0-30μm region and in particular 15-30μm. The histograms of drop size data will also show the differences in number of droplets in the larger diameter size classes but due to the remaining drops from spray 1 and the variations in efficiency of the laser beam transmission, it is impossible to make any conclusions about the effect of the split injection strategy on the number of droplets in the smaller drop size classes. If the beam transmission efficiency is reduced, the PDA technique can fail to pick up some of the droplets in the smaller drop size classes as shown by (Mojtabi 2010). This study
showed that in dense sprays, 1-D PDA gives a more accurate representation of the true numbers of droplets in each size class. This is because 1-D PDA reduces the number of beams within the system which must reach the receiver for a signal to be accepted, hence increasing the likelihood of the full range of droplets, in terms of both size and velocity, being captured.

Previous investigations have pointed out a reduction in drop size for the start of the second spray in comparison to the same period of the first spray in split injection strategies for both diesel (Wigley 1998) and gasoline (Schmid 2010) applications.

The data presented in Figure 5-18 show the raw droplet diameter data for the first 150μs of spray 1 (left) and spray 2 (right) as the spray reaches 40mm downstream of the injector. It would be potentially misleading to attempt to draw any conclusions about the smaller drop size classes, but it is clear that in the 15-30mm diameter drop size class there are far more data points recorded in spray 1 than spray 2. While there is some potential for droplets not to be included due to the previously discussed transmission issues, it appears in this instance that, in this relatively sparse region of the spray, beam transmission is not a major issue with droplet number within the tip of spray 2 more than double that of spray 1.

![Figure 5-18 - Raw drop size data in early stages of injection 1 (left) and injection 2 (right)](image)

The raw droplet data are shown both inboard and outboard of the spray axis in Figure 5-19. It can be seen in these diagrams, similar to the previous diagram, that there appears to be fewer
droplets in the 15-30μm size class in the tip of spray 2, particularly outboard of the spray axis. This is consistent throughout the whole spray stream showing that it is not the case that the droplets have merely migrated to another part of the spray stream.

Outboard of the spray axis should theoretically give the best transmission efficiencies as the beams travel through only areas of relatively low droplet density, when compared to the spray core. This therefore means that the conclusions drawn from data acquired in this region are the most reliable.

It is possible that the spray moving inboard for the second spray can partly explain the differences observed between the outboard and inboard results. However, this cannot fully explain the differences as at an axial distance of 40mm from the injector the inboard movement of spray 2 appears to be only around 1-2mm, but cannot be confirmed exclusively due to the spatial resolution of the PDA measurement grid.

The histograms of droplet size on the spray axis at an axial downstream distance of 40mm for the first 200μs of both spray 1 and spray 2 are shown in Figure 5-20. From this histogram it can be seen that there are many more small droplets passing through the measurement volume during the first 200μs of spray 2 than the equivalent time in spray 1. However, not all these droplets will be from the second injection event, and some will be droplets from the 1st injection tail which are either still progressing slowly downstream, or droplets from spray 1 which have been re-entrained into the spray. However, with the larger drop size classes (15-30μm) there is...
a definite reduction in the numbers of these occurring in the early stages of spray 2. As these larger droplets are undesirable, and hence are to be avoided, further histograms will focus only on these large drops.

Figure 5-20 - Drop size histograms for first 200μs of spray 1 and spray 2

Figure 5-21 - Drop size histograms for first 200μs of spray 1 and spray 2 with focus on larger drops
The equivalent histogram for the spray axis at 40mm downstream with focus on the larger drops is shown in Figure 5-21. This plot shows that the number of large drops is around 1.5-2 times greater in the spray 1 tip than the spray 2 tip.

Histograms were also produced for locations inboard and outboard of the spray axis to ensure this phenomenon was not just occurring locally. These are shown in Figure 5-22.

Inboard of the spray axis there appears to be a similar number of droplets within the aforementioned larger drop size classes. It also must be noted that the spray axis has moved inboard so it is possible more of the large droplets may tend towards this inner edge. Also of note is that as this is the inner edge of the spray the results are more susceptible to transmission issues if the spray density is different between the different measurement periods.

However, outboard of the spray axis there is a clear difference in the number of droplets within the larger size classes between the tips of spray 1 and spray 2. The differences of large drops between the spray 1 tip and spray 2 tip has a ratio of around 2:1. Although there are at least twice as many droplets validated by the PDA system during the spray 2 measurement time there are fewer droplets in most of the larger size classes in comparison with spray 1.
Overall the histograms presented suggest that the number of droplets in the 15-30μm drop size classes is reduced for spray 2 in comparison to spray 1 and the large droplets in the tip of spray 2 tend towards the inner edge of the spray. One further point is that the drop size data here show no evidence of droplet coalescence between the tip of the second injection and the tail of the first.

It also must be noted that although the transmission levels are lower in spray 2, the droplet number is still at least double that of spray 1. Many of these droplets are fully atomised drops from the tail of spray 1 which are interacting with the tip of spray 2, and further investigation would be required to confirm the effect this has on the appearance of the histograms.

This reduction in large droplets at a downstream distance of 40mm will be highly advantageous for engine operation as it reduces the likelihood of high levels of emissions from piston and wall impacts of large drops which lead to pool fires.

### 5.1.5 Conclusions from XL2 Investigations

A number of conclusions can be drawn from this investigation.

It was shown that the general operation of the injector during the second injection was greatly affected by the first injection. As there was a needle bounce occurring at the end of injection this affected the opening event of the second injection under various dwell times. It was also shown that the faster rise to peak current of the injector drive current related to a quicker opening event. A minimum dwell time for the solenoid operation was found to be 400μs.

Imaging and PDA results showed the spray morphology of the first and second injections was different, namely that for the shorter dwell times when the needle bounce interfered with the start of the second injection the spray width was increased. This increase in spray width occurred mainly on the inboard side of the spray and appeared to occur because of an amalgamation between the fuel from the second spray and the fuel from the needle bounce.

PDA results showed that at an axial distance of 10mm from the injector tip sprays 1 and 2 presented a similar velocity profile, however downstream of this point, there was an increase in the maximum velocity occurring in spray 2 which is due to entrainment caused by the wake of spray 1.

Finally, the investigations focussed on the differences in the numbers of large droplets produced during the early stages of injection for an injection 2 event in comparison to an injection 1. Raw droplets data and histograms of droplet diameter showed a marked reduction in the number of droplets with diameter between 15-30μm. The XL3 investigation will build on these results and explanations will be discussed later.

### 5.2 XL3 Investigations

Further investigations were undertaken with the successor to the XL2 injector, the XL3 injector. Again a special 3-hole nozzle manufactured for research purposes was used due to its advantages for optical access. This injector features, among many other improvements, a
stronger spring for the purpose of closing the needle, and thus eliminates the needle bounce. Therefore the needle opening of the second injection is as such “unaided” by any needle bounce. With this more predictable behaviour it was decided to investigate a wider variety of dwell times. The minimum dwell time investigated was 250μs as this represented the minimum response time of the solenoid. 500, 1000 and 2000μs dwell times were also investigated as they represented a good range, with the 2000μs case being the limit of dwell time which will still allow multiple injections in a single engine cycle.

There are similar trends seen with this injector in terms of both the injection drive current and the subsequent injection rate profiles as with the XL2 injector, Figure 5-23. There is an increase in the gradient of the initial rise in injection current for injection 2 in comparison to injection 1. For the longer dwell time cases the increase in the slope gradient is very small, however, for the shorter dwell time cases, 250 and 500μs, the increase, and therefore reduction in time to peak current, is much more significant. The effect on injection rate is that for the longer dwell times, there is a small advance in the start of injection timing, however, for the short dwell time cases there is both an advance in the start of injection timing and an increase in the slope of the injection rate profile during the opening phase, which represents a faster needle opening velocity.

![Figure 5-23 - Injector drive current (left) and injection rate profile (right)](image)

It must be noted at this point that the total fuel mass delivery will be different for the different dwell time cases, and this must be considered in the context of other conclusions. These differences in total fuel mass delivery must be carefully monitored in an engine to control air to fuel ratio.

5.2.1 General Spray Morphology

Again it is anticipated that the behaviour of spray 2 will be different in the far field from that of spray 1. Both far-field shadowgraphy and PDA will be utilised to characterise the overall spray morphology along with the differences in spray penetration for the different dwell time cases.
5.2.1.1 Imaging Study

The far-field imaging study is used to determine the effects of the interaction of spray 2 and spray 1. As previously stated the medium which spray 2 must propagate through is different to that which spray 1 encounters. Therefore it is likely that behaviour of the second spray will be different to that of spray 1. Qualitative analysis of the images shows a general similarity in terms of spray shape between the first and second sprays as they propagate downstream.

The first key comparison is that of overall spray penetration, Figure 5-25, and the potential of penetration reduction through the use of split injection strategies. As a reference for
comparison a single injection of 1ms will be used, which should inject a similar amount of fuel to the split injection strategies investigated. The values of penetration are calculated from a mean of 32 injections, with maximum deviation in the range ±7mm. It was shown that 16 images were sufficient to calculate penetration to an accuracy of ±0.5mm maximum from the equivalent calculation from 32 images. The plot of penetration against time shown in Figure 5-25 shows that with the shorter dwell times of 250 and 500µs the penetration is similar to that of the single 1ms injection. This is because of the entrainment effect on the 2nd injection, caused by the wake of the 1st injection. The entrainment effect reduces the shear force imparted upon the 2nd spray and hence the 2nd spray travels faster than the 1st spray. This leads to the combination of the 2 sprays in the downstream zone.

However, with the longer dwell times of 1ms and 2ms, the penetration is reduced in comparison to the single 1ms injection. This is because both the tip of spray 1 and spray 2 encounter a high level of shear resistance which reduces the spray velocity. This is due to the weaker entrainment effect.

As previously discussed, the volume of fuel injected per injection is different for the different injection strategies. Therefore, the plot in Figure 5-26 was produced, in which the penetration is normalised by the total injected mass for that injection strategy, as determined by the integral of the injection rate curves, Figure 5-23. This plot shows that with all split injection strategies there is a reduction in penetration at 4000µs AESOI, when normalised by total injected fuel mass. For the shortest dwell times, the penetration reduction is of the order of only around 2-4%. However, for the longer dwell times the reduction is larger, of the order of 5-10%.
5.2.1.2 PDA Results

In this section the PDA results will be investigated, with the differences in axial velocity given emphasis. As shown in the previous section for the shorter dwell time cases the penetration rate of spray 2 is greater than that of spray 1.

