An optical investigation of cavitation phenomena in true-scale high-pressure diesel fuel injector nozzles

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AN OPTICAL INVESTIGATION OF CAVITATION PHENOMENA IN TRUE-SCALE HIGH-PRESSURE DIESEL FUEL INJECTOR NOZZLES

By
Benjamin A. Reid
MEng. (Hons) DIS

A Doctoral Thesis submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of Loughborough University

2010

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ABSTRACT

Efforts to improve diesel fuel sprays have led to a significant increase in fuel injection pressures and a reduction in nozzle-hole diameters. Under these conditions, the likelihood for the internal nozzle flow to cavitate is increased, which potentially affects spray breakup and atomisation, but also increases the risk of causing cavitation damage to the injector.

This thesis describes the study of cavitating flow phenomena in various single and multi-hole optical nozzle geometries. It includes the design and development of a high-pressure optical fuel injector test facility with which the cavitating flows were observed. Experiments were undertaken using real-scale optical diesel injector nozzles at fuel injection pressures up to 2050 bar, observing for the first time the characteristics of the internal nozzle-flow under realistic fuel injection conditions.

High-speed video and high resolution photography, using laser illumination sources, were used to capture the cavitating flow in the nozzle-holes and sac volume of the optical nozzles, which contained holes ranging in size from 110 micrometers to 300 micrometers. Geometric cavitation in the nozzle-holes and string cavitation formation in the nozzle-holes and sac volume were both observed using transient and steady-state injection conditions; injecting into gaseous and liquid back pressures up to 150 bar.

Results obtained have shown that cavitation strings observed at realistic fuel injection pressures exhibit the same physical characteristics as those observed at lower pressures. The formation of string cavitation was observed in the 300 micrometers multi-hole nozzle geometries, exhibiting a mutual dependence on nozzle flow-rate and the geometry of the nozzle-holes. Pressure changes, caused by localised turbulent perturbations in the sac volume and transient fuel injection characteristics, independently affected the geometric and string cavitation formation in each of the holes.

String cavitation formation was shown to occur when free-stream vapour was entrained into the low pressure core of a sufficiently intense coherent vortex. Hole diameters less than or equal to 160 micrometers were found to suppress string cavitation formation, with this effect a result of the reduced nozzle flow rate and vortex intensity. Using different hole spacing geometries, it was demonstrated that the formation of cavitation strings in a particular geometry became independent of fuel injection and back pressure once a threshold pressure drop across the nozzle had been reached.
ACKNOWLEDGEMENTS

I will be forever grateful to my supervisors Professor Colin Garner and Professor Graham Hargrave, who gave me the opportunity to undertake this research project, and allowed me to take on the added responsibilities that became available after the early departure of the project’s Research Associate. Their constant support and encouragement, which began when they supervised my undergraduate project work, has given me the freedom to pursue my research and provided me with experiences that have tested my capabilities in a wide range of situations.

I would especially like to thank Professor Hargrave for the countless times I have bounced ideas and thoughts off him regarding experimental methods and the interpretation of results, no matter how outlandish they may have been. The technical input provided Mr. Edward Long, Dr. Graham Wigley, Mr. Tim Justham and Dr. Simon Jarvis was invaluable to the development of my research techniques. And I cannot neglect to mention the priceless experience of working with Mr. Peter Wileman. His technical expertise and dedication to taking the mick at every available opportunity has provided valuable extra dimensions to my professional development.

The services of the University technical staff was also much appreciated. Pete Wileman’s partner-in-crime, ‘Big’ Dave Britton, was always on hand to solve a problem or simply shift something that no-one else could shift. Mr Jagpal Singh’s services in the metrology lab were exceptionally helpful. The workshop staff, headed by Mr. Bob Ludlam, were always happy to squeeze a specialist request from me into their often busy schedule – in particular Mr. John Hales and Mr. Mick Porter, with Mick’s beer festival info. also being much appreciated. Notable mentions must also go to Mr. Dick Price and Mr. Brian Mace, Mr. Dave Travis and Mr. John Doorman in the Thermo Lab. The encouragement and advice of Mr. Bob Temple was also greatly appreciated. I would also like to thank Mr. Andy Sandaver for aiding my efforts in polishing sapphire and thank the team in the electronics workshop of Mr. Dave King, Mr. Steve Hammond, Mr. Tim Rowe and Mr. Alan Trahar. Thanks also to Mr. Keven Smith for dealing with the endless stream of deliveries I managed to accumulate, and to Mrs. Sue Cramp and Mrs. Bridgette Shipman for ordering everything in the first place.

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The technical and financial support provided by Caterpillar Inc. Made this work possible in the first place and I would like to thank Cathy Choi, Suzy Danby, Kaimei
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“That which does not kill us, makes us stronger”

- F. Nietzsche
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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_f$</td>
<td>Cross sectional flow area</td>
<td>$m^2$</td>
</tr>
<tr>
<td>$c$</td>
<td>Speed of sound</td>
<td>$m.s^{-1}$</td>
</tr>
<tr>
<td>$C_c$</td>
<td>Vena contracta coefficient</td>
<td>-</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Discharge coefficient</td>
<td>-</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Injection average discharge coefficient</td>
<td>-</td>
</tr>
<tr>
<td>$D$</td>
<td>Nozzle hole diameter</td>
<td>$m$</td>
</tr>
<tr>
<td>$D_{inlet}$</td>
<td>Nozzle hole inlet diameter (for k-factor)</td>
<td>$\mu m$</td>
</tr>
<tr>
<td>$D_{outlet}$</td>
<td>Nozzle hole outlet diameter (for k-factor)</td>
<td>$\mu m$</td>
</tr>
<tr>
<td>$dQ$</td>
<td>Flow volume</td>
<td>$m^3$</td>
</tr>
<tr>
<td>$dt$</td>
<td>Unit of time</td>
<td>$s$</td>
</tr>
<tr>
<td>$f_r$</td>
<td>Laser run frequency</td>
<td>$Hz$</td>
</tr>
<tr>
<td>$f_b$</td>
<td>Laser burst frequency</td>
<td>$Hz$</td>
</tr>
<tr>
<td>$k$</td>
<td>Nozzle conicity factor (k-factor)</td>
<td>$\mu m$</td>
</tr>
<tr>
<td>$L$</td>
<td>Nozzle hole length</td>
<td>$m$</td>
</tr>
<tr>
<td>$m$</td>
<td>Fuel mass</td>
<td>$kg$</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Fuel mass flow rate</td>
<td>$kg.s^{-1}$</td>
</tr>
<tr>
<td>$n$</td>
<td>Refractive index</td>
<td>-</td>
</tr>
<tr>
<td>$P_1$</td>
<td>Sac volume pressure</td>
<td>$Pa$</td>
</tr>
<tr>
<td>$P_2$</td>
<td>Nozzle back-pressure</td>
<td>$Pa$</td>
</tr>
<tr>
<td>$P_f$</td>
<td>Fluid Pressure</td>
<td>$Pa$</td>
</tr>
<tr>
<td>$P_{rail}$</td>
<td>Fuel rail pressure</td>
<td>$Pa$</td>
</tr>
<tr>
<td>$P_v$</td>
<td>Fluid vapour pressure</td>
<td>$Pa$</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>Pressure drop under obstruction</td>
<td>$Pa$</td>
</tr>
<tr>
<td>$r$</td>
<td>Radius</td>
<td>$m$</td>
</tr>
<tr>
<td>$R$</td>
<td>Vortex core radius</td>
<td>$m$</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$Re_{av}$</td>
<td>Injection average Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$Re_{\Gamma}$</td>
<td>Circulation Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$t_a$</td>
<td>Shutter activation time</td>
<td>$s$</td>
</tr>
<tr>
<td>$t_b$</td>
<td>Laser burst duration</td>
<td>$s$</td>
</tr>
</tbody>
</table>
t_f - Laser fire delay s

\( t_s \) - Sutter activation delay s

\( \tau_{\text{TTL-SOI}} \) - Time difference between injector TTL and SOI s

\( \Delta t \) - Laser pulse separation time s

\( u \) - Fluid velocity m.s\(^{-1}\)

\( U \) - Upstream fluid velocity m.s\(^{-1}\)

\( \bar{V} \) - Average nozzle-flow velocity m.s\(^{-1}\)

\( V_\theta \) - Angular velocity rad.s\(^{-1}\)

We - Weber number -

\( Z \) - Ohnesorge number -

\( \mu_l \) - Liquid dynamic viscosity kg.s\(^{-1}\).m\(^{-1}\)

\( \Gamma \) - Circulation m\(^2\)s\(^{-1}\)

\( \rho \) - Upstream fluid density kg.m\(^{-3}\)

\( \rho_l \) - Liquid density kg.m\(^{-3}\)

\( \sigma \) - Surface tension kg.s\(^{-2}\)

\( \sigma_{\text{av}} \) - Injection average cavitation number -

\( \sigma_f \) - Fully developed cavitation number -

\( \sigma_i \) - Incipient cavitation number -

**ABBREVIATIONS**

\( \mu \)-PIV - Micro particle image velocimetry

ACEA - European Automobile Manufacturers’ Association

ASOI - After start of injection

CAE - Computer aided engineering

CARB - California Air Resource Board

CFD - Computational fluid dynamics

CI - Compression ignition

CMM - Coordinate measuring machine

CO\(_2\) - Carbon dioxide
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>DSLR</td>
<td>Digital single lens reflex</td>
</tr>
<tr>
<td>ECM</td>
<td>Electronic control module</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>EOI</td>
<td>End of injection</td>
</tr>
<tr>
<td>F</td>
<td>Front hole of multi-hole</td>
</tr>
<tr>
<td>FEA</td>
<td>Finite element analysis</td>
</tr>
<tr>
<td>FIE</td>
<td>Fuel injection equipment</td>
</tr>
<tr>
<td>fps</td>
<td>Frames per second</td>
</tr>
<tr>
<td>HCCI</td>
<td>Homogeneous charge compression ignition</td>
</tr>
<tr>
<td>ID</td>
<td>Internal diameter</td>
</tr>
<tr>
<td>LDV</td>
<td>Laser Doppler velocimetry</td>
</tr>
<tr>
<td>MDU</td>
<td>Main drive unit</td>
</tr>
<tr>
<td>MP</td>
<td>Mega pixel</td>
</tr>
<tr>
<td>NMOS</td>
<td>n-type metal oxide semi-conductor</td>
</tr>
<tr>
<td>NOP</td>
<td>Nozzle opening pressure</td>
</tr>
<tr>
<td>NOx</td>
<td>Oxides of Nitrogen</td>
</tr>
<tr>
<td>NRTC</td>
<td>Non-road transient cycle</td>
</tr>
<tr>
<td>OD</td>
<td>Outer diameter</td>
</tr>
<tr>
<td>PDA</td>
<td>Phase Doppler anemometry</td>
</tr>
<tr>
<td>PIV</td>
<td>Particle image velocimetry</td>
</tr>
<tr>
<td>PM</td>
<td>Particulate matter</td>
</tr>
<tr>
<td>RL</td>
<td>Rear-left hole of multi-hole</td>
</tr>
<tr>
<td>RME</td>
<td>Rapeseed oil methyl ester</td>
</tr>
<tr>
<td>RMS</td>
<td>Root mean square</td>
</tr>
<tr>
<td>rpm</td>
<td>revolutions per minute</td>
</tr>
<tr>
<td>RR</td>
<td>Rear-right hole of multi-hole</td>
</tr>
<tr>
<td>SMD</td>
<td>Sauter mean diameter</td>
</tr>
<tr>
<td>SOI</td>
<td>Start of injection</td>
</tr>
<tr>
<td>TDC</td>
<td>Top dead centre</td>
</tr>
<tr>
<td>TTL</td>
<td>Transistor-transistor logic</td>
</tr>
<tr>
<td>VCO</td>
<td>Valve covered orifice</td>
</tr>
<tr>
<td>VOF</td>
<td>Volume of fluid</td>
</tr>
</tbody>
</table>
CHAPTER 1

INTRODUCTION
1.1 BACKGROUND

The automotive industry continues to face the challenge of ever more stringent emissions legislation, coupled with a trend towards the consumer demand for increased performance (EC Communication, 2007; ACEA, 2007a). Addressing both of these issues has brought about significant advances in both the theoretical understanding of engine operation and the practical application of new technologies.

Legislation has forced engine manufacturers to explore many aspects of the engine’s operation in order to reduce emissions whilst maintaining, and often increasing, power output across the full range of engine speeds and loads. Of critical importance in achieving this, the Fuel Injection Equipment (FIE) has undergone considerable development, with injection pressures over 1000 bar now commonplace in Compression Ignition (CI) diesel engines as part of a fully programmable, electronically controlled common-rail fuel injection system.

These developments have been made in an attempt to gain greater control over the combustion process and hence optimise engine performance within the limits of the legislation. Electronic control allows an almost infinite combination of injection timing, duration and multiple injection events on any given cycle, enabling engineers to more closely manage pollutant formation under varying engine operating conditions. However, it is the quality of the spray produced during each injection event that ultimately limits the resulting combustion quality. Consequently, improving the atomisation of the fuel and optimising the charge preparation remains fundamental to achieving emissions compliant engine operation.

To this end, fuel injection pressures have risen in an attempt to improve atomisation, with common-rail injection systems now providing relatively constant pressures above 1500 bar, irrespective of fuel pump/engine speed (Afzal et al., 1999). Most importantly of all, the fuel injector nozzle design has come under increased scrutiny, with its geometry directly affecting the spray characteristics and therefore the atomisation behaviour of the injector. For this reason, a better understanding of the nozzle flow characteristics, their controlling factors and their effects on the exiting
1.2 Emissions Legislation

As will be discussed further in Chapter 2, the importance of cavitation as a primary break-up mechanism has become apparent, with numerous investigations focusing solely on this flow phenomenon. In its simplest form, ‘geometric cavitation’ develops at the edges of the nozzle inlets, where the flow experiences a significant drop in static pressure as it accelerates into the nozzle, generating a significant axial pressure gradient. This combines with the strong curvature of the streamlines at the nozzle inlet edge, which generates a radial pressure gradient that is strongly dependent on nozzle geometry, to take the local fluid pressure below the fluid vapour pressure. The ensuing vapour cavities and bubbles that are formed in this low pressure region become entrained in the surrounding fluid and a two-phase flow exists within the nozzle. Due to fluid pressure recovery within the nozzle and/or the high ambient pressure of the combustion chamber, these vapour regions implode and have a significant effect on the primary spray break-up (Baumgarten, 2006). However, the benefits that cavitation can bring to spray atomisation are countered by the negative effects of bubble collapse near the internal surfaces of the nozzle. The surface shocks caused by bubble collapse lead to the localised deformation and pitting of the nozzle surface. This cavitation damage eventually results in fatigue failure and material loss, adversely affecting the fuel injector performance and potentially causing complete component failure (Asi, 2006). Therefore, understanding and accurately predicting cavitation is critical to the long term performance of an injector design.

1.2 EMISSIONS LEGISLATION

Since the 1960s with the creation of the California Air Resource Board (CARB) in the USA, governments have imposed increasingly stringent legislation on engine manufacturers to limit their exhaust pollutant output (Gerard and Lave, 2005). At each revision these standards have become more and more restrictive, to the point where limits currently being phased in (based on engine power output) by the US
Tier 4 off-highway legislation for 2011, require both nitrogen oxides (NOx) and particulate matter (PM) emissions to be further reduced by about 90% based on their Tier 3 levels of 2006 (DieselNet, 2010a).