It can be seen that from Figure 5-27 (left), which displays the axial velocity at 10mm downstream of the injector on the spray axis, the velocity profile for injections 1 and 2 are very similar. This shows that the initial spray is similar for both injections in terms of velocity, which shows similar injector behaviour.

However, for the velocity profiles further downstream, 50mm, shown in Figure 5-27 (right) there are major differences between each of the two injections and for the different dwell times investigated. For the shortest dwell times the second injection displays a significantly higher velocity than the first injection, and eventually the first injection will combine with the second injection, which is seen in the imaging study.

![Z=10mm and Z=50mm Axial velocity profiles on spray axis at 10mm (left) and 50mm (right)](image)

5.2.2 Drop Size Investigations

As before, it was essential to determine if the production of large droplets in the initial stages of injection was suppressed during the second injection. 1-D PDA results were investigated for a downstream distance of 30mm to examine the numbers of large drops produced. A
downstream distance of 30mm was chosen in this case, as opposed to 40mm in the XL2 investigations, due to the penetration of this spray being lower than that of the XL2 injector.

PDA was again used to compare the drop size for the first 200µs of the second spray in comparison to the same period of the first spray for a variety of dwell times and high-zoom near-nozzle imaging will be used to explain the trends observed within the PDA results further downstream.

5.2.2.1 PDA Results

In this section the PDA results presented are from 1-D PDA. It must be remembered that at the time the tip of spray 2 passes through the PDA measurement volume, there will be residual droplets from the tail of spray 1. These droplets are of the well atomized type and have a $D_{32}$ of around 8-9µm. For this reason, it would be misleading to draw any comparisons based on the smaller drops within the spray. However, the tail of the spray has little to no large drops (>15µm), whereas the tip of a spray holds many drops in this size class. For this reason comparisons between the droplet sizes for the first and second spray will focus on the number of large drops present in the tip (first 200µs after the spray reaches the measurement volume) of each spray.

The histograms of drop size for the four dwell times investigated, shown in Figure 5-28, are taken at an axial distance of 30mm from the injector at a number of measurement points.
through the spray stream. In all cases the number of large drops is higher in the tip of spray 1 than the tip of spray 2.

The detail for droplets in the tip of spray 2 for the four dwell times is shown in Figure 5-29. This plot reveals that although the number of large drops is reduced in the tip of the 2\textsuperscript{nd} injection in comparison to the 1\textsuperscript{st} injection, the effect is more prominent with shorter dwell times. If there were to be any coalescence between the drops from the two injections this would be most prominent with the shortest dwell times. The lower numbers of large drops occurring with these dwell times shows there is very little coalescence between the two sprays.

![Figure 5-29 - Drop size histograms for different split injection strategies, comparison of first 200\(\mu\)s of injection 2](image)

It is clear that the number of large drops occurring at a position 30mm downstream of the injector is reduced in the tip of the 2\textsuperscript{nd} injection in comparison to the 1\textsuperscript{st} injection tip. With the analysis utilised up until this point it would be misleading to draw any conclusions into which physical phenomenon are responsible for this.

Although there is a reduction in the number of large drops in the tip of the 2\textsuperscript{nd} spray, the tip of any injection contains more large drops than are found in the steady state spray stream. Therefore it must be clarified how the average drop size for the duration of the spray event compares between the single 1ms injection and the split injection strategies investigated. From Figure 5-30, it can be seen that with a number of drop size characterisation techniques the overall mean drop sizes are very similar, in most cases even being smaller, which shows no downside to split injection strategies in terms of drop size. The increase for the 500\(\mu\)s dwell time for \(D_{32}\) and \(D_{90}\) is due to the presence of a small number of large droplets which dominate
these mean values. These large drops are known to occur infrequently from these injectors and their presence is not dwell time dependent.

![Bar chart showing drop size comparisons for various characterisation methods](image)

**Figure 5-30 - Drop size comparisons for various characterisation methods**

### 5.2.2.2 Near-Field Imaging

As previously stated it is clear that in the tip of the 2\textsuperscript{nd} injection in a split injection strategy the number of large drops occurring is reduced in comparison to the first injection. What must be determined is the cause of this difference in numbers of large drops. Firstly it could be due to the fact the 2\textsuperscript{nd} injection is directed into the wake of the 1\textsuperscript{st} injection leading to different break-up behaviour. Alternatively, it could be because the early spray formation caused by the faster needle rise is fundamentally different, in terms of the initial formation of ligaments and large droplets which propagate downstream, causing the large droplets at 30mm. It was considered to be unlikely that the wake effects could be responsible for the reduction in large drop formation, so it was decided to investigate the spray formation in the first few mm after nozzle exit.

The PDA technique shows poor statistical accuracy in the near field of the injector due to the high optical density of the spray in this region, even with the use of 1-D PDA, so it was essential to employ an alternative technique to capture the nature of the spray within the first few millimetres to evaluate the nature of the spray immediately upon exit of the nozzle. The near-field imaging technique was chosen as it effectively highlights the larger droplets and ligaments within the spray. This technique is only effective in relatively sparse sprays, such as those at the start and end of injection for a GDI injector. The amount of multiply scattered light from the large number of small droplets during the steady state injection period renders this technique effectively useless.
One initial observation is that although the PDA investigation revealed that differences in the numbers of large drops which were measured during the first 200µs of the two sprays, the imaging study revealed a steady-state, well atomised, spray state was achieved within 50µs from the start of visible injection, for all opening events.

However, during the first 50µs after the visible start of injection, the nature of the spray is very dependent on the needle opening time. This is shown in Figure 5-31, which displays snapshots, taken using the high-zoom Mie technique, of the spray shortly after exiting the nozzle for the different conditions. The images are processed in such a way that the large droplets and ligaments in the spray are highlighted.

The initial spray for injection 1 shows evidence of many large droplets and ligaments, particularly away from the spray axis. With the shorter dwell times of 250 and 500µs these large droplets and ligaments are not present. This is because the needle opening time is reduced and therefore the rise to steady-state injection pressure within the nozzle is reduced, which reduces needle seat throttling. For the longer dwell times of 1 and 2ms evidence of these large droplets
and ligaments exists in some injections and not in other injections. With the longer dwell times, 2 different modes exist due to variations in the needle opening transient time. Sometimes the pressure rise is relatively slow, similar to that for a standard injection, while on other occasions the pressure rise is relatively quick as is the case for an injection 2 event with a short dwell time. These findings correlate with the PDA findings at 30mm downstream, that there is a reduction in the number of large drops for all dwell times studied. This happens due to a faster needle opening event and therefore a faster pressure rise in the nozzle, which causes an improved level of atomization in the early stages of injection. The reason this effect is less pronounced for longer dwell times is because the faster pressure rise only occurs in some of the long dwell time injection 2 events and the times it does not occur the spray is more similar to that found during an injection 1. The possibility of pressure waves within the fuel lines having any effect on the initial spray formation was ruled out by installing a longer fuel line, 5m as opposed to 2m, with no effect being shown on the spray behaviour.

In addition to this, further analysis of the injection rate data reveals some minor inconsistencies in the opening period of the 2nd injection with a long dwell time, as shown in Figure 5-32 which has the injection rate during the opening period for 10 injection 2 events. It must be noted that these differences are very small, around 10μs maximum, so will not lead to any major discrepancies in fuelling. However, in this instance lead to a shift in the initial spray behaviour, transitioning between two markedly different states. This phenomenon of the variations in opening time for the 1ms and 2ms cases is as of yet unexplained, but is likely influenced by a number of factors relating to the solenoid and fuel pressure fluctuations with the injector.

![Figure 5-32 - Inconsistent injector opening timing for 2nd injection with long dwell time](image)

### 5.2.3 Conclusions from XL3 Investigations

Conclusions from the investigations featuring the XL3 injector are as follows.

Firstly, this injector showed much more predictable behaviour due to the lack of a needle bounce event at the end of injection, hence the start of the second injection was more regular. The reduced time for the needle opening phase was again observed due to the eddy currents remaining in the solenoid from the first injection.
Far-field imaging showed the propagation of the sprays as they travelled downstream and from this the overall spray penetration was measured, which showed a slight reduction in penetration with short dwell times and a more significant reduction for longer dwell times.

Similar to the results for the XL2 injector, there was a clear reduction in the number of large drops produced in the early stages of injection for all dwell times investigated. PDA results showed this reduction was most pronounced for the shorter dwell times. High-zoom Mie-imaging was used to determine the nature of the spray during the early stages of injection. This revealed the presence of a large number of large droplets and ligaments for the start of injection 1, which were not present in the early stages of injection 2 for the short dwell time cases. However, for the longer dwell time cases this removal of large droplets and ligaments occurs in some injections but not in others and this is assumed to be the reason for the less pronounced reduction in numbers of large droplets measured downstream by the PDA technique.

5.3 Conclusions

A combination of the results from the investigations with both the XL2 and XL3 investigations lead to a number of interesting conclusions about the use of split injection strategies with GDI engines and GDI solenoid driven injectors.

Firstly, the importance of understanding the injector behaviour when the gap between successive injections is very short was shown. GDI engines require accurate fuel metering and therefore it is imperative to fully understand how the use of split injection strategies affects not only the morphology of the spray, but also the mass of fuel it delivers into the cylinder. Therefore it is advised to only use injectors in which the needle bounce has been suppressed for split injections.

For all cases, the rise to peak injector drive current, relative to the start of injection trigger, was reduced for an injection 2 in comparison to an injection 1. This is due to eddy currents which remain within the solenoid from the first injection aiding the current rise for the second injection. This leads to a quicker opening event, as confirmed by the rate tube study, and therefore a reduced time spent with the needle seat gap having a throttling effect. This quicker injector opening led to significant differences in total fuel mass delivery for different split injection strategies. For modern GDI engines operating under homogeneous conditions it is imperative to maintain correct air to fuel ratios, thus careful monitoring of the injected mass is essential in any engine operating under split injection strategies.

Both imaging and PDA results showed the differences in spray propagation in both the near and far field of the injector. Due to interactions with the needle bounce, for the second injection with the XL2 injector the spray proved to be wider than the first injection, however, with the more regular behaviour of the XL3 injector the spray morphology was more regular. Penetration was tracked as the sprays progressed downstream, and it was shown that for all dwell times investigated there was a reduction in spray penetration compared to an equivalent fuel mass single injection, which is an essential objective of split injections. However, this reduction is only
of the order of 5-10% as there is a strong entrainment effect on the second spray caused by the wake of the first spray.

Finally, significant analysis was undertaken on the nature of the initial stages of the second injection. It was found, using a high-zoom Mie-imaging technique, that the number of large drops and ligaments produced during this phase is reduced significantly in comparison to the equivalent phase of the first injection. PDA results confirmed that this lead to a lower number of these droplets in the downstream region, which also shows that levels of coalescence between the two injections is at very low levels. The reason for the suppression of the production of these large droplets and ligaments is due to the reduced time for the needle to fully open, and therefore the reduced time in which the needle seat gap has a throttling effect, as this produces unfavourable conditions within the nozzle holes for efficient atomisation.
6 Results – Flash Boiling

This section will deal with the results of an investigation into the behaviour of multi-hole GDI injectors under various injector temperature and ambient air pressure conditions. The aim of these variations in conditions is to induce flash boiling conditions and both qualify and quantify changes in injector behaviour, spray propagation and total spray break-up which occur with the addition of the flashing phenomenon to the spray process.

6.1 Test Points

The objective of this work was to test over a range of injector temperatures and ambient air pressures which are representative of those occurring in a GDI engine. The investigation used a number of single component fuels suitable for modelling in follow up work in collaboration with Continental Automotive (the sponsors of this work). Single component fuels were chosen to remove the additional complexities provided by a multi-component fuel under superheated conditions. It was also of significant interest to have test points with an equal superheat degree (SD), but different temperatures and pressures.

This investigation used two single component test fuels which were representative of gasoline fuel: n-heptane and iso-octane. For the n-heptane testing there were 6 test points as shown in Figure 6-1, which includes the saturation pressure curve for the fuel. Three of these test points were also repeated with iso-octane fuel, as in Figure 6-2, with their relation to the saturation pressure curve for the fuel shown. These two fuels were chosen as they have similar boiling point curves in addition to similar break-up characteristics, so should have similar spray morphology and atomisation under superheated conditions.

![Figure 6-1 - Test points for n-heptane fuel](image)
A full list of the test points is documented in Table 6-1. For the close to ambient conditions (P=0.9bar T=23°C), the superheat is sufficiently negative that superheat effects can be neglected.

Table 6-1 - Test points for flash boiling testing

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Ambient air pressure (bar)</th>
<th>Injector temperature (°C)</th>
<th>Superheat (SD) (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>n-heptane</td>
<td>0.9</td>
<td>23</td>
<td>-75</td>
</tr>
<tr>
<td>n-heptane</td>
<td>0.1</td>
<td>23</td>
<td>-12</td>
</tr>
<tr>
<td>n-heptane</td>
<td>0.3</td>
<td>50</td>
<td>-12</td>
</tr>
<tr>
<td>n-heptane</td>
<td>0.1</td>
<td>47</td>
<td>12</td>
</tr>
<tr>
<td>n-heptane</td>
<td>0.3</td>
<td>74</td>
<td>12</td>
</tr>
<tr>
<td>n-heptane</td>
<td>0.1</td>
<td>71</td>
<td>36</td>
</tr>
<tr>
<td>iso-octane</td>
<td>0.1</td>
<td>23</td>
<td>-12</td>
</tr>
<tr>
<td>iso-octane</td>
<td>0.1</td>
<td>47</td>
<td>12</td>
</tr>
<tr>
<td>iso-octane</td>
<td>0.1</td>
<td>71</td>
<td>36</td>
</tr>
</tbody>
</table>

Previous investigations had revealed vastly different spray collapsing behaviour between injectors with different cone angles (Mojtabi 2011) (Marriott 2008) (Zhang 2011 (b)) (Xu 2013). For this reason it was determined that this investigation should use an injector with a cone angle which was representative of a large majority of production injectors. The injector chosen was therefore a Continental XL3 6-hole 60° cone angle injector. Injection duration was set at 2ms, which was considered sufficient that any transient behaviour (for example cooling of the air charge by the early stages of injection thus having an effect on the later stages of injection) would not be overlooked.
PDA testing was conducted at two circumferential locations around the injector centre separated by 30°. This gave information in the main and the interstitial streams. However, due to a terminal injector malfunction, data were only captured for the interstitial stream for the final test point in Table 6-1. Data were acquired on a plane 40mm downstream of the injector tip at a number of radial positions. The test positions are shown in Figure 6-12. The 40mm measurement plane was chosen as it was in a region of sufficiently low optical density that laser beam transmission efficiency would not adversely affect results, but sufficiently close to the injector that the majority of the spray would pass beyond.

6.2 Efficacy of 70° Test Rig

The high pressure/high temperature cell featuring 70° windows, from which many of the results in this section are obtained, was manufactured specifically for this work so it was first necessary to determine the validate its conducting PDA measurements, and in particular the effect of the thick windows on the results obtained.

It was previously shown that when conducting PDA analysis within a chamber, it is possible to not resolve all of the smaller droplets in the spray (Aleiferis 2013). This is because these droplets reflect the least amount of light energy towards the receiver, and when this low light level is coupled with the energy losses as the transmitted signal passes through the thick window, the likelihood of sufficient light energy being detected at the receiver is reduced in comparison to an experiment without optical windows.

The plot of the raw data for the case of atmospheric temperature and pressure, Figure 6-3, firstly on the atmospheric rig (left) and secondly in the new spray chamber (right) shows that some smaller droplets have been missed.

![Figure 6-3 - Raw drop size data without (left) and with (right) optical windows](image)

The effect on the D_{10} and D_{32} profiles is shown in Figure 6-4, with the time span of 0.5-3ms being that in which the main body of the spray passes through the measurement volume, (a 2\textsuperscript{nd} measurement is included from inside the chamber to show the high levels of repeatability). These plots show that, likewise, a number of larger droplets have failed to be recorded with the windows present as shown by the similarity in the temporal mean values, thus showing the rig to display acceptable levels of accuracy in terms of recording a sufficient number of droplets. It
should be noted that the spikes in the D32 profile are due to the short time bins used in this particular analysis, and a single or possibly two large droplets producing a misleading mean value.

![Graphs of D10 and D32 profiles for with and without optical windows](image)

The equivalent data for axial velocity are shown in Figure 6-5, which again shows that although the number of droplets detected is reduced the profile remains similar for both cases.

![Graphs of raw droplet axial velocity data without (left) and with (right) optical windows](image)

The effect on the mean axial velocity profile is shown in Figure 6-6. In the main body of the spray, there is a clear reduction in the recorded mean velocity. This signifies that a number of the fastest moving droplets fail to be recorded. The general profile however remains very similar, and in the stages of injection where the spray density is lower the profiles match very closely. What is noted is that the peaks and troughs of the velocity profile occur at the same time, thus showing the ability of the results to represent major changes in the spray velocity. It would, however, be misleading to believe the exact values of mean velocity to be truly representative.
Figure 6-6 - Mean axial velocity profile for with and without optical windows

It must be noted that the plots shown here were produced from results taken during the final commissioning of the rig. As testing and alignment procedures were optimised further improvements in the droplet numbers, and therefore the reliability and repeatability of the results, was achieved.

6.3 Changes in Overall Spray Morphology

From the previous research into flashing multi-hole GDI sprays discussed in section 3.2.3.4, it is clear that when superheat is applied to the fuel there will be a substantial change in the overall spray morphology. A number of processes involved in the propagation of a flashing multi-hole spray can be characterised through the use of various imaging techniques; shadowgraphy, global Mie and planar Mie, along with 2-D PDA data.

6.3.1 Imaging

The images acquired in this section are obtained in the spray chamber detailed in section 4.2.7.1. Images were obtained using three methods. The shadowgraphy technique was used for global spray propagation in the injector plane. Global Mie-imaging was used to track global spray dispersion in the transverse plane. Finally, planar-Mie was used for tracking the spray footprint as the spray passed through a plane 40mm downstream of the injector tip, which was also the plane investigated by the PDA technique. Example images for each of the three imaging techniques are shown in Figure 6-7. All images shown are single-shot images. In the remainder of this results section presented images are chosen as they are deemed to be the best images to represent a significantly larger set of images, 32 images, taken at that condition. Three images are shown at the same operating condition in Figure 6-7 to highlight the repeatability of the measurement techniques. The thin line which appears across these images is a thermocouple wire. Care was taken when assembling the rig to ensure the thermocouple wire did not interfere with the spray. Unless otherwise stated, images are taken at 1100μs after electronic start of injection (AESOI), as this was deemed to be an appropriate time for comparison as, for all cases,
the spray had penetrated through the majority of the chamber, but not impacted upon the chamber walls or the optical windows.

<table>
<thead>
<tr>
<th>Shadowgraphy</th>
<th>Global Mie</th>
<th>Planar Mie @ Z=40mm</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1.png" alt="Image 1" /></td>
<td><img src="image2.png" alt="Image 2" /></td>
<td><img src="image3.png" alt="Image 3" /></td>
</tr>
<tr>
<td><img src="image4.png" alt="Image 4" /></td>
<td><img src="image5.png" alt="Image 5" /></td>
<td><img src="image6.png" alt="Image 6" /></td>
</tr>
<tr>
<td><img src="image7.png" alt="Image 7" /></td>
<td><img src="image8.png" alt="Image 8" /></td>
<td><img src="image9.png" alt="Image 9" /></td>
</tr>
</tbody>
</table>

6.3.1.1 **Shadowgraphy**

The results obtained within the 90° spray chamber are presented in Figure 6-8.

In the close to ambient conditions case the 3 streams (the other 3 streams are hidden behind the visible streams) can be clearly seen and distinguished.

For the marginally negative superheat cases (SD=-12°C) the overall spray pattern appears to be similar to that for the ambient case, however, the spray seems more disperse (due to the higher amount of light energy passing through the spray and reaching the camera). There is a very small increase in penetration, particularly for the lower pressure case, which in this instance can be attributed to the reduced air resistance encountered by the spray due to the reduced
ambient pressure within the chamber. The differences between these two cases with an equal superheat will be discussed later.

In the two cases which represent a small degree of superheat (SD=12°C) there are a few additional differences observed. Firstly the spray cone angle is reduced slightly which suggests there is a low pressure zone forming in the centre of the spray cone which is causing the spray

![Image of shadowgraphy results](image-url)
to collapse. Secondly there is some evidence of fuel being entrained into recirculating vortices on the spray tip (most clear for the P=0.3 bar T=74°C case), which suggests flash boiling is taking place and leading to the creation of many small drops which are easily entrained into these vortices. In these cases the penetration is similar to that of the ambient conditions case. The reduced back pressure should lead to an increased penetration, but in these cases this is opposed by the smaller drops created by flash boiling having reduced momentum due to their reduced mass.