European emissions standards have generally lagged considerably behind their U.S. equivalents. Prior to 1992, the European regulation of heavy duty diesel engine emissions was comparable in its stringency to the US regulations of the 1970s, and could be met, therefore, with very little design effort from engine manufacturers (Faiz et al., 1996). Following this, the European Directive (88/77/EEC), subsequently amended by Directive (91/542/EEC), resulted in the current framework of European emissions standards for heavy duty engines. The first stage standards (Euro I) adopted in 1992, however, remained comparable to 1988 US legislation and Euro II standards, adopted in 1996, were comparable to the US standards of 1991 (Faiz et al., 1996; DieselNet, 2010b). Subsequent introduction of transient test cycles as part of the Euro III standard adopted in 2000 brought greater parity to the standards of the two bodies. However, even now, the Euro VI standards, published in 2009 becoming effective in 2013, are comparable to the current US 2010 standards (DieselNet, 2010b). Such disparities complicate the streamlining of engine development and pressure has been placed on the regulatory authorities to harmonise global emissions standards as a result (ACEA, 2007b; ACEA, 2009).

For the off-highway engine market, harmonisation has been more forthcoming. Euro Stage I/II standards for off-highway applications, phased in from 1999, were part harmonised to the levels of the US Tier 1/2 legislation introduced from 1996. Euro Stage III/IV standards were then harmonised to the corresponding Tier 3 and 4 US limits. This also saw the introduction of non-road transient cycle (NRTC), which was developed by both regulatory bodies to provide a common engine test procedure (DieselNet, 2010c). However, the harmonisation of emissions limits has not detracted from the significant challenge that is posed to engine manufacturers in meeting the ever more restrictive regulations. Figure 1.1 illustrates the progression of the emissions limits placed on NOx and PM emissions for off-highway engines, with
power outputs between 130 and 560 kW, since the introduction of the Euro Stage I limits in 1999.

![Progression of NOx and PM emissions limits for off-highway compression ignition engines with rated powers between 130 and 560kW. Data from DieselNet (2010c)](image)

Although it is anticipated that meeting the Euro Stage III B (US Tier 4 interim) and Euro Stage IV (US Tier 4 final) limits will require some form of exhaust gas after-treatment, the role of in-cylinder methods to reduce engine-out emissions remains a significant one. Turbo-charging and exhaust gas recirculation (EGR) strategies along with increased cylinder and fuel injection pressures are cited as means by which pollutants can be reduced using more traditional diesel combustion strategies and hardware (Johnson, 2009).

Additionally, recent developments in the European passenger car and light-duty engine market have followed the example of the California ARB in adopting a legislative approach to the control of CO₂ emissions (EC Regulation, 2009). Prior to this, engine manufacturers had agreed to voluntary reductions in their fleet-average CO₂ emissions as part of the EU’s commitment to the Kyoto Protocol (Kyoto Protocol, 1998). However, failure to achieve the 140 g CO₂/km target set for 2008/2009, and the projected failure to meet the final 120 g CO₂/km objective by 2012, forced a revision of the policy. The new revised target of 130 g CO₂/km for passenger cars, to be phased in over a three year period from 2012, was agreed in
addition to a new long term target of 95 g CO₂/km for 2020 (see Figure 1.2). If the average CO₂ emissions of a manufacturer's fleet exceed this limit value, an excess emissions premium will be levied for each vehicle registered, based on the g/km by which they exceed the limit.

![Figure 1.2 Evolution of CO₂ emissions by fuel for the EU27. Data from European Communication (2009)](image)

The addition of CO₂ output restrictions to the already restrictive pollutant emissions legislation will require an increase in efforts to achieve improved engine efficiency and reduce fuel consumption. Central to this will be advanced combustion strategies and the ability to burn new fuel blends such as bio-fuels. The heavy-duty and off-highway markets, although exempt from the current CO₂ regulations, will continue to use improved fuel efficiency in combination with emissions compliance as a major selling point. With the diesel engine representing one of the most efficient energy converters available today, the incentive to continue its development in the face of increasing pressure for lower CO₂ emissions is strong (Oltra and Saint Jean, 2009). Therefore, further improvements must be made to the current “on-engine” technologies that have advanced the diesel engine over the past 20 years or so; a critical component of this being the FIE.
1.3 SPRAY BREAK-UP

The atomisation of the injected fuel and the subsequent rate of mixing with the charge air are fundamental to achieving desired engine performance and emissions (Baumgarten, 2006). Atomisation of the fuel serves to increase the effective surface area of the injected fuel quantity, aiding vaporisation, whilst maximising the distribution of the fuel in the cylinder. For these requirements to be met, a careful balance between atomisation and spray penetration must be achieved. The development of new advanced combustion modes such as HCCI, poses a significant challenge to achieving this balance across the full range of engine operation, with the potential range of cylinder pressures at SOI greatly increased. It is important therefore to develop an understanding of the fundamental mechanisms that control spray break-up and atomisation in order to better optimise injector design for a specific application (Lefebvre, 1989).

These mechanisms have been investigated, both theoretically and experimentally, for over a century. Of these investigations, the break-up of a liquid jet injected through a circular “plain nozzle” has been studied most frequently. These studies have established that the spray properties are influenced by a large number of parameters such as the velocity profile and turbulence at the nozzle exit and the physical and thermodynamic states of both the liquid and the gas; internal nozzle flow effects are also of great importance and will be discussed separately later.

As the fluid jet exits the nozzle, the dominant forces acting on the fluid change as the velocity of the jet is increased. This results in different regimes of spray breakup, dependent on the relative velocity and properties of the jet and surrounding gas. Reitz and Bracco (1986) reported four distinct regimes; the Rayleigh regime, the first and second wind-induced regimes and the atomisation regime, shown in Figure 1.3. At lower jet velocities the breakup is governed by the Rayleigh regime. Under these conditions the unbroken jet length increases with increasing jet velocity, and the droplets formed typically have a diameter greater than that of the nozzle hole diameter. Rotationally symmetric disturbances of the entire jet body, with
wavelengths beyond a critical value, are amplified by surface tension forces and initiate the jet breakup (Lefebvre, 1989).

Beyond this, further increases in jet velocity see a reduction in the breakup length as the governing regime changes to the first wind-induced breakup regime. Here, the forces from the Rayleigh regime are amplified by aerodynamic forces due to the relative velocity between the liquid and the gas. As a result, jet break up is accelerated, albeit still many jet diameters downstream of the nozzle, and droplet size is reduced to approximately that of the jet diameter. Further increasing the jet velocity sees a transition to the second wind-induced breakup regime. During this regime, turbulence within the jet creates unstable short-wavelength surface waves that are then augmented by the aerodynamic forces present. Unlike the Rayleigh regime which sees the jet break up as a whole, the jet now breaks up from the surface and is eroded inwards.

The final regime, and most important for the purposes of this study, is atomisation. The nature of the disintegration in the second wind-induced regime gives rise to two separate measurements; the intact surface length and the core length – relating to the start of surface breakup and complete disintegration of the jet respectively. The atomisation regime is reached when the intact surface length approaches zero. There may still be present an intact core of several nozzle diameters in length present, but the surface disintegration begins almost immediately upon exiting the nozzle.

Figure 1.3: Images showing examples of (a) the Rayleigh regime, (b) the first wind-induced regime, (c) the second wind-induced regime and (d) the atomisation regime. Adapted from Leipertz (2005)
Theoretically the above regimes have been defined by the relationship between the Reynolds number and the Ohnesorge number.

The dimensionless Ohnesorge number ($Z$) is derived from the Reynolds number ($Re$) and the Weber number ($We$) by eliminating jet velocity, in order to define the relationship between the viscous and surface tension forces. These are defined as

\[ Re = \frac{\rho_l u D}{\mu_l} \]  \hspace{1cm} (1.1)

\[ We_l = \frac{u^2 D \rho_l}{\sigma} \]  \hspace{1cm} (1.2)

\[ Z = \frac{\sqrt{We_l}}{Re} = \frac{\mu_l}{\sqrt{\sigma \rho_l D}} \] \hspace{1cm} (1.3)

This derivation includes all of the relevant fluid properties, namely $\sigma$: surface tension at the liquid-gas boundary, $\rho_l$: liquid density, $\mu_l$: liquid dynamic viscosity as well as nozzle hole diameter, $D$. Later studies discovered, however, that atomisation could be enhanced by increasing the gas density and so to take this into account Reitz
suggested the inclusion of the gas-to-liquid density ratio, extending the Ohnesorge diagram to include this additional dimension (Baumgarten, 2006).

The breakup mechanisms of the first two regimes are reasonably well understood, with linear stability theories able to predict the form of the breakup curve under these conditions fairly well (Chigier and Reitz, 1996; Lin and Reitz, 1998).

Beyond this, the shape of the breakup curve becomes unclear. Separate studies have described differing trends in the breakup length with further increases in jet velocity (Lin and Reitz, 1998). This makes predictive theory at higher Reynolds numbers more difficult as the characteristics of the spray now become influenced by the flow conditions inside the nozzle.

**1.4 CAVITATION**

Although turbulence in the nozzle remains an important factor in the ensuing spray development, at high jet velocities the influence of nozzle-flow cavitation must also be included and becomes integral to both understanding the nozzle flow conditions and the initiation of spray breakup.

Cavitation occurs when areas of the flow are subjected to pressures below the vapour pressure, $P_v$, of the liquid, forming a two-phase mixture that affects the turbulent behaviour of the downstream flow. In a fuel injector, the edges of the nozzle-hole inlet are therefore the most common region in which cavitation occurs, as the flow is accelerated from the sac region into the nozzle-hole. A combination of the drop in the fluid pressure, $P_f$, as the flow accelerates and the tendency for the flow to separate from the nozzle wall, if the inlet edge is sufficiently sharp, creates an area of low pressure around the circumference of the nozzle inlet. As flow velocities are increased, this low pressure region can extend to the length of the nozzle-hole, creating a two-phase flow regime at the nozzle-hole exit.
A useful comparison can be made between cavitation and boiling; both are essentially phase change processes, where boiling can be defined as the process of nucleation that occurs when a liquid temperature is raised at constant pressure, whilst cavitation is the process of nucleation that occurs when liquid pressure is reduced at constant temperature.

In both physical processes, nucleation forms precursory sites of the vapour phase in the liquid phase, which, dependent on the thermodynamic properties of the liquid, can grow to form macroscopic vapour bubbles that are maintained within the liquid phase. Two distinct types of nucleation exist depending on the physical properties of the liquid undergoing the phase change. For pure liquids, nuclei may be formed by the temporary microscopic voids that result from the thermal motion of the liquid. This form of nucleation is termed homogeneous nucleation. More common for practical engineering applications though, is the heterogeneous nucleation form. Here, nuclei form at the boundary between the liquid and a solid interface; either as ‘surface nuclei’ on the walls of the liquid container or on the surface of ‘free-stream’ nuclei; small particles or microbubbles, present as contaminants in the bulk of the liquid (Brennen, 1995).

Under depressurisation, an absence of nucleation sites within a liquid can lead to a continuation of the liquid state down a theoretical isotherm to a ‘metastable state’ –
so called because contaminants may lead to instability and a transition to an equilibrium point in the gaseous state. A liquid in the metastable state is said to be under tension, with the magnitude of the tension given by the pressure difference between the saturated vapour pressure and the pressure reached on the isotherm. Such conditions allow the local fluid pressure to fall below the vapour pressure without any phase change occurring. Subsequently, liquids that are sufficiently lacking in contaminants can withstand relatively high levels of tension, until the processes of homogeneous nucleation take effect and cavitation occurs. To this effect, Joseph (1998) proposes a criterion of cavitation inception based on fluid tension. In the presence of sufficient nuclei, however, phase change will occur as soon as the vapour pressure is reached, or even before, if a contaminant gas is present. Therefore, the prediction of cavitation inception is a complex problem requiring accurate knowledge of the liquid properties, which, in the case of nucleation-site population, are extremely hard to quantify (Brennen, 1995).

Following nucleation, the nuclei may grow to form macroscopic cavitation bubbles, which then coalesce or collapse depending on the surrounding pressure field and liquid properties.

The importance of understanding cavitation in the design of hydrodynamic applications has led to a large number of experimental and computational studies, which have attempted to quantify the parameters that affect its formation and subsequent interaction with the surrounding flow-field. Experimental Studies have, until now, been limited to obtaining optical flow data from scaled experimental rigs. Fluid pressures are often reduced and flow geometries increased, particularly in the case of diesel injector flows, which, as previously mentioned, typically operate at pressures exceeding 1500 bar in nozzle holes less than a few hundred μm in diameter. This leaves little or no knowledge of the actual internal flow behaviour at realistic operating conditions. Numerical simulations have allowed these flow environments to be investigated beyond the scope of the experimental studies; however, suitable data to validate the simulated results have not been available. The research presented in this thesis establishes a novel test facility capable of providing
such data and, for the first time, examines the behaviour of cavitating flows at the fuel injection pressures of current common rail fuel injection systems.

1.5 THESIS OVERVIEW

*Chapter 1* of this thesis has discussed the requirement for the continuing reduction in emissions from modern internal combustion engines. The optimisation of fuel injection equipment (FIE) is presented as a critical factor in meeting future legislation, highlighting the importance of understanding the processes that govern fuel atomisation. The theories of spray break up and cavitation, and the role of cavitation as a spray breakup mechanism, have been introduced.

*Chapter 2* reviews the current literature on cavitating nozzle flows, establishing the current understanding in this field of research; the wider field of cavitation research is also acknowledged. Focus is placed on experimental investigations, but a short review of computational approaches is also included. The limitations of the previous studies are evaluated, providing a basis from which the experimental techniques presented in this research can be compared.

*Chapter 3* describes the design and development of the patented high-pressure optical fuel injector rig. Also presented are the design of the optical nozzle assembly and an overview of the different nozzle geometries tested. Additionally a summary of the fuel injection rate tube’s operating principles is included.

*Chapter 4* introduces the imaging and illumination techniques used to capture the optical data from optical nozzle assembly. The methods developed to synchronise the imaging, illumination and data logging hardware to the rig’s operation are also described.

The experimental results discussed in *Chapter 5* focus on the cavitating flow phenomena observed in the nozzle-holes of both the single and multi-hole nozzle geometries. The results of the comparative study with two blends of biodiesel are also presented.

*Chapter 6* then builds on the observations made in Chapter 5 to provide a more detailed discussion of string cavitation formation in the sac volume. A new technique for acquiring quantitative flow velocity data from the optical nozzle is also presented.
Chapter 7 summarises the major conclusions of this research and outlines potential areas of further work

1.6 CONTRIBUTION TO KNOWLEDGE

The work presented in this thesis has made novel contributions to both experimental capabilities and physical understanding. Experimental capabilities include:

1. A patented test rig (“High pressure cavitation system”, US Patent Number 20100024537) that allows diesel fuel to be injected through an optical nozzle assembly at pressures up to 2000 bar (200 MPa) into a liquid or gaseous back-pressure of up to 150 bar. This experimental apparatus has enabled the study of cavitating nozzle flows up to the maximum working pressure of typical current common-rail fuel injection systems; far exceeding the pressure range of studies so far reported in literature.

2. The development of an imaging technique to allow simultaneous observation of physical flow structures and fluorescent particles in a seeded flow. This allowed the traditional technique of deriving fluid velocity from particle tracking methods to be applied to the two-phase flow and restricted geometry of the optical nozzle assembly.

The results achieved through the use of these experimental capabilities contribute significantly to the body of research on cavitating flows. These contributions include:

3. The observation of cavitating nozzle flows in true-scale injector flow geometries at fuel injection pressures up to 2050 bar. All previous experimental studies (to the author’s knowledge) have been limited to lower pressure flows or have used scaled geometries to obtain their results.

4. Imaging of a multi-hole-nozzle flow geometry at injection pressures up to 2050 bar under both transient and steady state injection conditions. This has allowed the behaviour of hole-to-hole variations and interaction to be observed across a comprehensive pressure range.
5. Defining the conditions that lead to the formation of string cavitation. Empirically derived geometric and flow property constraints are proposed.