Finally, for the case with significant superheat (SD=36°C) there is a clear spray collapse, with the formation of a tulip shaped spray which has been observed previously (Mojtabi 2011), (Zhang 2012). Many droplets are entrained into recirculating tip vortices, which suggests a significant amount of flash boiling has occurred promoting the production of a large number of small droplets. The axial penetration is increased as the spray collapses into the core and effectively forms a single spray stream, thus with increased momentum. Flare flashing is defined as the complete collapse of the spray along with the formation of interstitial streams and disappearance of the original spray streams. The superheat degree required for this to occur in this investigation is shown to be lower than some other researchers have observed (Zhang 2012), (Zeng 2012 (b)), however, is in line with others (Marriott 2008). The published results which this work is in agreement with are for similar cone angle injectors, but the data which it is not in agreement with are on much wider cone angle injectors. This shows the importance of the injector cone angle and therefore levels of stream to stream interaction and entrainment on the likelihood of occurrence of spray collapse and flare flashing.

### 6.3.1.2 Global Mie

Global Mie-imaging results follow a similar pattern to those obtained through shadowgraphy and are summarised in Figure 6-9.

For the non-flashing case the 6 individual spray streams are easily distinguished. For the slightly negative superheat cases (SD=−12°C) there is a widening of the individual spray plumes, but not sufficient to promote strong stream to stream interactions.

For the slightly superheated cases (SD=12°C) there is clearly a further increase in stream to stream interaction. This leads to more fuel being present in the central region between the spray streams, while some fuel is maintained in the original streams. It appears that at this superheat level there is some flash boiling which leads to a widening of the spray streams and therefore an increased stream to stream interaction. It is believed this increased interaction leads towards the formation of a fuel barrier between the inside and outside of the spray cone, and therefore the increased likelihood of entrainment of droplets into the spray core.

For the highest superheat case (SD=36°C), which has already been identified as undergoing significant flash boiling, the existence of fuel in the centre of the spray and in the gaps between the original spray streams, the interstitial streams, can clearly be seen. These results effectively show that fuel has migrated away from all areas where it would be located under normal conditions to all areas within the spray cone where it would not exist under normal conditions.
These results suggest that the spray collapse and formation of a tulip shape spray is interlinked to the formation of interstitial spray streams, therefore promoting the idea that the spray collapse is responsible for the formation of interstitial streams.

Again the results show that the value of superheat required for formation of interstitial streams is in line with published results from narrow cone angle GDI injectors (Marriott 2008), but not wide cone angle injectors (Zhang 2012), (Zeng 2012 (b)).
6.3.1.3 Planar Mie @ Z=40mm

Planar Mie-images, Figure 6-10, build on the global Mie images to show detail of the spray morphology. For the first three test points the planar Mie results show the same trends as for the global Mie, namely the widening of individual spray plumes.

<table>
<thead>
<tr>
<th>P=0.9 bar T=23°C (SD=-75)</th>
<th>P=0.1 bar T=23°C (SD=-12)</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1.png" alt="Image" /></td>
<td><img src="image2.png" alt="Image" /></td>
</tr>
<tr>
<td>P=0.3 bar T=50°C (SD=-12)</td>
<td>P=0.1 bar T=47°C (SD=12)</td>
</tr>
<tr>
<td><img src="image3.png" alt="Image" /></td>
<td><img src="image4.png" alt="Image" /></td>
</tr>
<tr>
<td>P=0.3 bar T=74°C (SD=12)</td>
<td>P=0.9 bar T=71°C (SD=36)</td>
</tr>
<tr>
<td><img src="image5.png" alt="Image" /></td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
</tbody>
</table>

Figure 6-10 - Planar Mie results @ Z=40mm and @ t=1100μs AESOI

However, for the superheated cases some interesting features are highlighted by the planar Mie testing. The two cases with superheat=12°C show evidence of stream to stream interaction, and exhibit a completely different spray pattern as a result. Images acquired at a later time indicate both conditions induce a similar spray collapse as a result of the stream to stream interactions and subsequent formation of an effective fuel barrier between the outside and inside of the spray cone. These results will be dealt with in more detail in Section 6.4.
Finally the SD=36°C case shows a similar pattern as observed with global Mie, but the concentration of fuel in the central region of the spray is shown to be more pronounced from these images.

According to transition criteria (Kitamura 1986) there is a relationship between Jakob number and Weber number which controls the point at which flash boiling break-up becomes dominant over mechanical break up. The Jakob and Weber numbers have been calculated such that superheat required for transition can be calculated using the parameters in Table 6-2. This calculation leads to a Weber number of 10.3 and a Jakob number of 1.17Xsuperheat degree. With reference back to Figure 3-20, for a Weber number of 10.3 the required Jakob number for transition is of the order 80-100. Thus the required superheat in this case is in the range 65-85°C according to this correlation. While this correlation has been proven for jets, it is not yet clear whether it will hold true for a spray produced by a multi-hole injector. The results presented here suggest that for a superheat of 36°C the spray is significantly different, and beyond the transition criteria. This is likely evidence of the effects of multiple spray streams in a small area on each other leading towards flare flashing at a lower Jakob number than an equivalent single jet.

<table>
<thead>
<tr>
<th>n-heptane value</th>
<th>n-heptane value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid density (kg/m³)</td>
<td>682</td>
</tr>
<tr>
<td>Surface tension (mN/m)</td>
<td>20.14</td>
</tr>
<tr>
<td>Gas density (kg/m³)</td>
<td>4.15</td>
</tr>
<tr>
<td>Specific heat (J/Kmol)</td>
<td>224.64</td>
</tr>
<tr>
<td>Latent heat of vaporisation (kJ/mol)</td>
<td>31.8</td>
</tr>
<tr>
<td>Velocity (m/s)</td>
<td>100</td>
</tr>
<tr>
<td>Droplet diameter (μm)</td>
<td>5</td>
</tr>
</tbody>
</table>

### 6.3.1.4 Further Planar Mie

Planar Mie results were also taken at a number of downstream distances (5, 10, 20 and 40mm) for the flare flashing case. The images, which are shown in Figure 6-11, show that in the near vicinity of the injector, 5 and 10mm downstream, both mains and sparse interstitial streams still exist. However, by the time the spray has reached the downstream measurement locations, 20 and 40mm downstream, the mains streams have disintegrated completely and fuel now exists in the collapsed spray core and the wide interstitial streams.
It can therefore be assumed that for this case the bubble growth and droplet shattering process which occurs with flash boiling will be occurring mainly in the first 10mm of the spray path. This corresponds to a time span of around 0.1-0.2ms. These values agree with the earlier quoted Razzaghi model (Razzaghi 1989), which estimates bubble growth time to be in the order of 0.02-1ms, with an increase in superheat correlating to a reduction in growth time.

6.3.2 PDA Data

As previously mentioned 2-D phase Doppler data were gathered on a plane 40mm downstream of the injector tip, through both the original stream position, which will be referred to as the main stream, and through the gap between a pair of mains streams in which new spray streams are expected to form under significant superheat conditions, which will be referred to as the interstitial streams. Data are presented in the form of moving average plots of axial and radial
velocity. The full data set in terms of axial and radial velocity is included in the appendices in Section 9.1.

The PDA measurement locations are shown by the red dots in Figure 6-12. Although for this test condition (close to atmospheric test) the measurement points only just extend to the outer edge of the spray, for the flashing cases with a collapsed spray these measurement points were found to be easily sufficient. The measurement points are shown in tabular form in Table 6-3.

![Figure 6-12 - PDA measurement locations](image)

<table>
<thead>
<tr>
<th>Downstream distance (mm)</th>
<th>40</th>
<th>40</th>
<th>40</th>
<th>40</th>
<th>40</th>
<th>40</th>
<th>40</th>
<th>40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial distance (mm)</td>
<td>0</td>
<td>2.5</td>
<td>5</td>
<td>7.5</td>
<td>10</td>
<td>12.5</td>
<td>15</td>
<td>17.5</td>
</tr>
</tbody>
</table>

The main characteristic of the axial velocity profile for a flashing spray, the mains spray axial velocity profile is shown in Figure 6-13, is the increased velocity in the spray core locations, and a reduction in velocity levels as the radial distance from the injector axis is increased. For the measurement points on the edge of the spray there is a period soon after the spray reaches the
measurement plane where the axial spray velocity exhibits flow back towards the injector. This shows evidence of recirculating tip vortices passing through the measurement volume at this time.

Another feature is high levels of similarity between the mains and interstitial streams. This is because of the high level of droplet entrainment into all areas of the spray. The mean droplet velocity at the eight measurement locations in the interstitial streams is shown in Figure 6-14.
The radial velocity profile displays some very interesting characteristics, Figure 6-15. The start of injection is associated with a large number of droplets having a relatively large positive axial
velocity. However, this is quickly followed by a significant drop in velocity to negative levels. This is due to a recirculating tip vortex passing through the measurement volume during this time period, which is also highlighted by a negative axial velocity at this time. After this time, the vortex’s effects are still evident as the spray continues to collapse inwards, albeit with a much reduced force.

The nature of this vortex is shown in more detail in Figure 6-16. This plot is of the raw droplet data, with each dot on the plot representing a droplet which was measured by the phase Doppler system. However, the medium and large drops (>4µm) are removed from this plot and only the small drops (<4µm) are shown. As the size of a droplet is roughly related to its momentum and therefore inversely related to the droplets ease of entrainment, the small drops will highlight the effect of the air motion on the spray.

In this plot the strong vortices causing negative velocities around 0.25-0.5ms after the spray has initially reached the measurement volume are shown, and the residual effect on the continued collapse can be seen throughout the remainder of the injection period.

The test points with reduced superheat also show some interesting characteristics. The imaging study revealed that these test points display increased levels of spray dispersion and increased stream to stream interaction. The effect of this on the axial velocity profile for a test point with superheat degree=-12°C is shown in Figure 6-17. This plot shows that the highest initial velocities are towards the outer edge of the spray, in the locations which would be expected for a spray into atmospheric conditions. However, as the injection progresses the axial velocity in
the centre of the spray increases, from a value of close to zero for the first half of the spray, to a value of around 60 m/s.

Figure 6-17 - Axial velocity profile for partially flashing spray (P=0.1 bar T=23°C) in mains stream

Figure 6-18 - Radial velocity profile for partially flashing spray (P=0.1 bar T=23°C) in mains stream
The corresponding plot of radial velocity, Figure 6-18, shows the reason for this increase in velocity in the spray core region. The reduction in radial velocity which occurs as the spray progresses shows evidence of a weak spray collapse, which will increase the likelihood of droplet entrainment into the spray core. As more droplets converge in this area, their collective momentum will increase, thus increasing their ability to penetrate through the air resistance. One thing to note is that for this condition the spray collapse occurs much later than for the flare flashing conditions (P=0.1bar T=71°C). This is because with the lower superheat degree there are a lower number of the easily entrained droplets produced through flash boiling.

For the cases with medium superheat (SD=12°C) the PDA results indicate that the spray behaves in a manner in between the two extreme cases shown here. There is a moderate strength spray collapse which occurs at a time intermediate to the two cases shown. The effects of the spray collapse continue to be seen throughout the remainder of the injection period. There is some evidence of droplets being entrained into the recirculating tip vortices, but the velocity changes are not as sudden as for the flare flashing case, showing the vortices have less strength at these conditions. The spray collapse again leads towards increased axial velocity in the spray core region.

6.3.2.1 Differences between Main and Interstitial Streams under Flashing Conditions

The formation of interstitial streams and disappearance of mains streams under flash boiling conditions for multi-hole GDI sprays has been observed by many authors (Dahlander 2006), (Mojtabi 2011), (Zhang 2012), but the mechanisms behind this dramatic change to the spray morphology have yet to be fully understood. The previous imaging results show that for the highest superheat test point (SD=36°C) there is a clear spray collapse together with formation of interstitial streams. Hence, the PDA data from this point will be examined in detail along with some additional planar Mie-images which were acquired from a number of axial planes downstream of the injector tip.

The imaging study, from Figure 6-11, presented that there was a clear spray collapse towards the spray core and PDA results from this region reveal the behaviour in this area to be stable through the circumferential direction, thus showing the fuel in this area is mainly influenced by its entrainment into this area rather than any influence remaining from its injection into the chamber.

Therefore, to find information about the formation of interstitial streams it is essential to examine data points from outside of this core. The radial velocity profile for both the mains and interstitial streams at the measurement point 7.5mm radially from the injector axis are shown in Figure 6-19. This point was chosen as it represents both the most marked difference in velocity profile, and a position where the imaging study revealed there should be significant differences. The plot reveals a very similar profile for the majority of the injection period; however, between 1-1.5 ms there is a clear difference with the mains streams presenting a significant negative value during this period, representing an increased level of collapsing in this region of the spray, as opposed to the region in which the interstitial streams exist.
The basic theory of flash boiling is that when droplets have undergone flash boiling they will form a large number of daughter droplets which consequently have very low momentum, hence would be easily entrained. Therefore if the interstitial streams are formed of droplets which have already undergone flash boiling in the near-nozzle region and the mains streams are formed of a combination of droplets which have already undergone flash boiling in addition to some which are still to undergo a flashing event, this could explain the differential entrainment. Those drops in the interstitial streams will have had some momentum gain as they travel downstream; whereas once a droplet in the mains stream undergoes flash boiling it will have a sudden loss of momentum, and therefore an increased likelihood of entrainment away from this region. This description correlates well with the planar Mie-imaging on a number of downstream planes, Figure 6-11, which shows fuel to migrate away from the mains streams as the spray progresses downstream.

For this theory to be plausible there must be a corresponding reduction in axial velocity during the same time period. This is confirmed by the plot in Figure 6-20. It is anticipated that if this were to be correct there would also be significant motion from the mains streams to the interstitial streams during this period. However, this is impossible to characterise with the phase Doppler system in its current configuration, but it is suggested that any future work in this area using the phase Doppler technique should investigate the cross plane motion.

The patterns shown here were repeated for the measurement points from 5mm for the injector axis outwards, effectively the area from which the spray collapses into the core region.

![Figure 6-19 - Radial velocity @7.5mm from injector axis for P=0.1bar T=71°C](image)

The patterns shown here were repeated for the measurement points from 5mm for the injector axis outwards, effectively the area from which the spray collapses into the core region.
As a further addition to this theory, where superheat is sufficiently great as to produce a second flashing event, meaning the small daughter droplets produced by flash boiling can undergo bubble nucleation and growth again, the formation of new spray streams in the gaps between the interstitial streams (the original location of the mains streams) has previously been observed (Mojtabi 2011).

![Figure 6-20 - Axial velocity @7.5mm from injector axis for P=0.1bar T=71°C](image)

**6.4 Comparison between Test Points with Equal Superheat**

Under the vast array of conditions into which GDI sprays are injected in an engine, there will be a number of competing drivers for the spray development. Firstly, the spray break-up due to surface instabilities imparted upon the spray in both the injector nozzle and as it propagates downstream will affect spray break-up. However, there will also be changes to the break-up process due to the ambient pressure environment and due to the heated fuel. The results from test points with equal superheat aim to detail how suitable the parameter of superheat is to describe the changes to the spray development, or whether it is the case that superheat is supplemented by other changes to the environment and the description of how the conditions affect the spray is in fact a combination of a number of factors.

It was noted previously that the spray patterns for points with equal superheat showed a great degree of variation, particularly for a superheat of 12°C.
6.4.1 Superheat Degree=-12°C

Analysis of the shadowgraphy images, Figure 6-21, shows a similar spray pattern in terms of cone angle, however, the spray appears to be more dispersed for the lower pressure case. Reasons for this are unknown, but it can be speculated that the lower surrounding pressure may allow a wider range of droplet trajectories for any droplets which have undergone flash boiling. Another difference can be observed in the spray immediately upon exiting the nozzle. For the lower pressure case there appear to waves forming through the spray, which are possibly aiding the dispersion process.

<table>
<thead>
<tr>
<th>P=0.1 bar T=23°C (SD=-12)</th>
<th>P=0.3 bar T=50°C (SD=-12)</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1.png" alt="Shadowgraphy images for SD=-12°C" /></td>
<td><img src="image2.png" alt="Shadowgraphy images for SD=-12°C" /></td>
</tr>
</tbody>
</table>

Analysis of the planar Mie-images for different time steps during the injection, Figure 6-22, complement the findings of the shadowgraphy results. From the moment the spray reaches the measurement plane, at 40mm below the injector tip, the spray from the lower ambient pressure case appears to be more disperse, thus promoting a higher level of stream to stream interaction and therefore higher mixing rates. However, as the injection progresses with time the overall shape of the sprays, if not the concentration of fuel, becomes more similar, and by 1500µs AESOI the spray shapes appear to be very similar.

Therefore, it is assumed that the overall motion field (both air and fuel motion) is very similar for both these cases. It is also noted that for similar flow fields to produce a similar overall spray pattern a similar drop size distribution would be required. This is because drops of different sizes have vastly different levels of momentum, and therefore a different likelihood of entrainment. This suggests that for these cases the overall level of flash boiling is similar (this can be confirmed or disproved through analysis of the drop size data).

It can, however, not be discounted that the initial spray behaviour for these two cases appears to be slightly different. One suggestion is that in the short time before flash boiling occurs the spray morphology is controlled by the ambient pressure conditions. However, later spray
behaviour is injected into a flow field already developed through the flash boiling process, therefore tending towards a common spray pattern.

<table>
<thead>
<tr>
<th></th>
<th>$P=0.1 \text{ bar \ } T=23^\circ \text{C (SD=-12)}$</th>
<th>$P=0.3 \text{ bar \ } T=50^\circ \text{C (SD=-12)}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$t=1000\mu s$</td>
<td><img src="image1.png" alt="Image" /></td>
<td><img src="image2.png" alt="Image" /></td>
</tr>
<tr>
<td>$t=1100\mu s$</td>
<td><img src="image3.png" alt="Image" /></td>
<td><img src="image4.png" alt="Image" /></td>
</tr>
<tr>
<td>$t=1200\mu s$</td>
<td><img src="image5.png" alt="Image" /></td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
<tr>
<td>$t=1300\mu s$</td>
<td><img src="image7.png" alt="Image" /></td>
<td><img src="image8.png" alt="Image" /></td>
</tr>
</tbody>
</table>
Further analysis into these slight differences can be conducted with the phase Doppler results, as this allows effective tracking of the spray behaviour through the whole of the spray period.
The plot of radial velocity on the inner edge of the mains spray for the two cases, Figure 6-23, reveals that in this area the sprays are very similar.

However, on the outer edge of the spray, Figure 6-24, it can be seen that the spray collapses earlier for the 0.3 bar 50°C case. Although the evidence of this only occurs at a time of around 1.5-2ms AESOI, it must be noted that this is the behaviour at 40mm below the injector tip, and this will occur at an earlier time in the more immediate vicinity of the injector. This explains the increased spray density towards the spray core for this case over the 0.1 bar 23°C case which was noted from the imaging study.

![Figure 6-24 - Radial velocity on outer edge of spray @15mm](image)

### 6.4.2 Superheat Degree=12°C

Again for these cases the trends observed through shadowgraphy, Figure 6-25, appear very similar to the SD=12°C cases. The increased spray dispersion and formation of surface waves upon exit from the injector are still present for the lower pressure case, while the cone angle for both sprays appears to be very similar.

However, for the higher pressure case there is some evidence of a weak spray collapse behind the spray tip and also evidence of a small amount of entrainment of droplets into recirculating tip vortices. These features may well be present for the lower pressure case, but are obscured by the increased level of spray dispersion. There is a reduction in penetration for the higher pressure case, which could be somewhat due to the higher density medium through which it must pass, but also the tip vortices could in this case be hindering penetration.
The Mie-images for these cases display a number of interesting features, as shown in Figure 6-26. Firstly, the initial fuel which reaches the measurement volume at \(Z=40\text{mm}\) is vastly different for the two cases. For the lower pressure case the fuel forms a ring-like structure, whereas for the higher pressure case the six original spray streams still exist but fuel has been transported into the spray core.