6. A direct comparison between the cavitating properties of ultra low sulphur diesel and bio-diesel variants (B20 and B100) up to injection pressures of 2050 bar. The propensity to cavitate is observed with injections into both liquid and gas.

1.7 PUBLICATIONS ARISING FROM THIS WORK


CHAPTER 2

LITERATURE REVIEW
2.1 INTRODUCTION

This chapter reviews the current literature on cavitating flows, focussing primarily on experimental studies of cavitating nozzles and those closely related to fuel injection. The wider field of research into submerged-body cavitation is also acknowledged, as too are the computational studies that accompany much of the experimental work. Vortex cavitation and vortex-vortex interactions provide a background to the studies of string cavitation in fuel injectors. The current understanding of the topic is summarised and the need for further experimental study is discussed.

2.2 THE EFFECT OF CAVITATION ON DISCHARGE COEFFICIENT

In any real nozzle, under non-cavitating or cavitating flow conditions, the separation of the flow at the nozzle-hole entrance forms a vena contracta, effectively reducing the flow area of the nozzle, as illustrated by Figure 2.1.

![Figure 2.1 Schematic of the vena contracta formed in a nozzle-hole entrance](image)

The presence of the vena contracta, combined with friction losses in the nozzle, results in the actual nozzle flow-rate being less than the theoretical value for the same pressure drop. The discharge coefficient describes the ratio of these two flow rates by the following equation,

\[ C_d = \sqrt{\frac{\rho_l \bar{V}^2}{P_1 - P_2}} \]  

(2.1)
2.2 The Effect of Cavitation on Discharge Coefficient

where \( \bar{v} \) is the actual average flow velocity through the nozzle, \( \rho_i \) is the fluid density, and \( P_1 \) and \( P_2 \) are the upstream and downstream fluid pressures respectively. The studies of Bergwerk (1959) and Spikes and Pennington (1959) first highlighted the significant effect that cavitating flow has on the discharge coefficient of a nozzle. In both studies the cavitating flow was quantified by the Cavitation Number, defined as

\[
\sigma = \frac{P_1 - P_2}{P_2 - P_v} \tag{2.2}
\]

where \( P_1 \) is the upstream pressure, \( P_2 \) is the downstream pressure and \( P_v \) is the vapour pressure of the fluid. This can be further simplified to

\[
\sigma = \frac{P_1 - P_2}{P_2} \tag{2.3}
\]

to take account of the fact that \( P_v \) is generally negligible (\( \approx 5 \) kPa for diesel fuel) compared to \( P_2 \).

The dimensionless cavitation number describes the linear dependence between downstream pressure and the pressure difference at the inception of cavitation, but neglects any influence that geometry may have on the creation of low pressure regions. It cannot therefore be used to predetermine the cavitation inception conditions for a specific nozzle.

An alternative version of the cavitation number, called the cavitation index can also be used to define cavitating flow conditions. It is defined as

\[
\sigma = \frac{P_1 - P_v}{1/2 \rho U^2} \tag{2.4}
\]

This version is more often used for external flows around solid bodies where \( \rho \) and \( U \) are the upstream fluid density and velocity respectively. For this form of the cavitation number, low values signifying a higher likelihood of cavitation. The definition described in Equation (2.2) is better suited to describing conditions in a nozzle flow and therefore this version will be used throughout this study.
2.2 The Effect of Cavitation on Discharge Coefficient

Spikes and Pennington plotted the discharge coefficient as a function of cavitation number and Reynolds number for a range of 1.6 mm Perspex nozzles with different length to diameter (L/D) ratios. The inception of cavitation in a nozzle with L/D = 1 brought about an instant change of up to three per cent in the discharge coefficient. Similarly for an L/D = 2 a change in cavitation number from 1.5 to 6.5, saw a 17.5 per cent reduction in the discharge coefficient. Their experiments consistently showed that for a cavitating nozzle, variation in discharge coefficient was almost entirely dependent on changes in the cavitation number, with large variations in Reynolds number having little effect.

Bergwerk (1959) presented similar results, plotted the same way as Spikes and Pennington, with the discharge coefficient plotted as a function of the cavitation and Reynolds numbers. The data confirmed the change in dependency for the discharge coefficient from the Reynolds number to the cavitation number upon the inception of cavitation, or the cavitation boundary. Below this boundary, the discharge coefficient contour lines run mainly in a vertical direction, displaying a dependence on the Reynolds number. Above this boundary, as was also noted by Spikes and Pennington, the contours run approximately constant, as the discharge coefficient becomes dependent on the cavitation number.

Soteriou et al. (1993, 1995) carried out work on both large and real scale nozzles and showed that the coefficient of discharge was independent of the nozzle scale, also concluding that cavitation number controlled the nozzle discharge. Nozzle choking was claimed to have been observed during their submerged nozzle testing, whereby further decreases in the downstream pressure did not result in a change to the flow-rate once a specific, critical, cavitation number had been reached. It was deduced that as the cavitation extended across the entire cross section of the nozzle, and the flow became two-phase, a radical drop in the sonic speed of the flow was induced, allowing choked flow to occur at flow velocities well below the sonic speed of the liquid fuel. An alternative explanation for this observation is provided by the subsequent experimental study of Chaves et al. (1995), which directly references the work of Soteriou et al. and is further substantiated by the one-dimensional modelling.
2.2 The Effect of Cavitation on Discharge Coefficient

work of Schmidt and Corradini (1997). Chaves et al. observed similar trends in the discharge coefficient but also provided local flow velocity measurements across the hole outlet. Results showed that flow velocities near the walls were close to the mean exit velocity, calculated from the overall discharge, whilst the centreline velocity of the nozzle flow was far higher than the mean. Therefore, the velocities calculated from the discharge measurements, using the geometric hole area, were not a measure of the true exit velocity. Thus, two-phase choking was ruled out as local flow velocity increases were still observed within the nozzle-flow. The one-dimensional modelling approach taken by Schmidt and Corradini (1997) neglected wall shear between the vena contracta and the nozzle exit to represent the presence of vapour in this region. It was shown that the effective flow area of the nozzle tended towards the flow area of the vena contracta with increasing cavitation number, creating a jet with significantly smaller flow area and correspondingly high flow velocity. This produced effective flow velocities that agreed with discharge data and local jet velocities that agreed with the findings of Chaves et al. As a result it could be concluded that the discharge coefficient tended towards a value that represents the fraction of the nozzle area occupied by the jet.

The distinctive change in the nozzle discharge coefficient caused by the onset of cavitation has been used by Payri et al. (2004, 2008) and Schmidt et al. (1995) to determine the transition from non-cavitating to cavitating flow in metal (i.e. non-optical) nozzles. Characterisation of the nozzle exit flow allowed the onset of cavitation in the non-optical nozzles to be inferred by the observed reduction in mass flow-rate and effective velocity.

The studies of Park et al. (2006) and Lee et al. (2006) compared the performance of a new piezo-driven diesel injector against that of a more common solenoid-driven design. The piezo-driven injector was observed to have a shorter nozzle opening time, which resulted in a more rapid acceleration of the flow in the initial stages of the injection. This was observed to increase cavitation levels during the opening phase of the needle, which resulted in increased exit velocities and increased spray tip penetration. Lee et al. (2006) include a computational modelling study of the
work, carried out using the code described in Giannadakis (2005), which confirmed the increased levels of cavitation brought about the more rapid transient. Therefore, the results of these studies suggest that realistic experimental and computational results must also take into account the influence of the transient behaviour of the injection characteristics.

2.3 INTERNAL NOZZLE-FLOW CHARACTERISTICS

Many studies regarding nozzle cavitation relate their work to diesel injector nozzles. Trying to recreate the small dimensions and high pressure transient nature of a real diesel injector has proved difficult for optical visualisation and has meant that many different approaches to this area of study have been taken. Low pressure optical work is a common method of visualising the cavitation structures within the nozzle. This has been carried out on both real scale and, as is more often the case, large scale models. By matching real scale Reynolds and cavitation numbers in large scale models, the flow velocities, and hence driving pressures required, are much reduced and visualisation of the flow field within the nozzle becomes easier.

Soteriou et al. (1993, 1995) and Arcoumanis et al. (2000, 2001) have carried out experimental work on both real and large scale nozzles. They found the large scale models give sufficiently similar results to the real scale nozzle to allow useful, more detailed data to be obtained. Both studies observed cavitation inception at similar Reynolds and cavitation numbers for both nozzle sizes. In comparing their results with those of Soteriou et al., Chaves et al. (1995) noted that although cavitation inception may occur under similar flow conditions, the structure of the cavitation itself did not scale. It was noted that the bubble transit time in a real scale nozzle is of the order of the length of the nozzle. As a result it is not possible to scale the ratio of bubble collapse time to bubble transit time; nor is it possible to scale the relative diameter of the bubble to the nozzle diameter. Arcoumanis et al. (2000) and Martynov et al. (2006) highlight further limitations to classical scaling theory when applied to cavitating flows. Parameters such as the wall roughness and temperature, as well as the properties of the liquid-wall interface, and liquid quality cannot be scaled by the dynamic flow similarity employed in large-scale nozzle-flows.
Ganippa et al. (2001) observed the developing structure of cavitation phenomena using a large scale nozzle, with the hole perpendicular to the nozzle axis to produce an asymmetric inlet flow at the hole entrance. They also report that with increasing cavitation number, there is a transition from travelling bubble cavitation at inception through a stage of cloud cavitation, which increases in density before the final cavitation sheet stage was reached. The link between cavitation and enhanced spray breakup was also observed through the asymmetry of the nozzle flow. Cavitation formation was more prominent on one side of the hole, due to the asymmetric flow, and this correlated with an increase in spray breakup on the same side of the spray. Shedding of the cavitation from the cavitation cloud was also linked to periodic increases in the spray dispersion.

Gavaises et al. (2002) used 20x scaled up models of Bosch six-hole mini-sac and Valve Covered Orifice (VCO) injectors to carry out a detailed study of the cavitation structures in both the sac and the nozzle-holes, to determine the correlation between the cavitation structures and the flow turbulence. As they were unable to recreate the fast opening of the needle, which occurs in real sized production injectors, they were forced to operate under various fixed needle-lift conditions and rely on the rapid operation of the pumps to simulate the transient flow at the start of injection. Using a working fluid that refractive index matched the acrylic nozzle (32% Tetraline (1,2,3,4 – Tetrahydronaphthalene) 68% oil of Turpentine by volume) and high-speed digital video, they were able to capture the cavitation inception. They noted that in the multi-hole arrangement, in all cases investigated, cavitation inception did not occur simultaneously in all holes, but rather occurred randomly hole-by-hole with short delays. They observed how the structure of the cavitation within the nozzle then developed, with increasing cavitation number, from the incipient bubble flow through pre-film to film stage cavitation. These stages had been previously observed (Arcoumanis et al., 1999, 2001) and describe how the cavitation makes a transition from individual bubbles to the formation of large vapour pockets. In nozzle arrangements such as these the flow enters the nozzle asymmetrically. This results in asymmetric pressure distribution in the nozzle entrance and the formation of a single cavitating region on the “upper side” of the nozzle.
Point velocity measurements, obtained via laser Doppler velocimetry (LDV) in the study by Roth et al. (2002), were taken at regular intervals in the non-cavitating (lower) region of the nozzle and used to determine the Turbulent Kinetic Energy levels present. Their results supported the hypothesis that the occurring break-up of the cavitation structures is driven by the turbulent kinetic energy and therefore reduces turbulence levels in the bulk fluid flow while it transits through the nozzle. Cavitation induced turbulence, originating from the nozzle inlet region, is predominantly consumed by the bubble break-up process. The RMS velocity levels downstream of the cavitating region thus appeared relatively constant and independent of cavitation number. As the liquid moved downstream turbulence levels in the bulk fluid flow reduced asymptotically to the level of a non-cavitating flow. It was determined that bubble transit time was not sufficient to allow violent bubble collapse to increase flow turbulence within the nozzle.
More realistic flow conditions were studied by Badock et al. (1999), who observed cavitation in axisymmetric real-scale, single-hole optical nozzles at fuel injection pressures up to 600 bar (60 MPa). A Bosch common rail fuel injection system was used to provide transient fuel injection events and a test-oil, with similar fluid properties to diesel, was used as the working fluid. By using a light sheet method of illumination they were able to determine a continuous liquid core in the nozzle flow, surrounded by a thin annulus of cavitation film. This was found to be present at fuel injection pressures up to 600 bar, leading to the hypothesis that an intact liquid core leaves the nozzle-holes at diesel-like pressure conditions.

The only studies available in the literature that claim to have observed cavitation in real-scale nozzles at pressures of 2000 bar are those of Li (1999) and Li and Collicott (2006). These studies, which both use the same experimental data, provide images of the flow inside 200 µm diameter nozzle holes at fuel injection pressures up to 2200 bar. This was achieved using a pressure intensifier to amplify the fluid pressure of 70 bar, which was applied to the low pressure side of the intensifier by way of a compressed nitrogen supply, to inject fuel through acrylic orifice plates. It is not made clear how the authors managed to maintain fluid pressures of this level in simple acrylic nozzles without mechanical failure of the orifice plates. Images of the cavitating flow were limited by the 2 µs exposure time of the xenon-lamp illumination. For the flow velocities created, this length of exposure did not sufficiently freeze the flow motion; therefore, the authors relied on the mean-flow data that was captured using this method. By using an orifice tilted at 14° to the vertical, the authors were able to create an asymmetric hole-inlet flow, similar to that found in real fuel injectors. This slight asymmetry was sufficient for cavitation to form more readily on one side of the nozzle-hole than the other. In one example, a section of surface roughness is visible on the inner surface of the nozzle-hole, halfway down its length. Cavitation was seen to form off this region under conditions where cavitation was not formed at the hole-inlet. This led the authors to suggest that optically polished nozzle holes have the potential to suppress the formation of cavitation, which may otherwise form from surface imperfections in real nozzles.
2.4 NOZZLE GEOMETRY EFFECTS

The conical shape factor, or k-factor, of a nozzle describes the constant increase or decrease in the nozzle-hole diameter along its axis. The k-factor is usually defined as follows:

\[
k = \frac{D_{\text{inlet}}[\mu m] - D_{\text{outlet}}[\mu m]}{10}
\]  

(2.5)

Thus, converging nozzle designs are said to have a positive k-factor and divergent designs a negative k-factor. Blessing et al. (2003) present the results of a study looking at the effect of k-factor on cavitation formation in real-size optical nozzles at fuel injection pressures up to 800 bar. Three different nozzles with k-factors of -2.5, 0 and 2.5 were tested. Results showed that the convergent nozzle more readily suppressed cavitation formation, even at injection pressures of 800 bar. This observation was shown to correlate to a reduction in the ensuing spray angle, when compared to the results of the straight divergent nozzle designs. Furthermore the amount of cavitation correlated directly with the k-factor of the nozzle, with increasing levels of cavitation observed as the k-factor was reduced. For the divergent design (k = -2.5) the deviation of the nozzle wall, away from the nozzle axis along it length, promoted flow detachment and caused cavitation, in contrast to the convergent design where the opposite was true.

The effects of internal nozzle geometry were also investigated by Benajes et al. (2008), who conducted a study to determine the role of nozzle-hole convergence and cavitation in diesel combustion and pollutant emissions. Cavitating flow conditions of the non-optical fuel injector nozzles were again determined by observing the change in nozzle discharge coefficient. Converging and diverging nozzle designs with k-factors of -1.00, 0.38 and 1.47 were tested to measure their impact on fuel/air mixing time and resultant production of NOx and PM emissions. The presence of cavitation was found to greatly enhance mixing rates, more so when it occurred in convergent (+k) nozzles, due to the reduced effective flow area and increased jet velocities. Likewise, the reduction in fuel-rich regions, as a result of increased air entrainment in the flame lift-off length, meant that the levels of PM formation were
also reduced in convergent cavitating nozzles. As NOx formation was dependent on the flame temperature no clear correlation between nozzle design and the amount of NOx formed could be determined.

The previously mentioned study of Blessing et al. also applied hydro-grinding to the nozzle inlets. This treatment rounds-off the inlet edge producing an edge radius that smooths the pressure drop experienced by the fluid as it enters the nozzle-hole. This has the effect of reducing the pressure losses of the nozzle and thus increases the nozzle flow-rate and discharge coefficient for a given flow condition. Blessing et al. characterised the amount of hydro-grinding by the increase in the nozzle flow-rate that was observed after its application. Increases of 10.7%, 11.1% and 9.9% were achieved for the divergent, straight and convergent nozzles respectively. Payri et al. (2002) carried out a numerical study of the effects of inlet rounding and nozzle conicity. The extended volume of fluid (VOF) simulation results also showed a likewise increase in discharge coefficient, from 0.65 to 0.74, with increasing inlet rounding radius. Conducive to this result, it is also observed that inlet rounding reduces the formation of cavitating flow in the nozzle-hole. Thus it is deduced that inlet rounding is a significant control mechanism in the formation of cavitation. The suppression of cavitation formation in convergent nozzle design is also observed, and the authors suggest that convergent nozzles represent a good solution if cavitation is to be eliminated from nozzle design.

Hole-to-hole flow-rate variation was observed in a large-scale multi-hole VCO type injector by Arcoumanis et al. (1998) and Arcoumanis and Gavaises (1998). By varying needle eccentricity at fixed needle lifts, this variation was shown to be caused by deviation from the nominal geometric characteristics of the nozzle, within the manufacturing tolerances, that resulted in a non-uniform flow distribution within the sac volume. The changes in localised flow-rate within the nozzle had analogous effects on the cavitation development within each of the holes. Flow-rate changes attributed to the needle eccentricity and cavitation were also found to be a function of the needle lift, with the highest variations present at lower needle lifts. Subsequent studies by Bae and Kang (2000) and Karimi (2007) have shown this variation to
manifest itself as a variation in spray characteristics in real fuel injectors. Variation was attributed to transverse movement of the needle during opening, and showed similar reductions at higher needle lifts. Bae (2002) showed that this variation could be reduced by using a double guided needle to minimise its transverse movement.

2.5 EFFECT ON SPRAY BREAK-UP

Studies have noted the importance of cavitation as a primary spray breakup mechanism and its effect on the spray (Schmidt and Caorradini, 2001). The formation of a two-phase flow within the nozzle itself means that spray breakup is initiated before the flow exits the nozzle. This advances the secondary breakup of the spray when the exiting droplets and ligaments are subjected to the aerodynamic forces of the surrounding gas.

One of the first studies to highlight the influence of the nozzle on jet breakup was that of Bergwerk (1959). He observed cavitation in transparent acrylic nozzles and highlighted the importance of nozzle geometry in controlling the point of inception and amount of cavitation. He noted that through prolonged use at higher pressures, the entrance to the acrylic nozzle-hole became worn, increasing the pressure ratio required to achieve a cavity that extended the length of the nozzle-hole.

In his large-scale \((D = 2.5 \text{ mm})\) nozzle, he observed a phenomenon called hydraulic flip whereby the flow within the nozzle become completely detached from the nozzle walls. Under these conditions the fluid exits the nozzle as an unbroken, smooth column of liquid. On smaller scale work he found this phenomenon harder to reproduce, which he attributed to the effect of minor manufacturing defects being magnified and disrupting the flow. Although his study did not take a detailed look at the influence of cavitation on the ensuing spray, he concluded that the presence of cavitation had an obvious effect on the “ruffling” of the spray.

Saliba and Champoussin (2005) studied the effect of nozzle geometry and cavitation on the spray. They tested cylindrical, converging, diverging and converging-diverging nozzles and showed that the geometry of the converging nozzle suppressed
2.5 Effect on Spray Break-up

the formation of cavitation, and subsequently produced a spray angle that was smaller than the other nozzle types, where cavitation was present.

In their review of cavitation studies Schmidt and Corradini (2001) highlight the work of several research groups that have established the importance of cavitation on the spray. All have observed a marked change in the spray behaviour upon the inception of cavitation. As with Saliba and Champoussin, an increase in spray angle was observed, along with a decrease in the jet breakup length. There are differing opinions as to whether the energy released during bubble collapse in the exiting nozzle flow contributes to the primary spray break-up, either by increasing the turbulent kinetic energy of the jet or by causing direct localised jet break-up. Chaves et al. (1995) isolated the effects of cavitation on the spray by studying cavitation in low velocity flows. By keeping the Reynolds number low (Re ~ 2000 – 4000) and the length of the nozzle short, flow turbulence was given no time to develop. They also minimised the aerodynamic effects on the spray by injecting into atmospheric pressure. Where cavitation did not reach the nozzle exit, the jet remained smooth. Not until cavitation reached the nozzle exit (supercavitation) did the surface of the exiting jet become roughened. From this he concluded that cavitation causes finite disturbances that lead to the immediate atomisation of the spray. Further, turbulent-flow work showed that the spray angle reached a maximum, for a set back-pressure, when cavitation extended to fill the length of the nozzle. Further increases in pressure difference brought about minimal increases in spray angle. Only by increasing the back-pressure, and therefore density of the surrounding gas, were increases observed. It is, therefore, aerodynamic interaction with the surrounding gas that determines the outcome of the atomisation process once supercavitating conditions have been reached.

A study of the effects of cavitation on spray development using a large-scale 2D nozzle has been carried out by Sou et al. (2006, 2007) and Suh and Lee (2008). They each observe a significant increase in the atomisation and spray angle of the spray at the point where cavitation extends the entire length of the nozzle hole. Before this, reattachment of the separation boundary layer at midway points down the hole length caused unsteady wave formation on the exiting spray. LDV measurements confirmed an increase in exit flow velocities under cavitating conditions.
Goney and Corradini (2000) observe the spray development from non-optical VCO injector nozzles to identify the effects of ambient pressure, cavitation number and hole-inlet type on the spray characteristics. The results obtained from a ‘sharp’ hole inlet and a rounded inlet, with a radius of 0.05-0.06 mm, were compared and similar results to the previous studies were forthcoming. Discharge coefficients for the rounded inlet were always higher than those of the sharp inlet, and spray angles for the sharp inlet were larger than the rounded inlet cases – a sign of the cavitating nozzle flow.

Fath et al. (1997) used a laser-sheet illumination technique to investigate the spray structure close to the exit of a real-scale nozzle. Cavitation bubbles were observed to implode upon exiting the nozzle, physically disturbing the flow and increasing the air entrainment into the developing spray. They used these observations to suggest a new two-phase model for spray breakup that simultaneously included cavitation bubbles in the liquid column. Based on the observations, this model also assumed that cavitation bubbles persist for several microseconds outside of the nozzle. A double-pulsed laser technique was thus employed to measure the velocity of bubbles entrained in the liquid jet near the nozzle exit. Results agreed with the previous observations, with greater axial velocities measured closer to the nozzle axis, however, with increasing radial distance the radial velocity was measured to increase, capturing the aerodynamic deceleration of the jet’s outer surface. Bae and Kang (2006) also used laser-sheet illumination to observe the surface and internal structure of the transient diesel spray from a 5-hole VCO type injector, using Bosch common-rail fuel injection hardware to achieve injection pressures up to 1120 bar. They produced a phenomenological model of the spray that included the formation of spray ligaments in the internal flow structure. The authors link this breakdown of the flow structure to the collapse of cavitation bubbles at the nozzle exit, which they observed as discontinuities in the scattered light intensity emitted from the core region of the spray.

Payri et al. (2009) imaged the cavitating flow exiting from drilled steel plates by injecting diesel into a liquid environment. By backlighting the exiting flow they were able to observe the cavitation bubbles as they exited the nozzle-hole. Although the
2.6 Cavitation Damage

authors concede that the spray behaviour will differ from that of an injection into a gaseous medium, cavitation was still observed to have a measurable effect on the spray angle when it occurred. They also observed the hysteresis characteristics of cavitation inception. Previous studies (Spikes and Pennigton, 1959; Brennen, 1995; Mishra and Peles, 2005) have observed hysteresis in the value of the cavitation number for inception. Spikes and Pennigton observed that for a given driving pressure, when the downstream pressure was reduced then increased, the start and end of cavitation occurred at different downstream pressures. This can be related to the nucleation theory of cavitation inception, where cavitation bubbles already present in the nozzle flow act as nucleation sites. This causes cavitation to persist for an increased pressure range with increasing downstream pressure, and can lead to significantly different results depending on the direction of the pressure change.

2.6 CAVITATION DAMAGE

Chaves et al. (1995) also discuss the effects of the nozzle-hole inlet geometry on the development of cavitation in the nozzle. Although the study did not involve the use of rounded inlet nozzles, the discussion focuses on the effects of cavitation damage to the internal surfaces of the nozzle, and the resulting effects that this can have on the nozzle’s cavitating characteristics. Chaves et al. cite the work of Prescher and Schaffitz (1979) who observed cavitation damage to the needle seat and the inlet corner of the nozzle holes. Due to the transient flow geometry created by the opening and closing of the needle, the smallest flow cross-section is formed in this region of sac at the start and end of injection. The cavitation collapse induced in this flow region caused imperfections to form on the rounded inlets to the holes. This led to the conclusion that even if cavitation formation has been mitigated against by well rounded hole-inlets, after a period of operation cavitation is still likely to appear.

A study of cavitation damage in VCO and mini-sac type fuel injectors is presented by Gavaises et al. (2007). The internal surfaces of both injector designs were analysed by scanning-electron microscope after they had operated at fuel injection pressures up to 2000 bar for a sustained period of time. The cavitation damage caused to each injector can be seen in Figure 2.3 (VCO injector) and Figure 2.4 (mini-sac injector). The geometries of each injector were then recreated as scaled-up
optical models. The location of the cavitating flow structures observed coincided with the regions of surface damage exhibited by the real injectors. Computer modelling of the flows, using an advanced in-house cavitation model, successfully replicated the flow conditions and allowed a redesign of each nozzle geometry, which were then subsequently shown to be erosion free after further testing.

Erosion to the upper surface of the nozzle-hole entrance region in the VCO injector, shown in Figure 2.3(b), was consistent with the formation of geometric cavitation in this flow region. Importantly, the observed damage in Figure 2.3(c) to the ‘3 o’clock’ and ‘9 o’clock’ positions of the hole inlet, upstream of the injection hole inside the sac volume, was attributed to the formation of ‘string cavitation’, proving that this form of ‘dynamic cavitation’ was also capable of causing physical damage to the injectors internal surfaces. The studies of string cavitation will be addressed later in this chapter.

For the mini-sac type injector, significant erosion was apparent on the tip of the needle and also on the adjacent surface of the sac volume upstream of the nozzle-holes, which exhibited far less erosion that the VCO nozzle-holes. The damage to the
Cavitation formation processes have also received a significant amount of attention in flow environments other than those confined to nozzle flows. The analysis of cavitation formation around immersed bodies such as hydrofoils and propellers is essential if their performance is to be optimised and maintained. In slight contrast to nozzle flows, the presence of cavitation in such applications is almost always undesirable, and therefore the knowledge of cavitation is built around the desire to engineer it out of a component’s design.

The formation of cavitation off hydro-mechanic surfaces is often referred to as ‘fixed’, ‘attached’ or ‘sheet’ cavitation (Ceccio and Brennen, 1992; Foeth et al., 2006). This form of cavitation is associated with negative performance effects such as propeller thrust breakdown and cavitation erosion damage in hydraulic
Cavitation in External Flows

turbomachinery and marine propellers. It is also responsible for dangerous propeller-hull interactions and the production of unwanted acoustic emissions.

The formation of attached cavitation occurs when a liquid flow detaches from the leading edge of a lifting body, forming a vapour cavity in the immediate downstream region of the flow. Research efforts have focused on being able to predict the location of detachment and the flow conditions at inception (Arakeri and Acosta, 1973; Arakeri, 1975; Franc and Michel, 1985; Tassin Leger and Ceccio, 1998; Tassin Leger et al. 1998; Farhat and Avellan, 2001; Foeth et al., 2006) as well as the geometry of the cavity that is produced (Katz, 1984; Ceccio and Brennen, 1992; Tassin Leger et al., 1998). From these studies it has been shown that the cavity does not detach from the body at the minimum pressure point, but behind a laminar separation boundary that precedes this point.

Pereira et al. (2004) used an image correlation technique to extract the cavitation features from images of a cavitating propeller. By cross correlating a non-cavitating background image with the images captured during experimentation, they were able to clearly identify the cavitating vortices from the blade tips and the contour of the attached cavitation. Comparing these results to their theoretical hydrodynamic model, they were able to accurately predict the sheet cavitation area across a wide range of advance ratios and cavitation number values. For higher values of cavitation number it was observed that the sheet cavitation interacted with the tip vortex structure to form a leading-edge cavitating vortex. Vortex cavitation is a complex phenomenon that is common to many hydrodynamic applications, and is responsible for many of the negative effects of cavitation in such flows.

2.7.1 Vortex Cavitation

Regions of concentrated vorticity are common in many high Reynolds number flows of practical importance, ranging from eddies that are randomly distributed in space and time in turbulent flows, to the well defined and relatively stable vortices that, as previously mentioned, form on propeller tips. A comprehensive study of the fundamentals behind rotating flows and their wide ranging practical examples is
provided by Vanyo (2001). The review highlights the critical importance in understanding and predicting rotating flow phenomena in many areas of engineering and science, where the rotating fluid is either the principle phenomena of interest or an undesirable result of an object’s flow interaction.

The topology of Rankine’s combined vortex model is described, which provides a close approximation to many real vortex-like motions. The model, illustrated in Figure 2.5, consists of a rigidly rotating, or forced vortex, core with radius $R$ surrounded by an inviscid, or free vortex, with matching values of angular velocity, $V_\theta$ and fluid pressure, $P_f$ at $R$.

![Figure 2.5 Rankine’s combined vortex model – a rigidly rotating core surrounded by an inviscid vortex](image)

The static pressure at the vortex core is lower than that of the surrounding fluid, and for certain cases where the vortex circulation, $\Gamma$, is large enough, the core pressure may drop below the liquid vapour pressure. Thus, the inception of vortex cavitation
can occur when a small bubble or nuclei is exposed to the increased fluid tension that exists in the core. An extensive review of cavitation processes in vortical flows is provided by Arndt (2002) in which the formation and effects of various cavitating vortex structures are discussed. Similar to the collapse of cavitation bubbles in nozzle flows, vortex cavitation is suggested to be very damaging during the collapse phase. The formation, oscillation and collapse of vortex cavitation bubbles are studied in greater detail by Choi and Ceccio (2007) and Choi et al. (2009). It is shown that as an initially spherical bubble nucleus grows into an elongated bubble it may exhibit complex surface deformations and have volume oscillations that scale with a product of the core radius and the core tangential velocity. The volume history of the bubbles is shown to be highly sensitive to small perturbations in the flow conditions and thus the detailed, three-dimensional growth of the bubble has a significant influence on the final bubble shape. Therefore, whilst traditional scaling variables may be used to scale cavitation inception, they are insufficient for scaling the fully developed vortex cavitation.

Belahadji et al. (1995) provide evidence of three distinct forms of rotational structures in the cavitating turbulent wake of a wedge-shaped obstacle: primary span-wise vortices, secondary stream-wise vortices and finally near-wake vortices. Cavitation in the stream-wise vortex cores was found to physically affect the flow, compared to non-cavitating conditions, by decoupling the stretching and rotation rate of the flow structures. This finding was also observed by Iyer and Ceccio (2002).