These sprays should exhibit a moderate level of flash boiling and it is believed that the different spray shapes are due to the differential entrainment of fuel which has already undergone flash boiling. For the low pressure case \((P=0.1\text{bar})\) the droplets tend towards the gaps between the original spray streams, whereas for the higher pressure case they tend towards the spray core. Further analysis, possibly using the PIV technique, is required to fully understand the reasons for these different spray shapes. However, it is speculated that for the 0.3 bar case there is a greater pressure differential between the spray core and the area outside of the spray which is driving entrainment of the flashed droplets into the spray core.

As with the previous data set, later in the spray period the two cases converge to a similar result. This provides further backing to the belief that the dispersion of the later fuel emitted from the injector is heavily dependent on the initial flow field it encounters, whereas the initial fuel encounters static air, and the differences between the conditions investigated are due to the different ambient pressure conditions within the chamber.
<table>
<thead>
<tr>
<th>$t$ (μs)</th>
<th>$P=0.1$ bar, $T=47^\circ$C (SD=12)</th>
<th>$P=0.3$ bar, $T=74^\circ$C (SD=12)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td><img src="image1.png" alt="Image" /></td>
<td><img src="image2.png" alt="Image" /></td>
</tr>
<tr>
<td>1100</td>
<td><img src="image3.png" alt="Image" /></td>
<td><img src="image4.png" alt="Image" /></td>
</tr>
<tr>
<td>1200</td>
<td><img src="image5.png" alt="Image" /></td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
<tr>
<td>1300</td>
<td><img src="image7.png" alt="Image" /></td>
<td><img src="image8.png" alt="Image" /></td>
</tr>
</tbody>
</table>
The phase Doppler data can also provide an insight into the differences in terms of behaviour of the sprays at these two conditions. Data are presented from a number of radial measurement points (5, 7.5, 10 and 12.5 mm from the injector axis at 40mm downstream).

![Planar Mie images for SD=12°C](image1)

![Radial velocity in mains streams for P=0.1 bar T=47°C](image2)
The radial velocity profiles for the mains streams, presented in Figure 6-27 and Figure 6-28, show a generally similar profile. However, at the time which has been shown to be key to the spray collapse, around 1.3-1.8ms for this case, which is different for each condition due to different levels of flash boiling, the radial velocity is slightly higher for the 0.1 bar case. This explains the marginally larger dense spray region in the core during the later stages of injection observed in the planar Mie images.

![Radial velocity in mains streams](image)

**Figure 6-28 - Radial velocity in mains streams for P=0.3bar T=74°C**

Analysis of the velocity profiles in the interstitial streams, Figure 6-29 and Figure 6-30, shows a more pronounced difference. During the collapse period, the velocity profile for the 0.1 bar case is consistently positive, away from the injector axis. For the 0.3 bar case the profile is markedly different. For all four measurement points shown, there is a clear collapse of the spray. This explains that for the 0.1 bar case the fuel which has undergone flashing will tend towards the gaps between the original spray streams, whereas for the 0.3 bar case the same droplets are entrained into the spray core, even if they do initially tend towards the interstitial streams after undergoing flash boiling.
6.5 Drop Size Reduction in Flashing Sprays

It is well documented that a drop size reduction should be expected with flash boiling sprays (Dahlander 2006). However, it is necessary to quantify the reduction which can be expected for superheated multi-hole sprays. The drop size results will also provide insight into the test points...
which exhibited a weak spray collapse under non-superheated conditions, as any drop size reduction could be due to partial local flashing, possibly showing the influence of cavitation on the occurrence of heterogeneous nucleation in the nozzle hole.

The first drop size comparisons which must be investigated are basic time-averaged mean drop size parameters, such as $D_{10}$, $D_{32}$, $D_{v50}$ and $D_{v90}$. These plots in Figure 6-31 show a general trend of a reduction in drop size correlating with an increase in superheat. As before for test points with equal superheat the results show some, but not total, similarity. However, as the optical transmission will vary at different points in the spray for the different test conditions there will be significant errors in this analysis, so it is essential to look in more detail at alternative analysis methods.

![Figure 6-31 - $D_{10}$, $D_{32}$, $D_{v50}$ and $D_{v90}$ trends for all tested conditions](image)

These drop size data have been compared with the results of (Witlox 2010) and (Kay 2010), Figure 3-21. Both data sets show a 3-stage process in the relationship between superheat degree and SMD, with the greatest reduction in SMD being in the transition region between non-flashing and flare flashing.
As drop size data through the main and interstitial streams were gathered it was possible for this to be combined for each condition to produce histograms of the total number of droplets in each size bin occurring from electronic start of injection to 4ms after that time, thus capturing all the droplets occurring during the main body of the spray. This plot is shown in Figure 6-33.

The number of small droplets (<10µm) recorded is very dependent on the optical transmission through the spray, and is not of interest in this investigation as a large number of small droplets is an inevitable, and desirable, consequence of the flash boiling process.
Therefore, future histograms will focus only on the number of droplets in the size class above 10 µm, as shown in Figure 6-34. The number of large drops should be a good indicator of the level of flash boiling, as large drops have a high probability of undergoing flash boiling due to their high volume to surface area ratio.
From this plot it can be seen that there are very few drops with diameter over 25µm, hence future histograms will only use the range 10-25µm. Another improvement which can be made is to present the droplet number as a percentage of the total drops over the total test time at that condition, as in Figure 6-35. This helps account for different levels of beam transmission efficiency between the different spray patterns.

It is clear from this plot that the flash boiling process has a significant effect on reducing the numbers of large droplets within the spray. These large drops will be broken down to form a number of small droplets. This reduction in numbers of large drops shows an increased level of flash boiling.

The plot also shows a reduction in the number of large drops with an increase in the superheat degree. There is also some correlation between the histograms of the test points with equal superheat degree. Differences in the frequency presented in Figure 6-35 for these cases may be due to the number of small drops found, due to variations in the optical density of the spray as a result of the different spray pattern. This is confirmed for the SD=-12°C results by further analysis of Figure 6-34, where it can be seen that there is in fact more large droplets for the 0.1 bar 23°C case than the 0.3 bar 50°C case, which is not reflected in the histograms due to the reduced raw number of small drops recorded for the 0.3 bar 50°C case, as shown in Figure 6-35.

However, for the two cases with a superheat degree of 12°C, further analysis of Figure 6-34 shows that there is, in fact, a reduced number of large drops recorded for the 0.3 bar 74°C case in comparison to the 0.1 bar 47°C case. This could be evidence that the increased fuel temperature plays the dominant role in increased spray break-up for flashing sprays. As a
droplet progresses downstream at a high relative velocity to the surrounding air the pressure field local to the droplet, particularly as it deforms, will not be equal to the overall ambient pressure in the chamber, and therefore could create a localised higher superheat degree for a short time, thus increasing the likelihood of flash boiling.

6.6 Comparison between Different Fuels

It is well known (Zigan 2010 (b)) that different fuels will behave differently when emitted from the same injector due to their different properties, even at ambient temperature and pressure conditions. With the addition of superheat and therefore evaporation of the spray, this introduces another variable to account for the differing behaviour of the different fuels (Zigan 2011).

For this investigation two fuels with similar properties, Table 6-4, n-heptane and iso-octane, were chosen. Under non-flashing conditions there is little difference in the atomisation characteristics of these fuels, (Zigan 2011). Therefore they will act as suitable test fuels to assess the impact of superheat degree on atomisation performance.

<table>
<thead>
<tr>
<th></th>
<th>n-heptane</th>
<th>iso-octane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid density (kg/m³)</td>
<td>682</td>
<td>690</td>
</tr>
<tr>
<td>Kinematic viscosity (mm²/s)</td>
<td>0.6</td>
<td>0.72</td>
</tr>
<tr>
<td>Surface tension (mN/m)</td>
<td>20.14</td>
<td>18.77</td>
</tr>
<tr>
<td>Gas density (kg/m³)</td>
<td>4.15</td>
<td>4.68</td>
</tr>
<tr>
<td>Specific heat (J/Kmol)</td>
<td>224.64</td>
<td>242.5</td>
</tr>
<tr>
<td>Latent heat of vaporisation (kJ/mol)</td>
<td>31.8</td>
<td>30.8</td>
</tr>
</tbody>
</table>

It has already been shown that superheat degree is a somewhat suitable parameter to describe the behaviour of sprays, however previous data showed it to be incorrect to expect conditions with the same superheat degree to produce an identical spray. This section will attempt to explain the extent to which superheat degree is a suitable parameter to describe the expected behaviour of a spray from a GDI injector when considering different fuels.

Having very similar physical properties these two fuels should exhibit similar levels of stream to stream interaction, and as shown previously this is vital to the resultant spray morphology of flashing sprays, hence it is anticipated that there will be very few differences in spray morphology for equal superheat cases.

According to earlier shown correlations (Kitamura 1986) it is possible to calculate the expected correlation between superheat required to phase change between the two fuels. It was shown that phase transition varied with We⁻¹/³. Using the parameters listed in Table 6-3 and a droplet velocity of 100m/s and droplet diameter of 5μm the parameters for the fuels are calculated as in Table 6-5:
Table 6-5 - Weber and Jakob numbers for n-heptane and iso-octane

<table>
<thead>
<tr>
<th></th>
<th>n-heptane</th>
<th>iso-octane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weber number</td>
<td>10.3</td>
<td>10.7</td>
</tr>
<tr>
<td>Jakob number</td>
<td>1.17ΔT</td>
<td>1.16ΔT</td>
</tr>
</tbody>
</table>

As the Jakob number for transition is a function of $We^{-1/7}$ the superheat required for transition varies with $We^{-1/7}/Ja$ with $Ja$ representing a Jakob number constant. For both fuels this produces a superheat constant of 0.61 required for transition, which means the same value of superheat would be required for transition to flashing spray. This value is not the superheat required for transition, rather a value by which fuels with different parameters can be compared. It would be impossible to calculate the true value of superheat required for transition numerically, as the number would have to represent a large range of drop size and droplet velocity combinations.

The results for shadowgraphy, Figure 6-36, show a general similarity between the fuels over the three operating conditions.