The subsequent interaction of vortices with one another in the wake of lifting surfaces has received specific attention in the case of flapped aerofoils, due to their prominent application to aircraft design. Vortices, forming off the wing and flap-tips roll up with near-wake vortices formed in the span-wise wake of the wing (Choi et al., 2003) creating a single dominant vortical flow structure in the downstream flow region. Chen et al. (1999) study the interaction of the co-rotating vortices that are formed off the wing and flap-tip on one side of a wing. These two vortices are shown to merge, as illustrated by Figure 2.6, creating a single dominant vortex which counter-rotates with respect to the resultant vortex formed off the opposing side of
the wing. This vortex system poses a safety risk in the form of aircraft-vortex interactions, and is a primary reason for take-off and landing scheduling at airports. Using PIV it was shown that merging of the two co-rotating vortices occurred within 0.8 orbit times and that this time scale was insensitive to changes in the circulation Reynolds number, $Re_{\Gamma}$ ($Re_{\Gamma} = \Gamma/\nu$), and the relative strength ratio, $\Gamma_f/\Gamma_t$ of the two vortices.

The merger of two originally parallel, co-rotating vortices is studied in a water tunnel by Chang et al. (2007), who visualise the merger by way of the cavitation vapour at the vortex cores. During the orbit of the two vortices they observed vorticity bridging between the vortex streams upstream of the merger point, revealing the complex vortical interactions that are established during the merger’s initiation. As the cavitation number was reduced, the bridging phenomena became more frequent and spanned the flow region between the main vortices. At the lowest cavitation number this flow region became filled with an incoherent bubbly mixture, suggesting high levels of fine-scale vorticity and the transfer of vapour between the two vortex structures. Ortega et al. (2002) observed the interaction of unequal-strength counter-rotating vortex pairs using dye injection and PIV. In this study triangular flaps were used so that the wing-tip and flap vortices were counter-rotating. The weaker flap-vortices were observed to wrap around the stronger tip-vortices, eventually deforming into “Ω-shaped” vortex loops which shed into the span-wise wake – an observation that was successfully reproduced using a numerical modelling approach by Bristol et al. (2004).
The studies detailed here highlight the complex interactions that occur when both co-rotating and counter-rotating vortices form in close proximity to each other. These interactions have been observed in free-stream ‘unbounded’ conditions, where the flow interactions are not affected by geometry constraints - except in their initial formation. The formation of vortex cavitation in the confined geometry of injector nozzles, however, has not been observed until recently, and its effects on the nozzle flow are yet to be fully established. This is owed in part to the complexity and scale of the injector geometry, which makes detailed results gathering significantly more difficult. The limited experimental and computational studies of this particular phenomenon, which have for the most part been produced by Arcoumanis, Gavaises and their research group, will now be discussed, since this contributes significantly to the work contained in this thesis.

2.8 STRING CAVITATION

The formation of vortex cavitation within the sac volume of a fuel injector was first observed by Kim et al. (1997) and has since been observed in greater detail by the research group headed by Arcoumanis, who termed it ‘string cavitation’.

This cavitation phenomenon has been observed in large scale mini-sac and VCO type multi-hole injectors and also in real size nozzles. Kim et al. (1997) make brief reference to the formation of a “column type of cavitation” that formed between the discharge hole and the needle tip of a 10x-scale mini-sac injector geometry. They observed the “column” extending to the exit of the hole at a fixed needle lift of 0.7 mm, where its formation was accompanied by a marked change in the spray structure to a hollow-cone with very fine droplets. Arcoumanis et al. (1999) provides a more in-depth analysis of the formation of this type of cavitation in a 20x-scale 6-hole VCO type fuel injector. They note how its formation originated well inside the sac volume as a bubble stream, forming a continuous string that extended into the flow region between adjacent holes. This occurred in areas of the flow where the local pressure was otherwise expected to be equal to that of the high upstream pressure.
String Cavitation

The instantaneous flow structure was observed by tracing the movement of small bubbles that were introduced into the upstream flow. Despite the axisymmetric geometry of the nozzle, a cross-flow was observed in the sac volume, linking one side of the sac volume to the other. The formation of this cross-flow was attributed to the intermittent throttling of individual holes, by either cavitation or existing recirculation zones. This cross-flow then interacted with the high momentum annulus flow, created by the fluid entering the sac around the needle, and resulted in the coalescence of cavitation bubbles in the low pressure vortex core that formed, taking on the appearance of a vapour ‘string’.

It was observed that these strings developed transiently and intermittently between adjacent holes with the ends of the vortex extending into the inlets and diffusing into a conical bubble cloud. This then mixed with the geometric film cavitation structures further downstream. Similar to Kim et al. (1997), it was also shown that the position of the string formation was dependent on needle lift; however, string formation was observed to form across a range of needle lifts, albeit at lower frequency for lower lift levels. Testing in real-scale nozzles by Arcoumanis et al. (2000, 2001) revealed the formation of string cavitation in actual nozzle geometries. Similar interaction with the geometric cavitation in the nozzle-holes was observed, although in this geometry formation was only observed to occur at high needle lifts.

Gavaises et al. (2002) study the flow in a real-scale optical nozzle at fuel injection pressures up to 700 bar. One of the six injection holes in the standard fuel injector was replaced with a quartz window containing a single 0.17 mm hole that was aligned to the original hole position. They observe the swirling motion of the geometric cavitation in the nozzle holes, but did not identify the formation of cavitation strings. The results of Roth et al. (2005) capture the geometric cavitation in the same geometry at rail pressures up to 1350 bar. It was noted that the increases in rail pressure did not bring about any significant changes in the structure of the cavitation in the nozzle hole. The increased flow velocities also resulted in the accelerated development of cavitation in the nozzle holes, which quickly filled the hole with vapour and limited the useful data that could be obtained. By only using a transparent nozzle hole, the upstream flow behaviour was not captured and string
cavitation that may have formed in the nozzle hole was obscured by the geometric cavitation vapour.

Gavaises and Andriotis (2006, 2009) and Andriotis et al. (2007) present data obtained from a real-scale 5-hole marine injector with hole diameters of 1.5 mm. The studies report a significant link between the formation of cavitation strings in the nozzle holes and an increase in the spray angle. This observation had previously been made by Kim et al. (1997), but further investigation was not forthcoming. Simultaneous imaging of the nozzle-hole flow and the exiting spray was carried out, allowing the cavitating conditions in the internal nozzle flow to be directly correlated to the ensuing spray characteristics. Figure 2.7 illustrates the magnitude of the effect that was observed when a cavitation string was observed to be present in the injector hole.

Studies were also undertaken using two-hole and single-hole nozzle geometries, which represented the side and middle holes of the five-hole design respectively. This simplified spray imaging whilst, in the two-hole case, maintaining the transient hole-to-hole interactions of the cavitation strings in the sac volume. Further variation in design was added by manufacturing each of these geometries with both cylindrical-hole and tapered-hole (+k) designs. As discussed previously, the tapered-hole design had the effect of minimising the formation of geometric cavitation in the nozzle hole, allowing the effects of string cavitation to be isolated. In doing so it was shown that the influence of string cavitation on the spray angle was greater when
geometric cavitation was suppressed. The transient intermittent nature of the string formation within the sac volume therefore, led to the conclusion that string cavitation was a source of cycle-to-cycle and hole-to-hole variation in diesel fuel injectors. Consequently it was suggested that tapered holes may exhibit higher levels of hole-to-hole variation than cylindrical holes that contained geometric cavitation.

Additionally, it was frequently observed by Gavaises et al. (2002) in the VCO type injector that strings would originate on the needle surface facing a hole and extend downstream into the hole. At higher cavitation numbers this vortex induced a corkscrew type flow through the hole and it was suggested that this maybe the cause of hollow cone sprays coming from VCO type injectors observed in previous studies (Soteriou, 1995). Swirling flows of this nature have been studied at real scale in the sac and nozzle-holes of VCO type injectors by Chaves and Schuhbauer (2006). Using an injector with fixed needle lift and glass nozzle tip they were able to observe the development of cavitation in two nozzle holes. The holes were angled differently with respect to the nozzle axis, and the hole with the largest flow deflection exhibited the highest levels of cavitating flow. This was in agreement with Arcoumanis et al. (1998) who observed a similar phenomenon in a large scale nozzle, also recording a lower discharge coefficient for such holes. The asymmetry of the flow into each hole was observed to induce high levels of swirl in the nozzle holes. This was shown to affect the structure of the geometric cavitation as it was convected towards the hole exit. In contrast to the Rankine vortex topology of cavitation strings, the vortex breakdown was observed to occur in a spiralling form, where the vortex core was wrapped around the nozzle axis – Chaves and Schuhbauer refer to this as ‘pig-tail’ cavitation. This observation correlates to the interaction of unequal-strength counter-rotating vortices observed by Ortega et al. (2002) and therefore suggests the presence of multiple vortices in the nozzle-hole flow.

2.8.1 Cavitation Modelling

Alongside the experimental work undertaken by the research group led by Arcoumanis, an advanced in-house cavitation model, based on the Eularian-Langrangian approach, has been developed and validated against their collected
2.8 String Cavitation

experimental data. A comprehensive description of this model can be found in Giannadakis (2005) and Giannadakis et al. (2008) in which the various physical flow parameter considerations are discussed in detail. Although computational modelling of cavitation does not form part of this particular study, it will be shown later in this thesis how the unique, high-pressure flow data obtained, were used to the validate the advanced cavitation model (Giannadakis et al., 2008) at realistic common-rail fuel injection pressures for the first time.

Numerous examples of computational models that attempt to predict cavitation are available in the literature. Schmidt and Corradini (2001) provide extensive coverage of the topic in their review of internal nozzle flow research. An updated comparison of different cavitation model approaches can found in Giannadakis et al. (2007). Most models are based on the theories of cavitation inception discussed in this and the previous chapter. Inception is assumed to be a mechanically driven process relying on the existence of nuclei in the liquid phase which, in the presence of certain fluid conditions, grow to form cavitation bubbles and the subsequent complex cavitating flow structures. The transport of vapour produced from this initial assumption is then treated as either a continuous cloud, simulated using an Eularian frame of reference, or as a collection of discrete bubbles, which are tracked by a Eularian-Lagrangian approximation (Giannadakis et al., 2007). In most studies the initial bubble growth and collapse mechanism is based on numerical simulation of the Rayleigh-Plesset (R-P) equation (Brennen, 1995; Giannadakis et al., 2007). Additional considerations include bubble number density changes, bubble coalescence, restrictions to bubble growth, initial bubble number/size distribution and internal bubble initialisation pressure. At present, experimental data does not exist for these parameters under diesel injection conditions; therefore validation is based on the macroscopic accumulation of these effects. Data can be gathered qualitatively in the form of flow visualisation, or quantitatively through discharge coefficient measurements and optical velocity measurement techniques such a particle image velocimetry (PIV) and laser Doppler velocimetry (LDV).
Recent descriptions of the in-house model’s capability are provided by Andriotis et al. (2008) and Gavaises et al. (2009). A wealth of ‘geometric cavitation’ data, and superior understanding of its formation mechanisms, mean that this particular cavitation phenomena can be simulated with considerable accuracy. The formation of string cavitation, however, remains less well understood and accurate simulation of its formation is yet to be established. The formation of vortical structures, which act as tracers for the formation of cavitation strings, can be realised with current model. However, the formulation of an accurate string cavitation sub-model is still required to produce a comprehensive nozzle-flow cavitation model. Giannadakis (2005) cites the many numerical improvements that are needed if the vorticity fields and their low pressure cores are to be successfully captured by the simulation. These include an increase in the accuracy of the turbulence models along with finer computational grids and an increased understanding of the interaction between bubbles/nuclei and vortices.

The work of this thesis attempts to be complimentary the modelling efforts of the City University research group. In doing so, a unique data set is produced, which observes the behaviour of true-scale cavitation phenomena across the working pressure range of current high-pressure fuel injection systems.

**2.9 SUMMARY**

Cavitation is a widely investigated flow phenomenon that can take many different forms depending on the flow environment. Studies of the internal flow in simple, single orifice nozzle-flows, as well as the more complex geometry of fuel injector nozzles, have revealed the important role that cavitation can play in determining the nozzle-flow and exit spray characteristics. The conflict between the benefits that cavitation can bring to fuel atomisation, and the negative effects that cavitation damage can have on hardware components, has meant that understanding cavitation processes is of great importance to engine and FIE manufacturers. Recent trends in diesel fuel injection systems have seen substantial increases in fuel injection pressures and reductions in nozzle hole sizes. The currently available studies of cavitation have so far not been able to provide good quality data of the flow
processes that occur in these extreme flow environments, with experimental scaling only providing restricted comparisons. Furthermore, new insights into the role of string cavitation in the nozzle-flow highlight the fact that the knowledge of cavitation phenomena is far from complete. Until now, these flow structures have only been viewed in the sac volume of large scale injectors at fluid pressures of only a few bar, and only briefly observed in real-scale nozzle-holes at a several hundred bar (Roth, 2004). To gain better understanding of their formation mechanisms and provide data to comprehensively validate computational models, it is necessary to observe their behaviour in detail under realistic flow conditions.

2.10 CONCLUDING REMARKS

This chapter has reviewed the current literature on cavitating flows. The need to experimentally investigate the behaviour of cavitation phenomena in realistic diesel fuel injection conditions has been presented. The following chapter discusses the design and development of the test facility that was established to achieve this aim.
CHAPTER 3

EXPERIMENTAL EQUIPMENT AND TECHNIQUES
3.1 INTRODUCTION

In the previous chapters, the current understanding of cavitation phenomena in high pressure fuel systems has been discussed. In most cases this understanding has been developed through empirical observations, which have defined the general flow conditions under which cavitation is more likely to occur. This knowledge has, in turn, led to the development of numerous computational models that aim to predict the occurrence and characteristics of cavitation at conditions beyond the capabilities of experimental apparatus. Although validated experimentally using lower pressures or Reynolds Number matching techniques, the lack of in-nozzle empirical data at actual fuel injection system pressures means these models have, to date, relied on assumptions when simulating realistic injection events. This chapter discusses the design and development of a test rig, capable of delivering high pressure fluid to a real scale optical nozzle, overcoming the limitations met in previous studies.

Acquiring optical data from real sized nozzles at realistic fuel injection pressures would provide valuable insights into the behaviour of previously observed phenomena at elevated pressures, and also enable the validation of computer models across the working pressure range of modern FIE.

By designing the rig around an idealised nozzle geometry, specific flow characteristics could be observed and measured using a range of optical diagnostics techniques.

A collaborative design effort between Caterpillar Engine Research Europe, Loughborough University and the test rig manufacturers (Test Development Innovators Llc. based in the USA) ensured that the target requirements and capabilities were met.

The resulting design was capable of delivering both steady-state and transient diesel flows to a real size optical nozzle assembly at fluid pressures of up to 2000 bar (200 MPa). Injections into liquid (diesel) or gas (nitrogen) were made possible.
through the use of a rate tube or injection chamber respectively, with each being capable of providing a back pressure of up to 150 bar.

### 3.2 RIG OVERVIEW

The test rig comprised of two cabinets, shown in Figure 3.1 and illustrated in Figure 3.2. The larger, main cabinet, housed the fuel tank, main drive unit and fuel pumps, pressure accumulator and the majority of the electronics and control hardware for the rig. The smaller, optical bench housed the clamp for the optical nozzle assembly and mounts for both the downstream Injection Chamber and the Rate Tube. Additionally it housed the electronic control module (ECM) and further electronic hardware. At the front, the two cabinets were connected by a gantry which routed the necessary cabling between the cabinets. Towards the rear, the units were connected by the fuel delivery and return pipes, along with compressed air and high pressure gas lines, which were routed along the floor.

![Image](image.jpg)

*Figure 3.1 The high-pressure optical diesel fuel injection rig, located in the Wolfson Annex at Loughborough University*
3.3 Rig Features

3.3 RIG FEATURES

3.3.1 Fuel Delivery

To provide the most realistic in-nozzle flow conditions, regular diesel fuel [ref: Table 3.1: Fuel Properties] was used as the working fluid for the experimental work.

Figure 3.1 shows inside the rear of the main cabinet where the majority of the Fuel Injection Equipment (FIE) was housed. Based on a standard common-rail fuel injection system, two Bosch CP3 pumps, driven by a common main drive unit (MDU), delivered fuel to a single 1.15 litre fuel rail. Pressurised fuel was then delivered to the injection fixture on the optical bench via flexible high-pressure piping. The increased capacity of the system allowed the rig to operate under steady-state conditions at its safe operating maximum pressure of 2000 bar (rail) where the required fuel flow rate is in the region of 10 litres/min.

Rail pressure, \( P_{\text{rail}} \), was controlled via a CAT A4V4E3 ECM and CADeTWin ECM control software, which provided electronic control for the entire fuel injection system.
3.3 Rig Features

3.3.2 Fuel Injector

As well as steady-state operation, the rig could be fitted with a modified CAT CR200 common-rail fuel injector. This allowed realistic transient injection events to be studied in the optical nozzle. The fuel injector tip was modified from the standard six hole design and replaced with a single large diameter hole, positioned axisymmetrically at the end of the tip.

This modification effectively turned the injector into a high speed valve, producing injection durations ranging from 1-4 ms which were then fed into the optical nozzle downstream. All injection parameters were controlled via the CADeTwin software package in communication with the ECM. An artificial engine speed, input by the user on the rig’s touch-screen control panel on the main unit, was sent to the ECM by a timing box. From this, the fuel injection frequency could be altered between simulated engine speeds of 100 rpm up to the rated speed of 2500 rpm. For a 4-stroke engine cycle, this gives fuel injection frequencies in the range of 0.8-21 Hz.

Initial transient testing was carried out using this “continuous firing” method whereby the fuel injector fired as it would if it were running in an engine. This had the effect of firing the fuel injector when no data was being captured, placing unnecessary stress on the optical components of the downstream nozzle assembly. At
lower pressures this effect had less impact on the durability of the nozzle, but the switch to running at high pressures towards the end of the test schedule meant effort had to be made to prolong the life of the components. This was achieved by adapting the hardware to run as a “fire-on-demand” system, where each injection was manually triggered by the operator. A relay switch, placed between the speed-timing box and the ECM, was manually triggered and held open long enough to allow the timing-signal for a single injection to pass to the ECM.

The implementation of this change also simplified the triggering of the data capture equipment, the details of which will be discussed in Section 4.4.

The injector was also capable of accepting up to four injection signals per cycle – pilot, close-coupled pilot, main and close-coupled post. The duration and timing of each was independently controllable and although not a focus of this project, gives future flexibility for observing flow phenomena in modern injection systems, where the ability to utilise multiple injection is becoming more common.

### 3.3.3 Fuel Temperature Regulation

The fuel system operated as a closed loop, with fuel-return pipes from either the injection chamber or the rate tube being routed back to the fuel tank. In addition to the main high-pressure fuel circuit, a further, temperature regulated low-pressure sub-loop, shown in Figure 3.5, constantly re-circulated fuel through the tank, maintaining the fuel temperature to ± 1°C between 15°C and 90°C. This gave the ability for model validation to be carried out against both fuel temperature and pressure effects. Controlled heating of the fuel was achieved via a heating element located in the fuel tank. Due to the amount of mechanical work exerted on the fuel during compression and injection, a certain amount of additional heating also took place on the high pressure side of the fuel circuit. To ensure a stable fuel temperature is maintained in the fuel tank, an external chiller unit provided a water/glycol mix maintained at 6°C to a solenoid controlled heat exchanger in the low-pressure sub-loop. A separate fuel pump maintained fuel flow around this sub-loop, allowing the
fuel temperature to be maintained regardless of whether the main drive unit was engaged.

![Figure 3.5: Schematic of the low-pressure, temperature-controlled fuel loop](image)

3.3.4 Fuel Filtering

Due to the closed loop nature of the fuel system and the requirement for fuel filtering, there was concern that over time the fuel could become “over-clean”. As discussed in the previous chapter, the presence of foreign particles, acting as nucleation sites within the fuel, is important from the point of view of fundamental cavitation and bubble formation theory, and as such, artificial nucleation sites form a critical component of the computer models used by City University. Removal of potential nucleation sites through repeated filtering of the fuel, therefore, could have created a potential source for ambiguities between the experimental and computational results. However, accurate identification and measurement of the nucleation site density is impossible beyond a certain point and therefore was not measured as part of this study. To limit the impact of this effect though, a fuel filter bypass valve was fitted to the fuel circuit to allow the fuel to circulate, unfiltered, for a period time and thus prevent a further decrease in the density of nucleation sites within the fuel.
3.3.5 Fuel Properties

The fuel tank, located in the main cabinet, has a capacity of 56 litres and was filled with standard ultra low sulphur diesel fuel. To ensure the relevant fuel properties were known for both experimental calculations and for input into the computer model, a sample was tested by Intertek Labs, Sunbury. Tested to traceable standards, the results are shown in Table 3.1

<table>
<thead>
<tr>
<th>Analysis</th>
<th>Standard</th>
<th>Results</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density @ 20°C</td>
<td>IP365</td>
<td>0.8299</td>
<td>kg/l</td>
</tr>
<tr>
<td>Density @ 30°C</td>
<td>IP365</td>
<td>0.8229</td>
<td>kg/l</td>
</tr>
<tr>
<td>Density @ 60°C</td>
<td>IP365</td>
<td>0.8017</td>
<td>kg/l</td>
</tr>
<tr>
<td>Surface Tension @ 25°C</td>
<td>ASTM D971</td>
<td>29.7</td>
<td>Dynes/cm</td>
</tr>
<tr>
<td>Air Saturated Vapour Pressure</td>
<td>IP394</td>
<td>5.0</td>
<td>kPa</td>
</tr>
<tr>
<td>Dry Vapour Pressure Equivalent</td>
<td>IP394</td>
<td>1.0</td>
<td>kPa</td>
</tr>
<tr>
<td>Kinematic Viscosity @ 20°C</td>
<td>IP71</td>
<td>3.916</td>
<td>cSt</td>
</tr>
<tr>
<td>Kinematic Viscosity @ 30°C</td>
<td>IP71</td>
<td>3.120</td>
<td>cSt</td>
</tr>
<tr>
<td>Kinematic Viscosity @ 60°C</td>
<td>IP71</td>
<td>1.830</td>
<td>cSt</td>
</tr>
</tbody>
</table>

These properties, selected through discussion with City University, provided the model with actual fuel data from the experimental working fluid, and thus would serve to increase the accuracy of the final model simulations.

By using a commercial fuel source instead of a single-component fuel-substitute the aim was to ensure the experimental set-up remains as faithful to real life conditions as possible. Questions over whether observed phenomena would be re-created in a real fuel, when using a single-component substitute, are eliminated, and greater credibility can be given to the results obtained. It can also be said that successful replication of the observed flow phenomena in a real fuel by the computer model, will increase confidence in the models’ ability to produce an optimised nozzle design for a real injector.

3.4 NOZZLE ASSEMBLY

3.4.1 Introduction

The optical nozzle assembly consisted of a sandwich of metal and optical plates, which formed the required nozzle geometry. Sapphire and borosilicate plates, that will be detailed later, provided optical access to the sac and nozzle-hole regions.
Metal plates were then used to provide support to the nozzle-hole plate and, as will be discussed, create the required geometry in the sac volume.

### 3.4.2 Nozzle Design

Three different nozzle geometries were initially designed to induce differing cavitating flow structures and thus develop the understanding of their formation and interaction, as well as fully test the capabilities of the computer model. The simplest of these geometries was an axisymmetric single-hole design, intended to enable the study of geometric cavitation development in the nozzle-holes. As previously discussed, this form of cavitation structure has been characterised in some detail through empirical studies and extensive computer modelling. Therefore, it served as a useful starting point experimentally, where expected phenomena could be observed and initial validation of the model could be provided, before more complex flow regimes were examined.

The other two geometries were designed to optimise the likelihood of inducing string cavitation in the sac volume. Sharing common sac geometry, the nozzle plates contained three holes arranged in a triangular pattern. The designs of these multi-hole plates differed only in the diameter of these holes; one set being 110 µm the other, 300 µm. Figure 3.6 shows the common hole-layout for both designs.

![Figure 3.6 Detail view of the multi-hole nozzle layout for 110 µm design. Units in mm](image)
To allow testing to progress up to realistic injection pressures, the plates were manufactured from sapphire and chemically-toughened borosilicate. The weaker mechanical properties of the borosilicate, compared to the sapphire, meant it was only able to withstand a maximum predicted fluid pressure in the region of 500 bar. This was significantly lower than the predicted achievable pressures with the sapphire (≈ 2000 bar), but still provided valuable optical data beyond the scope of previous, similar, studies.

Initially, manufacturing limitations prevented the 110 µm multi-hole nozzle plate from being manufactured out of sapphire; therefore the borosilicate was initially used for this design. As a result, both multi-hole designs were manufactured out of borosilicate, with the 300 µm design also being manufactured out sapphire. The 300 µm single-hole nozzle was only manufactured from sapphire.

The outer dimensions of the plates were specified by the plate design that had been used in a previous high-pressure injection test rig, enabling use of the old plates in the new rig if required. It was however, not possible to manufacture the 25 mm outer diameter (OD) of these plates from borosilicate; the extrusion method of their manufacture was limited to an OD of 10 mm. Accordingly, certain hardware features, namely the optical cassette and clamping mechanism, both discussed later, were designed to be interchangeable, allowing two plate diameters to be used. For simplicity, these features, and the plates themselves, will be discussed in generic terms, as the outer diameter of the plate is largely irrelevant to the primary function of the component.

The initial batch of sapphire plates had been manufactured with a 25 mm OD, but when new plates were required, the design was altered to conform to the 10 mm OD of the borosilicate plates. During testing the 10 mm OD had proved sufficient to enable easy handling and assembly of the plates, with the excess size of the 25 mm plate judged to be unnecessary.
As well as reducing the size of the sapphire plates to 10 mm OD, the design was changed for both materials, to include a polished flat on the forward facing edge of the plates.

Figure 3.7 shows the original plate design, which included parallel flat faces that were used for locating and aligning the nozzle assembly in its holder, which will be described further in Section 3.4.4.2. The face through which the nozzle flow was viewed remained curved however, and due to manufacturing limitations was not polished to an optical standard. To overcome this and improve the optical quality of the plates, an additional, optically polished flat was specified for this front face. Figure 3.8 shows the two designs, and the significant increase in image quality achieved by using the new design is illustrated in Figure 3.9, with flow features approximately 10 µm in size now visible. Further development of the imaging and illumination techniques will be discussed in the following chapter.
3.4 Nozzle Assembly

Figure 3.9 Comparison of image quality using a curved front surface (left) and a flat front surface (right). Note the slight decrease in apparent hole diameter caused by removal of the curved surface.

The manufacturer selected to produce the new plates, Agate Products Ltd., were also able to produce multi-hole plates of sapphire with hole diameters less than 200 µm. Being at the limit of their manufacturing capabilities, it was specified that the holes be drilled to the minimum diameter they could achieve with an optically polished inner surface. The resulting plates contained holes of 160 µm diameter which would allow an alternative test case to the 300 µm multi-hole to be investigated at high pressure (>500 bar).

3.4.3 Preliminary Design Simulations

To produce the flow characteristics required to develop a ‘string’ in the sac volume, preliminary simulations were run of a sac geometry based on that of a mini-sac type injector, where the string phenomena had previously been observed.

The model data (see Figure 3.10), provided by City University, firstly depicts the velocity stream-lines created when a single phase simulation of a typical mini-sac nozzle inlet geometry is carried out. A prominent vortical structure can be seen in the sac volume in close proximity to the nozzle entrance. This flow structure correlates closely with City’s experimental findings, where string cavitation is observed to form in a similar location.

To produce data over the full operating pressure range of a real injector required the geometry of the mini-sac injector to be simplified. Doing so enabled the cost effective manufacture of the optical parts from materials able to withstand the high pressures of a realistic injection event, and also simplified the application of optical diagnostics to the internal flow.
Therefore, an injector nozzle geometry designed specifically to create an intense vortical structure near the nozzle inlet was developed. Successful re-creation of the flow conditions in the simplified injector by the computer model would then allow more complex injector geometries to be simulated, with a high level of confidence that the results obtained were accurate over the full operating pressure range of the injector.

A number of design iterations were simulated to determine the geometry most likely to create the desired flow characteristics in the sac volume. To re-create the asymmetric nozzle inlet flow pattern seen in the mini-sac injector, the flow area above the nozzle inlet was restricted and fuel enters the sac volume ‘off-axis’. To achieve this in reality, a percentage of the cylindrical sac volume was filled by an obstruction. Figure 3.11 shows the CFD models of three design iterations. The fluid path of the nozzle is shown in grey. For obstacle 1, fuel entered the sac volume from the top left and was forced through $90^\circ$ to flow under the sac obstruction and exit through the nozzle-hole on the right. The flow in the sac volume above the right hand nozzle-hole closely resembled that of the mini-sac injector, and therefore string cavitation was expected to form in this area. In the region where the flow was forced through $90^\circ$, a re-circulating flow structure was set up, perpendicular to the nozzle axis, where it was expected that the intensity of the vortex created would be great enough to affect string cavitation in the two nozzles directly below this region.
Of the geometries simulated, a variation of obstacle 2, where 50% of the sac area was obstructed, showed the greatest tendency to form intense vortical flow structures. In this design, simulated in Figure 3.12, two holes were positioned under the sac obstruction. As can be seen from the simulation results, each of these nozzles had a vortex positioned near its inlet, as in the mini-sac simulations, suggesting a high probability that string cavitation could be affected in both of these nozzle-holes. In addition to these vertical structures, a lateral vortex was shown to form in the recirculation zone along the edge of the sac obstruction. City hypothesise in their labelling of the images that the presence of this vortex could potentially produce an additional string that feeds into both nozzle-holes.
3.4 Nozzle Assembly

By positioning two holes under the obstruction, the complexity of the flow structure in the near-nozzle region of the sac was predicted to be greater than that of the single vortex seen in the mini-sac injector. Simplification of the geometry had therefore led to an apparent increase in the complexity of the injector flow and thus, would provide a more robust case for model accuracy if the observations could be successfully recreated experimentally.

3.4.4 Optical Nozzle Construction

Construction of the optical nozzle assembly was achieved by manufacturing separate plates, which would then be clamped together to form the required geometry. For the case of the multi-hole set-up described above, the assembly consisted of an optical nozzle plate, an optical sac plate and a metal obstruction plate. For the single hole set-up the obstruction plate was omitted from the assembly. In each case a metal nozzle-support plate was positioned below the nozzle-hole plate. Schematics of the two set-ups are shown in Figure 3.13

![Figure 3.13 Cut-away views (ref: Figure 3.7) of the two nozzle assemblies](image)

3.4.4.1 Obstruction Plate

The obstruction plate, positioned above the sac plate in the multi-hole assembly, has a semi-circular protrusion on its underside, which slotted into the sac volume. Its height was such that it left a 200 µm gap between its bottom surface and the nozzle plate, creating the restriction above the two rear nozzle inlets. A ‘D’ shaped hole, formed the opposing semi-circle, which continued through the plate to create a ‘D’
shaped inlet at the top of the assembly. These features are illustrated in Figure 3.14. An angle of 4.5° on the flat edge of the semicircular profile steadily reduced the flow area as it entered the sac volume. This increased the percentage of the sac diameter that was occupied by the obstruction to the required 50% specified by the CFD simulations, whilst minimising the restriction to the fluid flow at the ‘inlet’ to the plate.