For the hotter two results, 47°C and 71°C, there does appear to be a slight increase in the level of flashing for the n-heptane cases over the iso-octane cases, as determined by the marginal increase in the level of fuel drawn into the recirculating tip vortices. The iso-octane result for 71°C also does not exhibit an increase in penetration equal to that of the equivalent n-heptane result. This shows a slight decrease in the level of flashing.
<table>
<thead>
<tr>
<th>P=0.1 bar T=47°C</th>
<th>n-heptane</th>
<th>iso-octane</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1.png" alt="Image" /></td>
<td><img src="image2.png" alt="Image" /></td>
<td><img src="image3.png" alt="Image" /></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>P=0.1 bar T=71°C</th>
<th>n-heptane</th>
<th>iso-octane</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image4.png" alt="Image" /></td>
<td><img src="image5.png" alt="Image" /></td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
</tbody>
</table>

Figure 6-36 – Shadowgraphy images for n-heptane and iso-octane @ Z=40mm and t=1100μs AESOI

The planar Mie results, Figure 6-37, show overall similar spray patterns for the two test fuels. However, there is a higher tendency for the spray to collapse towards the core region for the iso-octane case, where the fuel appears to be more likely to congregate. The results at 47°C indicate that for the iso-octane case the flashed fuel tends towards the spray core, whereas for the n-heptane case it tends towards the gaps between the original spray streams, similar to the two cases with n-heptane, in which an equal superheat degree could either lead to a ring formation, or a partially collapsed spray. This is most likely due to different levels of stream to stream interaction causing different airflow patterns within the spray area.
Under the three conditions tested there is very little change in the overall spray morphology for the two different fuels. PDA data were used to analyse the differences in droplet trajectory over the full injection period.
For the first condition, P=0.1 bar T=23°C, there is very little difference seen in the imaging study and the PDA study shows that at all radial positions the spray behaviour in both the main and interstitial streams is nearly identical, as in Figure 6-38. This degree of similarity is repeated for all radial measurement points at this condition.

For the P=0.1 bar T=47°C cases the PDA results present a highly similar pattern for both fuels. However, the planar Mie-imaging shows a slightly different spray pattern at 40mm downstream of the injector tip.

The only slight difference observed in the PDA data, which is consistent at all points radially out from the injector axis, is that the difference in mean radial velocity between the mains and interstitial streams from the iso-octane case is higher than that for the n-heptane spray, which would promote different levels of entrainment and therefore different levels of spray collapsing. The differences between the two streams, in Figure 6-39, of 5m/s for n-heptane and 8m/s for iso-octane during the period 1.5-2.5ms AESOI show the likelihood of droplets staying in the mains stream area over the interstitial streams are therefore increased for the iso-octane fuel, thus explaining the differences in spray patteration. This difference is also present during the earlier stages of injection, with a peak radial velocity of 27m/s for n-heptane and only 23m/s for iso-octane. This correlates to a weaker recirculating tip vortex which was identified in the images in Figure 6-36. Analysis of the axial velocity profiles, which are detailed in full in the appendices, reveals no differences in these, showing the differences must be due to these slight differences in radial velocity profiles.
As for the highest superheat case (SD=36°C), data were only acquired for the interstitial streams with iso-octane fuel only the behaviour in the interstitial streams can be compared, as in
Figure 6-40. This shows that during the initial collapse period there is a more pronounced collapse for the iso-octane spray. However, without data from the mains spray it is unclear whether this is as the result of an overall increased strength of spray collapse. But the limited results presented here do suggest that the differing tip vortices between the different fuels are integral to the droplet entrainment procedure and therefore the overall spray morphology.

Finally, to determine differences in the level of flash boiling for the two fuels at the different conditions it is essential to examine the drop size profile at each condition, as in Figure 6-41. This shows that for all three conditions investigated there is a very similar drop size histogram. The differences in percentage for the highest superheat cases can be explained by a low droplet number count for the iso-octane results. However, for the moderate superheat cases (P=0.1bar T=47°C) the droplet count is similar for both results, and as such the slight differences are unexplained.

These results suggest that the majority of gasoline like fuels should behave in a similar manner both under standard and flash boiling conditions. The superheat degree is shown to be a major driver in the spray development and a relatively good predictor of spray atomisation.

This is not to say that superheat degree is a perfect predictor of atomisation and morphology from a GDI injector. Prior to the terminal failure of the prototype injector, it was planned to conduct tests on an alcohol fuel with a similar boiling point to n-heptane and iso-octane at 0.1 bar. This is because the physical properties of this fuel which control the spray break-up are vastly different to gasoline like fuels, hence the stream to stream interaction levels should be
very different from the n-heptane and iso-octane results, which theoretically should lead to
different levels of collapsing, and other changes in morphology, for equal superheat cases.

6.7 Conclusions

A number of interesting conclusions can be drawn from this work.

Firstly, the complete degradation of the standard spray shape as the superheat degree was
increased was shown. As the superheat degree approached zero (from negative) a widening of
the spray plumes was observed, along with a slight tendency of the spray towards the core
region. At superheat degrees slightly over zero this process was further evident, with fuel being
transferred to a number of locations under which it would not exist under non-flashing
conditions. A further increase in superheat led to the complete spray collapse, which has
previously been shown by many other researchers (Mojtabi 2011), (Zhang 2012). As expected
there was an associated increase in penetration and reduction in cone angle. The formation of
interstitial streams was also shown. This began to occur very soon after the fuel exited the
injector orifice, but the process of their formation, along with the disappearance of the initial
streams, was not complete until the fuel had travelled some distance downstream of the
injector tip (~20mm for this case).

Phase Doppler data showed a general similarity in axial velocity profiles within the spray for
each condition. However, there were more significant differences in the radial velocity profile,
showing the effect of low pressure regions particularly in the spray core region and entrainment
on the spray collapse and subsequent formation of interstitial streams. The PDA results also
show the significance of the recirculating tip vortex on the spray collapse and the subsequent
spray behaviour as it progresses downstream.

Interestingly, the imaging study revealed some major differences in the spray pattern for points
with equal superheat, but different temperature and pressure conditions. For one such pair of
tests fuel existed in the original spray stream location for both tests, but for one test condition it
additional had been transported into the gaps between each stream to form a ring, whereas in
the other had been transferred to the spray core. However, the PDA data showed the overall
mean droplet motion was very similar, with only minor differences existing. The fact that such
minor differences in velocity, and therefore entrainment field, can cause such differences in
spray pattern shows that the droplets produced by the flash boiling process must have very low
momentum, and hence be easily entrained.

The drop size data show the expected general trend of reduced drop size with an increase in
superheat. For high superheat degree, there are practically no large drops (>15µm) within the
spray. This is because the large drops have the highest volume to surface area ratio, so have the
highest likelihood of undergoing flash boiling. There is a clear correlation between increased
superheat and a reduction in the numbers of large drops within the spray.

The comparison between the results obtained with n-heptane and iso-octane fuel as expected
display a high degree of similarity. The results from imaging, phase Doppler velocity data and
Phase Doppler drop size data all show general similarity based on the superheat degree. However, there are a few differences shown between the different fuels at moderate superheat conditions, most notably the higher level of collapsing for iso-octane fuel than for n-heptane.

Overall these results show that for a narrow cone angle injector, similar to that which would be used in a GDI engine, the superheat degree required for flare flashing (the complete degradation of the spray shape) is relatively low (between 12-36°C). This value would regularly be exceeded in a GDI engine.

The results also have shown the importance of the entrainment of droplets within the spray on the resulting spray morphology from flashing GDI sprays. After the flashing process, there are many slow moving small droplets which are easily entrained, so tend towards areas with low pressure and low fuel concentration, hence the fuel being transferred to locations such as the spray core and interstitial streams.

6.8 Future Work

The body of work in this section clearly constitutes a large body of data and subsequent analysis, but by no means represents a complete analysis of flash boiling sprays for GDI multi-hole injectors. A number of factors have not been fully investigated and a list of future parameters to be tested is included in Table 6-6.

Table 6-6 - Future work on flashing sprays

| Fuels | • Test a greater range of fuels with increased differences in boiling point curves  
|       | • Test a number of alcohol fuels which will have different break-up characteristics before the influence of superheat is even considered 
|       | • Test a variety of fuel blends |
| Injectors | • Test a number of different injectors with differences in:  
|          | o Number of holes  
|          | o Cone angle  
|          | o Spray angle (and therefore stream to stream interaction)  
|          | o Orientation of holes  
|          | o Surface roughness inside nozzle holes (to test influence of cavitation on heterogeneous nucleation)  
|          | • Possible future work using acrylic or glass nozzles to visualise cavitation formation under superheated conditions |
| Simulation | • Much simulation work on this topic is currently being undertaken by Continental Automotive in collaboration with these results |
| Engine testing | • Testing possible in a real engine to test the influence of flashing sprays on emissions and fuel economy  
|               | • Testing also suggested in an optical engine to visualise the effect of fast moving input air motion on the spray dispersion due to large number of small droplets |
| Additional test points | • If there is a superheat degree above which the spray becomes stable  
|                        | • Point at which flare flashing begins  
|                        | • Point at which partial flashing begins |
| Additional experimental techniques | • PIV for visualisation motion across planes and entrainment fields in both the near and far-field  
• LIF for speciation with multi-component fuels and for tracking of liquid and vapour phases  
• Desire to know fuel temperature as it penetrates through the air, many techniques exist (eg. CARS) but none appear perfectly suited to this problem  
• Possibility of high-speed/high-resolution imaging to track droplet motion immediately after flash boiling event occurs |
7 Conclusions

A large number of influential sources have determined that the gasoline engine will continue to play a large part in vehicle propulsion for the foreseeable future. The increased use of hybridisation will increase the time until these products are driven out of the market by alternative propulsion methods (Allen 2011), (Fulton 2011), (Kaji 2004), (King 2007), (Offer 2010), (Pearson 2007), (Pischinger 2006), (Tanaka 2011), (Taylor 2008), (Tunison 2011), (Wagner 2013), (Yoshida 2011). However, what is also undeniable is that the gasoline engine must continually be improved to meet the growing demands in terms of emissions, fuel economy, noise and reliability, among other characteristics. The majority of gasoline engines produced at the current time are gasoline direct injection (GDI) engines and one of the key areas in GDI engines which can significantly influence many of these characteristics is the fuel spray.

The data and analysis in this thesis investigates the behaviour of the fuel injector and fuel spray under two novel operating strategies. Firstly the effect of split injection strategies which are designed to help these engines meet future strict emissions standards on hydrocarbon (HC) emissions through reduction of spray/wall impingement. Secondly the spray behaviour under flash boiling conditions, which can occur in the engine when running with high levels of throttling and a hot engine, therefore hot fuel.

7.1 Split Injection Strategies

As previously stated the concept of split injection strategies, in which the fuel is introduced into the cylinder in a number of short injections as opposed to a single longer injection, is used to reduce the HC emissions from the engine. As the fuel is introduced in a number of short bursts, it encounters a cumulatively higher level of air resistance thus reducing the overall spray penetration. The reduced penetration leads to reduced wall and piston wetting which have been shown to be a major source of HC emissions (Sandquist 2000).