Figure 3.14 Views of the Obstruction Plate showing ‘D’-shaped hole entrance on the topside (left) and protrusion on underside (right)

Figure 3.15 illustrates how the nozzle assembly aimed to recreate the asymmetric hole-entrance flow that was required by the model, and had been shown to affect string cavitation in similar flow geometries. This flow is compared to the simulated hole-inlet flow for the mini-sac nozzle that was previously shown in Figure 3.10

Figure 3.15 Schematic of the fluid flow-path through the optical nozzle assembly with comparison to the nozzle-hole flow for a mini-sac type nozzle.
3.4 Nozzle Assembly

3.4.4.2 Optical Cassette

Correct alignment of the plates was essential if the correct geometry was to be achieved. For the axisymmetric case, lateral alignment was needed to maintain the concentricity of the sac volume and the nozzle. The asymmetric case had the additional requirement of rotational alignment to maintain the correct position of the obstruction over the nozzle-holes. To achieve this, the plates were housed in a metal cassette during testing. The flats, machined onto the sides of the plates, aligned with flat surfaces on the internal bore of the cassette to obtain the correct rotational alignment. The location fit of the plates in the cassette bore ensured their lateral alignment.

To provide optical access to the plates whilst they were in the cassette, a section of the main body was removed, enabling the central bore to be viewed from the front and the rear, as shown in Figure 3.17.
3.4 Nozzle Assembly

The swept angle of the viewing slot provided sufficient access and flexibility for both the imaging and illumination equipment that will be discussed in the following chapter. Upon clamping, the cassette also served to align the nozzle assembly with the fuel delivery outlet from the injector. A ‘U’-shaped bracket, fitted to the base of the injector housing, provided a sealing surface for the injector tip and had an outer protrusion which formed a location fit with the internal bore of the cassette. When located in the cassette, a 1.17 mm diameter hole through the centre of the protrusion aligned with the sac volume, or the ‘D’ shaped hole of the obstruction plate, delivering the injected fuel quantity from the injector to the nozzle assembly.

Due to the axisymmetric flow geometry of the multi-hole nozzle assembly, and in particular, the off-axis positioning of the ‘D’-shaped hole inlet, two variations of the ‘U’-shaped bracket were required. For the axisymmetric case, the outlet from the U-bracket is positioned in the centre of the protrusion, aligning it with the centre of the sac volume in the assembly. To achieve correct alignment with the ‘D’-hole, for the asymmetric case, the position of the outlet was offset in relation to the assembly. To accomplish this, the protrusion was offset by 0.39 mm towards the rear of the bracket, thus moving the injector outlet hole, which remained on-axis with the injector, directly over the ‘D’-hole. To maintain the alignment of the protrusion with the cassette bore, a cassette holder with an offset of 0.39 mm was also used for the asymmetric case.
3.5 Clamping Mechanism

Figure 3.19 Schematic of the 0.39 mm offset applied to the protrusion of the U-shaped bracket to maintain alignment when using the obstruction plate.

3.5 CLAMPING MECHANISM

The protrusion from the U-bracket served as the top jaw of the clamping mechanism (see Figure 3.20), necessary to seal the internal geometry of the assembly under high fluid pressure. The bottom jaw consisted of the cassette holder, mounted on the cassette bridge. When clamped, the nozzle assembly would be compressed into the cassette and maintained at a desired clamping load.

Figure 3.20 Cut-away view of the clamping assembly. Drawing courtesy of TDI Llc.
An essential feature of the clamping mechanism was its ability to increase the loading on the optical plates in an evenly distributed and controlled manner. Sudden application of a load could cause the mechanical failure of the plates and therefore had to be avoided.

The clamping mechanism, shown in Figure 3.21 formed the upper structure of the optical bench. The four guide pillars minimised the pitch and yaw of the top jaw as it was lowered. The clamping load was applied at the centre of the platform, directly above the nozzle assembly, by means of a lead screw, driven by a stepper motor connected to a high-ratio gearbox. The gearbox and the pitch of the screw limited the clamp to a maximum translational velocity of 1 mm.s⁻¹.
3.5 Clamping Mechanism

The screw located into a solid steel block which bridged two load cells shown in Figure 3.22. The measured load was fed to the rig control software and allowed a user defined load to be applied and maintained. The control logic for the clamping procedure was divided into two parts; the first part saw the clamp lowered to a position just above the nozzle assembly, the second involved the top jaw making contact with the nozzle assembly and applying the load. Each of these stages was set to operate at a percentage of the maximum clamping velocity. For the first stage, this was typically 100%, and < 5% for the second stage. The distance over which the clamp travelled for each stage was also user defined, with safety logic employed to prevent damaging the assembly by over-travel during the first stage; an increase in load during the first stage saw the clamping procedure halted and the clamp retracted.

The clamping loads required for the borosilicate and sapphire plates were determined through a FEA analysis carried out by Caterpillar Inc. Loading had to be sufficient to maintain a seal between the plates at high pressure, whilst remaining within the
mechanical limitations of the material properties. A clamping force of 90 kN was calculated to be necessary for the larger sapphire plates and thus set the upper design limit for the capability of the clamp mechanism.

3.6 NOZZLE EXIT CONDITIONS

3.6.1 Introduction

Downstream of the nozzle assembly, a liquid or gaseous nozzle-exit boundary phase could be created through the attachment of a fuel injection rate tube or an injection chamber, respectively. Both fittings located and sealed against the underside of the cassette bridge (see Figure 3.23) making them independent of the nozzle assembly above.

Both apparatus were capable of providing the nozzle with a downstream back-pressure of up to 150 bar (15 MPa) to simulate the near-TDC in-cylinder pressures of a highly turbocharged diesel engine. This allowed nozzle characterisation over a wide range of cavitation numbers, which is driven by the pressure drop across the
3.6 Nozzle Exit Conditions

nozzle, and in reality, allowed simulation of the injection event across the entire intake crank angle range.

### 3.6.2 Injection Rate Tube

#### 3.6.2.1 Introduction

The fuel injection rate tube provided quantitative nozzle mass flow-rate data to complement the optical data captured by the imaging system for each recorded injection event. This allowed the injected fuel quantity to be derived and thus the discharge coefficients of the nozzle to be measured.

The rate tube was capable of providing a measure of the injected fuel quantity for an individual injection event when used in the transient-flow rig-configuration. From this, the performance of the nozzle could be determined through the nozzle discharge coefficient. This measure of nozzle performance could also be used to provide an additional criterion for computer model validation.

During its use in this study, the sensitivity of the rate tube to certain fuel-flow conditions, and also the subsequent calculations based on its geometry, was highlighted. The methods employed to ensure these sensitivities did not adversely affect the accuracy of the calculated results will also be discussed in this section.

#### 3.6.2.2 Rate Tube Operating Principles

The Fuel Rate Indicator was developed by Wilhelm Bosch (1966), and allows the instantaneous injection rate to be obtained from the pressure wave created when a nozzle discharges into a fuel filled tube. Its operation is based on the continuity equation, which indicates that the flow volume per unit of time \( \frac{dQ}{dt} \) is determined from the velocity of the flow through a known flow area. The flow gives rise to a pressure wave, travelling at the fluid speed of sound, \( c \), that is proportional in magnitude to \( \frac{dQ}{dt} \), which is then measured by a piezo-capacitive pressure sensor. Since the volume per unit time \( \frac{dQ}{dt} \) is a linear function of the measured pressure, correct calibration of the pressure reading allows a quantitative measure of an injection event, through integration of the pressure trace over time.
By containing the pressure wave in a control area, Bosch was able to apply the hydraulic pulse theorem to obtain the calibration factor. The pressure-velocity equation, valid for a single pressure wave in a transient flow pattern, is derived from the pulse theorem

\[ P_f = c \rho_f u \]  

where \( P_f \) is pressure, \( c \) is the fluid speed of sound, \( \rho_f \) is the fluid density and \( u \) is the fluid velocity increase imparted by the advancing pressure wave, assuming static initial conditions.

Substituting Equation 3.1 into the equation for fuel quantity per unit time:

\[ \frac{dQ}{dt} = A_f u \]  

where \( A_f \) is the cross sectional area of the tube at point of measurement, gives:

\[ \frac{dQ}{dt} = \frac{A_f}{c \rho_f} P_f \]  

Multiplying by density, \( \rho \), gives the mass flow rate:

\[ \dot{m} = \frac{A_f}{c} P_f \]  

Integrating over the injection event gives the injected fuel mass:

\[ m = \frac{A_f}{c} \int_{SOI}^{EOI} P_f \, dt \]  

where \( m \) is injected mass, and SOI and EOI are start and end of injection respectively.

A typical pressure trace, resulting from a 2 ms injection event is shown in Figure 3.24. Integration of the graph between SOI and EOI in this raw state, with the axis values as they are, would give a proportional measure of the injection quantity with arbitrary units.
3.6 Nozzle Exit Conditions

To achieve a quantitative measure of the injection quantity, the ordinate is multiplied by the $A_f/c$ factor in Equation 3.4, converting it to units of mass flow rate, illustrated in Figure 3.25. As will be discussed, gaining an accurate measure of this factor is critical to achieving accurate quantitative data.

![Figure 3.24 Typical rate tube pressure trace for a 2 ms injection event](image1)

![Figure 3.25 Pressure trace converted to show mass flow rate vs time. The x-axis has also been converted from us to seconds to simplify integration](image2)

3.6.2.3 Physical Set-up

In its practical application, the nozzle discharged into a predetermined length of calibrated hydraulic tubing. The length of the tube was sufficient to provide the
3.6 Nozzle Exit Conditions

pressure wave with a double-transient time that exceeded that of the injection
duration, thus preventing the reflected wave from distorting the measured trace. For
the case of this study, the tube measured 6.06m (20 ft.) in length from the location of
the transducer with a 4 mm ID; more than adequate for the maximum 4 ms injection
duration permitted by the CR200 injector. The end of the tube was restricted by a
needle valve, which allowed a variable back pressure to be created within the tube.
For the analysis of nozzle cavitation, where the back pressure was critical to the
nozzle-flow conditions observed, the ability to adjust the back pressure was of great
importance.

The pressure transducer was placed close to the nozzle exit to prevent degradation of
the pressure wave due to surface friction and maximise the time for the reflected
wave to return to the measurement area. For this study, the transducer was located in
the steel block at the top of the rate tube which locates the device into the cassette
bridge (see Figure 3.23). This placed it 25 mm downstream of the cassette exit
orifice.

In addition, a piezo-resistive transducer, diametrically opposed to the Piezo-
capacitive transducer, provided an absolute pressure reading for determination of the
back pressure in the rate tube. A comparison of the pressure traces produced by each
type of transducer is shown in Figure 3.26

![Figure 3.26 Side-by-side comparison of the output from the piezo-capacitive (left) and piezo-resistive (right)
pressure transducers.](image-url)
3.6 Nozzle Exit Conditions

3.6.2.4 Calculation of $A_f/c$ Factor

As described in the previous section, conversion of the rate tube pressure data from a qualitative measure of injection quantity to a quantitative one requires multiplication of the pressure data by the $A_f/c$ factor. It therefore followed that precise knowledge of the flow area and the speed of sound were prerequisites for accurate results.

Calculation of $A_f$ at the point of measurement was complicated by the presence of the pressure transducers and their seating geometries. The nominal pipe ID of 4 mm was used for initial calculations, assuming that the brief change in flow area around the pressure transducers would have a limited affect on the overall magnitude of the pressure wave.

Calculation of the speed of sound ($c$) was carried out by measuring the time taken for the reflected wave to return to the transducer. An average value of 9.05 ms, taken over ten injection events into a back pressure of 20 bar was determined as the time for the pressure wave to travel the 12.12 m distance. This gave a value of $c = 1340$ m.s$^{-1}$. This result was consistent with values of $c$ found in literature for diesel type liquids (e.g. Hutala and Vilenius, 2001; Tat and Van Gerpen, 2003) who gave figures in the region of 1300-1400 m.s$^{-1}$ at similar temperatures and pressures. The increase in $c$ with increasing fluid pressure was highlighted in the work of Hutala and Vilenius, and although the pressure range in their study far exceeds those created in the rate tube, changes in $c$ were observed, with $c$ increasing to 1355 m.s$^{-1}$ at a back pressure of 100 bar.

To confirm that these values for $A_f$ and $c$ produced accurate mass flow rate figures, the theoretical calculations were backed up by experimental measurement of the fuel delivery quantities. This was achieved by closing the valve at the end of the rate tube, turning it into a fixed volume. Each injection into the rate tube thereafter would result in a fluid pressure rise within the rate tube that was directly proportional to the quantity of fuel injected. After a set number of injections, a separate valve temporarily fitted at the end of the tube was opened, relieving the pressure in the tube and ejecting a quantity of fuel that was proportional to the pressure drop seen when the valve was opened. This quantity of fuel was then weighed using electronic scales.
It was possible to calculate the mass of fuel per bar of pressure for the final pressure drop by logging the pressure trace from the piezo-resistive transducer for this entire procedure, as shown in Figure 3.27; this was then related back to the pressure rise seen for each injection. A value of fuel mass for each injection could then be calculated and correlated with the integrated area under the pressure wave trace as illustrated in Figure 3.28

A correlation factor of 6.6 mg/bar was found to be consistent across three runs of the procedure.

Calculation of the fuel mass per injection via this method was then compared with the results obtained through the theoretical calculation. By using a value of $A_f$ for a pipe ID of 4 mm, and taking $c$ to be 1350 m.s$^{-1}$ as an average of the variation in $c$ with pressure, the results of the two methods were in agreement to within <1% across all injections.

This degree of confidence allowed the theoretical calculation of injected fuel mass to be used, which required less post processing of the raw pressure data and allowed clearer presentation of the results.

As a point of note, it can be seen from the trace in Figure 3.28 that the pressure level either side of the initial pressure wave remains unchanged, only increasing as the injected fuel quantity enters the rate tube.
3.6 Nozzle Exit Conditions

3.6.2.5 Rate Tube Signal Anomalies

The sensitivity of the rate tube to certain flow conditions was also observed. The first of these was noted under low back pressure conditions, where the advancing pressure wave caused a resonating pressure fluctuation in the tube, which is shown in Figure 3.29.

Figure 3.28: Analysis of a single injection event into the fixed volume

Figure 3.29: Resonant pressure fluctuation created in the rate tube under low back pressure conditions
3.6 Nozzle Exit Conditions

The phenomenon occurred at back pressures less than 10 bar for all fuel injection pressures and clearly produced results that were not representative of the actual injection profile. Its frequency of approximately 5 kHz was independent of pressure wave magnitude and suggested that a resonance was being set up by the internal pipe profile in the measurement region.

This limited experimentation to back pressures greater than 10 bar, which still provided a large range of pressures that could be tested (up to 150 bar) but meant that injections into equivalent atmospheric pressure were not possible.

A further characteristic of the pressure trace, referred to as the “cumulative signal error” by Payri et al. (2008) was also observed during these experimental studies. Payri et al. describe this error as the result of an increase in the reference pressure $P_0$ during the injection, caused by the absorption of the fuel flow energy. It represents itself in the pressure trace as a steady increase in the pressure level during the “steady-state” portion of the injection event.