Two generations of multi-hole injector were tested, the Continental XL2 and XL3. These tests used a 3-hole injector with a 90° cone angle which was specifically produced for research purposes.

The first point of note from these tests is that during split injection operation the actual opening and closing behaviour of the injector can be greatly altered. With an insufficiently long dwell time (the gap between subsequent injections) the reproduction of a reliable second injection is not possible. This minimum dwell time was found to be 400 and 250μs for the two injectors, respectively. However, for the XL2 injector there was also a needle bounce present which was shown to interact with the start of the second injection with the shorter dwell times tested. Further differences were also present due to eddy currents remaining in the solenoid from the first injection aiding the needle opening for the second injection. These eddy currents lead to a faster needle opening, which, as will be discussed later, is advantageous towards injector operation.
This quicker opening, however, may lead to some issues when these injectors operate split injection strategies in an engine. As the opening time, and thus total fuel mass delivered is not constant, it is imperative that this is monitored accurately to maintain correct air to fuel ratios.

Also of considerable importance in this investigation was the penetration reduction possible through the use of split injection strategies. Although the test points were not designed for maximum reduction of penetration a small reduction was observed of around 10% of the equivalent single injection.

As the second spray is injected into the wake of the first injection there is an inevitable entrainment effect. With the shorter dwell times this lead to the second spray exhibiting a higher mean velocity than the first spray and eventually “catching” it downstream. With the longer dwell times the entrainment effect is very weak and therefore the spray velocity and penetration rate are very similar.

With the interaction of the second spray and the needle bounce for the XL2 injector, the early spray development, in terms of morphology, was vastly different from that of spray 1. The spray axis location was shown to vary by as much as 2mm within the first 10mm downstream of the nozzle tip. This effect continued downstream with small differences being present in the overall spray angle.

The final, and perhaps most interesting, set of data focused on the differences in drop size within the first and second spray. During the steady state period of both sprays the mean drop size is very similar. However in the early period of injection (first 200μs after initial fuel exits the injector) there are significant differences. There are usually a number of large (>15μm) droplets present in the far-field (40mm downstream) of the injector in the early stages of injection for a single injection, or in this case an injection 1. However, with the use of split injection strategies the number of large drops recorded in the second injection is dramatically reduced. High-zoom Mie-imaging revealed that this is due to a more finely atomised spray being produced in the early stages of the second injection, with an improvement in atomisation correlating to a reduction in dwell time and therefore a reduction in the needle opening time.

These results show that where injection timing permits (at some load/speed points the fuel requirements in such a short space of time mean that the only way to introduce sufficient fuel is through a single injection) split injection strategies are highly advantageous for engine operation. The combination of penetration reduction with the opportunities for improved homogenisation without any major drop size penalties are highly advantageous and afford the engine designer with additional capabilities for improvement of engine operation. However, the possible improvements come with the caveats that for efficient use of split injection strategies over the lifetime of the engine the engine will require additional monitoring and control as the injector opening and closing behaviour may vary over time.
7.2 Flash Boiling

A liquid will boil when its temperature exceeds its boiling point, and with reduced ambient pressure (as can occur in the combustion chamber of a GDI engine) the temperature of the fuel injected can consequently be above its boiling point. Immediately prior to injection the fuel will be under significant pressure (100-200bar) and will be in a liquid state, however, upon injection into the cylinder the surrounding pressure drops rapidly and can lead to rapid boiling of the fuel. The surface of a fuel droplet may not be able to evaporate quickly enough to sustain the boiling, thus the inside of the droplet may begin to boil forming vapour bubbles within the droplet. These bubbles will begin to grow until the point at which they cause the droplet to shatter into a number of much smaller daughter droplets with momentum equal to that of the parent droplet. Fuel can be put into this state through heat it accrues from the cylinder head which gets hot as a result of normal engine operation and during certain stages of engine operation, such as throttled, when the in-cylinder pressure can fall to levels well below 1bar.

Testing used a Continental XL3 injector with 6 holes and a 60° cone angle. The injector is a prototype but represents an injector very close to what would be used as a production injector.

The first part of this testing was to confirm whether the results obtained within the test cell designed for this work were applicable for analysis which is usually used on imaging and PDA data. The results show that there are some discrepancies in the phase Doppler data due to losses of laser power in the thick optical windows of the test cell, but the overall temporal mean averages show a good level of agreement.

Initial analysis of flashing sprays focused on capturing the spray morphologies with various degrees of superheat and therefore levels of flash boiling. The standard spray under atmospheric temperature and pressure conditions was first characterised. With the addition of some heat and reduction of chamber pressure, but still to a point with negative superheat, an increase in atomisation and spray dispersion while still maintaining the 6 individual spray plumes was shown. With further heating up to a slightly superheated state there were significant changes in the spray morphology. The overall envelope into which the spray sat was similar to previous, however the planar results showed that fuel had migrated into different areas of the spray. In some cases the fuel migrated into the gaps between individual spray plumes in addition to some fuel remaining in the original spray streams to form a ring like structure. But in other cases the fuel collapsed towards the spray core in addition to the fuel which remained in the original streams. It is believed the fuel which migrates to occupy these new areas is the flashed fuel and the fuel in the original streams is the fuel which did not undergo flash boiling. With significant superheat there were further changes in the spray morphology. Fuel was drawn into recirculating tip vortices and the spray footprint changed such that fuel remained only in the areas other than the original spray streams: the spray core and the “interstitial” streams. With this high superheat degree it is likely nearly all the fuel will undergo flash boiling, hence the disappearance of the original spray streams.
Phase Doppler data of these conditions showed the importance of the radial velocity on the spray morphology. Under flashing conditions the many small droplets which have very low momentum are readily entrained into areas of the spray, such as the spray core, into which larger droplets would not be entrained. The discrete droplet phase Doppler data also showed that the droplets which were entrained into the recirculating tip vortices were the smallest droplets.

Test points (in terms of ambient pressure and fuel temperature) were chosen such that points with an equal superheat degree but different pressure and temperature conditions could be compared. The imaging results displayed a number of significant differences in overall morphology. However, the phase Doppler data showed a high degree of similarity between the droplet motion between the compared cases. This indicates that with the high likelihood of entrainment of the small droplets formed through flash boiling even small differences in the entrainment field can lead to major differences in the spray morphology.

Of significant interest in this work was the determination of drop size reduction through flash boiling. The theory of flash boiling points towards an inevitable reduction of drop size within the spray, however this has never been reliably quantified. The work here shows significant reductions with a reduction in $D_{32}$ from 10 to 6µm for a non-superheated and a 36°C superheat degree case respectively. Of note was that there were major reductions in the number of large droplets within the spray for superheated cases. This is because the larger droplets have the highest volume to surface area ratio, so are the droplets most likely to undergo flash boiling. There was also shown to be a correlation between increased superheat and a reduction in both mean drop size and the numbers of large drops. Cases with the same superheat degree were shown to have similar drop size histograms, but by no means identical.

A second data set was taken for three of the tested conditions using iso-octane fuel to compare to the previously discussed results for n-heptane. The results show a relatively high degree of similarity, from a spray morphology, droplet motion and droplet size perspective. Of note is that the iso-octane spray is more likely to collapse towards the spray core than the n-heptane spray.

Comparison of these results with other published work has shown the importance of stream to stream interaction and entrainment of droplets on the spray morphology under flashing conditions. The spray formed from narrow cone angle injectors, such as the injector tested here appears to collapse with a low superheat degree and even negative superheat can lead to significant changes in the behaviour of the individual plumes. However, results from the literature show wider cone angle injectors have a much higher superheat requirement to induce a spray collapse. This is because there is a lower level of the stream to stream interaction which is essential for the build-up of a low pressure zone in the spray core into which the small droplets created through flash boiling are entrained.

As to whether flash boiling sprays are desirable within a GDI engine is an argument to which the answer is as of yet unclear. An ideal spray will produce a lot of small droplets, without producing too many large droplets. Such a spray is produced through flash boiling. However, during
homogeneous operation (where there is a possibility of flash boiling) it is desired to have a highly repeatable spray over the full range of operating conditions. This is so the fuel/air mixing process can be controlled. With a spray which flashes under some operating conditions (high throttling) and not under the rest of the engine cycle this could cause problems with the homogeneity of the air/fuel mixture and possibly have an adverse effect on the engine performance. To make the use of flash boiling advantageous to engine operation would require some alterations to engine design to accommodate the differences in the sprays under flashing conditions along with vast advances in engine control. Such advances are expected in the near future meaning that the knowledge on flash boiling sprays gained through this work could be vital in future engine design.
8 Bibliography


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9 Appendices

9.1 PDA Flashing Velocity Data in Full

9.1.1 P=0.1 bar T=23°C n-Heptane

Mains Streams
Interstitial Streams

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**Axial Velocity (m/s)**

- **0**
- **2.5**
- **5**
- **7.5**
- **10**
- **12.5**
- **15**
- **17.5**

**Time AESOI (ms)**

0 - 4

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**Radial Velocity (m/s)**

- **0**
- **2.5**
- **5**
- **7.5**
- **10**
- **12.5**
- **15**
- **17.5**

**Time AESOI (ms)**

0 - 4
9.1.2 P=0.3 bar T=50°C n-Heptane

Mains Streams
Interstitial Streams

[Graph showing axial and radial velocity over time for different AESOI values]
9.1.3  P=0.1 bar  T=47°C n-Heptane

Mains Streams
Interstitial Streams

![Graphs showing axial and radial velocity over time](image-url)

- Axial velocity vs. Time AESOI (ms)
- Radial velocity vs. Time AESOI (ms)
9.1.4  \( P=0.3 \text{ bar} \) \( T=74^\circ C \) n-Heptane

Mains Streams
Interstitial Streams

Interaxial velocity (m/s)

Time AESOI (ms)

Radial velocity (m/s)

Time AESOI (ms)
9.1.5 P=0.1 bar T=71°C n-Heptane

Mains Streams
Interstitial Streams
9.1.6 P=0.1 bar T=23°C iso-Octane

Mains Streams
Interstitial Streams
9.1.7  P=0.1 bar T=47°C iso-Octane

Mains Streams
9.1.8 P=0.1 bar T=71°C iso-Octane

Interstitial Streams