As a result, the area under the pressure trace is artificially increased, giving rise to over-estimation of the injected fuel quantity. Payri et al. set out a methodology based on numerical solutions to correct this error. They highlight that the injection pressure, measured in the fuel rail, actually falls during the injection, making it impossible for the injection rate to increase.

The results of this study showed similar characteristics under certain conditions. Figure 3.30 shows an example of an injection profile where this cumulative error effect is quite prominent. This particular profile was taken at a relatively low injection pressure of 300 bar, exiting into a back pressure of 150 bar. Under these conditions nozzle cavitation is suppressed and therefore the discharge coefficient of the nozzle should remain relatively high.

Further evidence of this would be provided by comparing the injection profile of the exiting flow to that of the flow entering the nozzle. For a high discharge coefficient to be maintained, the difference in the shape of the inlet and outlet profiles would have to be small; conducive to minimal losses in the nozzle flow.
Figure 3.30: Rate tube pressure trace showing cumulative error profile

Figure 3.31 shows the sac pressure trace for a $P_{\text{rail}} = 300$ bar injection event. It can clearly be seen that the overall shape of the profile closely matches that of the exiting profile in Figure 3.30. This suggests that the “cumulative error” seen in the rate tube trace is not the result of signal error; rather it is caused by similar fluctuations in the sac pressure, passing through the nozzle with minimal alteration due to the high discharge coefficient.

Under cavitating flow conditions however, the rate tube pressure trace “flattens off” and the cumulative effect observed is less significant as illustrated by Figure 3.32.
At higher cavitation numbers, the energy provided by the driving pressure is more likely converted to cavitation rather than additional mass flow through the nozzle, and thus a reduction in the nozzle discharge coefficient is observed under cavitating flow conditions. The flattening of the rate tube pressure trace, in relation to the sac trace, can be attributed to this effect as cavitation begins to choke the flow and the pressure rise in the sac profile is lost to cavitation processes in the nozzle hole.

Therefore, it could be deduced that the injection profile measured by the rate tube was representative of the nozzle flow, and required no correction to adjust the profile if it appeared to be erroneous.

3.6.3 Injection Chamber

The injection chamber provides greater flexibility to the flow rate that can be passed through the nozzle. The inherent flow restriction created by the rate tube prevents it from acting as a suitable nozzle-exit flow boundary for constant-flow experiments. To allow steady state nozzle flows to be studied the nitrogen injection chamber was fitted, allowing back pressure to be set independently from nozzle flow rate. Previous studies (Soteriou et al., 1995) have shown that no differences to the in-nozzle flow conditions are observed between injections into liquid and gas; the results from the current study, presented later, serve to confirm this. Consequently, changing from one apparatus to the other did not require a separate interpretation of the results. This
has also allowed the results of numerous preceding nozzle-flow studies to be directly comparable, whether they used liquid or gas as the downstream medium.

A number of additional design features were required to mitigate the effects of injecting high fuel flow-rates into the chamber; to maintain a gaseous boundary condition for the nozzle, injected fuel would need to be removed to prevent a build up of fluid in the chamber. Additionally, the injected fuel would reduce the gaseous volume of the chamber and result in an increase in gas pressure, and hence nozzle back pressure. To overcome this, the chamber pressure is regulated by control valves at the gas inlet and outlet, and monitored by the rig safety logic.

Figure 3.33 Schematic of the injection chamber and associated controls. Adapted from drawing courtesy of TDI Llc.
3.6 Nozzle Exit Conditions

Figure 3.33 provides a schematic layout of the injection chamber and its control hardware. The main outlet pipe was connected to the same back pressure valve as the rate tube, allowing the flow rate of the fuel returning to the main fuel tank to be regulated. This, combined with a regulator on the outlet of the gas bottle, which was set to the desired gaseous back pressure, controlled the pressure within the chamber. The gas inlet, located at the top of the chamber, was controlled by a solenoid valve, which opened during testing to allow pressurised nitrogen into the chamber.

To prevent nitrogen from passing through the chamber and entering the fuel tank return pipe, a quantity of fuel was first injected into the chamber before a test began. The fuel quantity was initially set by injecting fuel until it was seen to reach the level of the fill valve, which remained opened during this filling phase. Once this level was reached, the fill valve was closed allowing the chamber to be pressurised. To ensure the fuel level in the chamber remained steady, the back pressure valve had to be manually adjusted so that the chamber outflow, at a given chamber pressure, matched the incoming injector flow. This required the back pressure valve to remain closed during pressurisation, only being opened once the fuel injection had begun. During testing, and when a change to the back pressure was required, the simultaneous adjustment of both the gas regulator and the back pressure valve was necessary until a stable pressure was reached within the injection chamber.

At the end of a test the dump solenoid was activated, venting the chamber pressure into atmosphere. The vent tank served to collect fuel vapour and aerosol escaping with the vented gas, which could then be reintroduced to the rig fuel supply when the tank became sufficiently full.

Two further outlet pipes were also connected to the chamber. The chamber drain pump, allowed fuel within the chamber to be removed in the event that the fuel level became too high, or to completely drain the tank before it was removed from the rig. The safety release valve, located on the remaining chamber outlet, was set to activate if the chamber pressure reached 210 bar. This initiated a rig shut-down that would
simultaneously purge the gas pressure to the vent tank and allow fuel to enter the fuel tank return pipe without passing though the back pressure valve.

When used for transient work, the small fuel quantities being injected had little effect on the pressure within the chamber; therefore, the back pressure valve could remain closed during the entire test procedure.

### 3.7 SAC PRESSURE MEASUREMENTS

One limitation of the nozzle assembly and cassette design was that it prevented direct measurement of the sac pressure, which, if the boundary conditions transferred to the model were to be accurate, was an essential requirement. It was therefore necessary to manufacture a ‘dummy’ assembly that contained a sac volume and outlet with the same dimensions as the optical nozzle, which allowed a pressure transducer access to the sac volume.

The dimensions of the Kistler 4067BC2000A2 pressure transducer prevented it from being located in the wall of the sac, the sac only being 2 mm high. Instead the pressure transducer formed the underside of the sac volume and the nozzle outlet was moved through 90° to exit horizontally from the side of the sac. The nozzle outlet was created by machining a ‘v-groove’ along the top surface of the new sac plate, which was then sealed by the protrusion on the u-shaped bracket when the fixture was clamped up. The cross-sectional area of the v-groove was designed to match that of the nozzles being used, requiring three different dummy assemblies to be manufactured.

Figure 3.34 Cut-away view of the dummy assembly for sac pressure measurement
The pressure transducer and connecting lead were accommodated in a supporting cylinder (Figure 3.36), which was placed below the dummy assembly and located in the cassette holder, aligning the sac volume with outlet from the U-bracket.

Surface profiles were taken across the v-grooves to determine their exact cross sectional area. The relationship between the nozzle outlet area and the peak sac
3.7 Sac Pressure Measurements

pressure could then be used to determine the sac pressures for the optical nozzle assembly.

![Figure 3.37 0.2 mm v-groove, area = 0.09 mm²](image)

![Figure 3.38 0.4 mm v-groove, area = 0.2 mm²](image)

![Figure 3.39 0.6 mm v-groove, area = 0.27 mm²](image)

Due to the shape of the cutting tool, the areas of the grooves varied from the design requirement and machining limitations prevented sac pressure measurements with an outlet area equivalent to the 110 µm multi-hole nozzle; the smallest of the three
designs. However, the other two outlet areas closely matched those of the 300 µm multi-hole and single hole designs and so could be used to extrapolate the pressure for the smaller nozzle. A certain degree of error could be catered for due to the greater sensitivity of the cavitation number to back pressure. This was a parameter that could be accurately measured and therefore minor discrepancies in the sac pressure would not have significant impact on the final results.

Initial repeatability tests showed very little variation (<1%) in the sac pressure profile between injections. This allowed a single injection to be used for analysis of a certain test condition, without the need for averaging over a number of injections.

![Figure 3.40 Sac pressure traces showing repeatability of injection event. P_{rail} = 400 bar](image)

Rail pressure was monitored by the ECM via a pressure transducer positioned approximately 400 mm upstream of the nozzle assembly. Pressure traces taken using the dummy assembly revealed a significant pressure drop across the CR200 injector, resulting in lower sac pressures than the desired rail pressure, P_{rail}, input to the ECM. A pressure drop offset was therefore applied for both experimental calculations and accurate boundary condition representation for the computer model.
As expected, the larger cross-sectional area of the 300 µm multi-hole nozzle exit saw the most significant pressure drop of the three test cases. The graph below shows the pressure trace for a 2 ms injection event with a desired rail pressure of $P_{\text{rail}} = 400$ bar.

It is immediately obvious that the sac pressure achieved remained below the desired level, with the maximum pressure level also falling short, peaking at 350 bar just before the end of injection. Therefore, at no point during the injection was the desired pressure level actually reached in the sac volume. For both the experimental calculations and the steady state boundary condition input for the computer model, a single value of pressure upstream of the nozzle, $P_1$, is required. The transient nature of the injection pressure profile prevents an accurate single value from being obtained, and therefore, an average value of pressure is taken across the main duration of the injection. For the $P_{\text{rail}} = 400$ bar case above, the average value of the sac pressure was taken to be 300 bar.

A similar study was carried out using an equivalent outlet area to the 300 µm single-hole nozzle. Again, a pressure drop was observed, although as expected, this was not as severe as the multi-hole case. The following figures show how peak sac pressure
fell below the desired level for the lower rail pressures, but reached it in the case of the higher pressures. In all cases the majority of the pseudo steady-state region of the injection remained below the desired pressure.

The apparent skew of the pressure traces suggested a progressive pressure build up in the sac and therefore it was thought that an increase in injection duration may result in the peak pressure being maintained for a longer period of time. Figure 3.44 shows
the results for injection durations of 3 ms and 4 ms at $P_{\text{rail}} = 300$ bar and $P_{\text{rail}} = 600$ bar.

The pressure peak that occurred around 1.8 ms into the injection remains; beyond this the pressure returned to a level just below the desired pressure. This ruled out the occurrence of a pressure build-up and was more likely a characteristic of the injector. By returning to a lower pressure level after this peak, it was shown that extending the injection duration did little to increase the average sac pressure to a value that was closer to the desired rail pressure.

A further study was carried out to assess the sensitivity of the pressure trace to the sac volume. With the obstruction plate in place, the volume of the sac in the multi-hole testing was significantly reduced. It was therefore important to confirm that a reduction in the volume would not have an affect on the sac pressure. Accordingly,
the obstruction plate was added to the dummy assembly, replicating the sac volume used in the optical assembly. When compared to the un-obstructed sac pressure trace, as shown in Figure 3.45, there is no significant variation in the trace profile between the two cases

![Sac Pressure, bar](image)

*Figure 3.45 Comparison of sac pressure traces for an obstructed and un-obstructed sac volume*

It can therefore be concluded that alterations to the sac volume do not affect either the profile of the pressure trace or the pressure levels that are reached during an injection event.

### 3.8 CONCLUDING REMARKS

This chapter has presented the design and development of the high pressure optical diesel injection rig along with descriptions of the key hardware components. The following chapter will focus on the hardware and techniques that were developed to enable the capture of optical and pressure data from the nozzle assembly.
4.1 INTRODUCTION

In the previous chapter, the apparatus required to create the experimental conditions were discussed. This chapter will now discuss the techniques and apparatus used to capture the optical data from the optical nozzle assembly.

The acquisition of optical data was a fundamental design requirement of the test rig, and as such was considered a priority at each design stage that affected it. However, work remained to design and develop the illumination and image capture techniques, external to the rig design, that would deliver high quality optical data.

Imaging in the form of high speed video and high-resolution digital SLR single-shot photography were used to capture images of the nozzle flow under both transient and steady-state flow conditions. Illumination was provided by an Oxford Lasers LS-20-50 Copper Vapour laser, capable of producing a pulse repetition rate of 50 kHz with a pulse duration in the order of 10-30 ns. With the maximum achievable pressure drop across the nozzle creating flow velocities in the region of 500 m.s\(^{-1}\), the 10-30 ns exposure time equates to a flow displacement of approximately 10 µm. This was sufficient to essentially freeze the fluid motion in the nozzle holes, equating to sub-pixel displacement at the magnification levels employed to view the sac volume and nozzle holes in a single frame.

Although primarily used to illuminate the high speed video imaging, the adaptations required to use the copper vapour laser for single shot illumination will also be described in this section. The addition of a Nd:YAG laser to the experimental set-up as part of the particle image velocimetry (PIV) study (covered in Chapter 6), also had the benefit of vastly simplifying the single-shot image capture. Here, the use of the Nd:YAG laser will be described in terms of this application, with the details of the set-up for the PIV study being described in Chapter 6.
For high speed imaging, the copper vapour laser was synchronised to a Photron APX-RS high speed camera to produce a single exposure per frame at frame rates up to 42,000 frames per second (fps). For single shot imaging a 7.5 mega pixel (MP) Olympus E-330 digital SLR camera was synchronised to a single laser pulse; also producing an exposure-time independent of shutter speed.

The details of each camera set-up and the development of their corresponding methods of illumination will now be examined.

4.2 LENS SYSTEM

A Nikon Micro Nikkor 200 mm f/4 D lens, combined with a bellows and extension rings, formed the main lens system shown in Figure 4.1. This system was developed for both high-speed imaging and single shot photography, with further extension rings being added to provide similar levels of magnification for the larger, 7.5 MP chip of the SLR camera.

This approach provided the necessary magnification, whilst maintaining a sufficient working distance from the test piece of approximately 300 mm; minimising the risk of lens damage in the event of a nozzle failure.

The size of the flow region to be imaged was constrained by the nozzle length, which was to be imaged in full. Above this, the near-hole region of the sac volume was also included to allow the flow conditions near the hole-inlets to be observed. This resulted in an imaged flow region between 2-3 mm in height.

The entire lens system and camera sat on a lead-screw-driven traverse, which in combination with the bellows, allowed fine adjustment of both the image focus and magnification. Figure 4.2 shows the lens body fixed to a 2-axis micrometer stage, which provided fine adjustment of the image position. This 3-axis set-up gave complete flexibility to the image capture
4.2 Lens System

Figure 4.1 Lens system shown with Digital SLR camera

Figure 4.2 Nikon lens mounted on 2-axis micrometer stage. This in turn is mounted on the traverse providing 3 axis of adjustment.
The magnification requirements of the system increased as testing progressed to look at ever smaller flow regions. The addition of extra extension rings, as a means of increasing magnification, eventually became impractical due to the length of the lens system and the light levels that could be achieved, and so changes to the lens itself were required. The Nikon macro lens was replaced with a standard Olympus 14-45 mm lens, reverse mounted to the bellows. By reverse mounting the lens, the optics behaved as a macro lens, producing high levels of magnification in a more compact system that now only required the use of two or three extension rings.

The compact nature of the system and the reduced working distance of the lens made this set-up unsuitable for the high-speed camera, whose body was too large to fit between the platforms of the optical bench at the reduced working distance. Instead the original system of extension rings was fitted with a telephoto doubler, which provided a satisfactory increase in magnification with the light levels available.

4.3 CAMERA HARDWARE

High speed imaging was carried out using a Photron FASTCAM APX-RS High speed camera, shown in Figure 4.3. Its 1024x1024 pixel CMOS chip was capable of full frame imaging at 3,000 fps, however the camera was most frequently used at a reduced image resolution of 512x512 pixels. This allowed it to operate at speeds up to 10,000 fps, driving the laser at its optimum frequency of 10 kHz when synchronised to the camera frame rate. As the results discussion will highlight in later chapters, this frame rate was still not sufficient to capture smooth frame-to-frame image transitions in the cavitating nozzle, but did allow the entire flow region to be imaged in a single frame at the magnification employed. For early results analysis this was of greater importance, with higher frame rates, and the reduced image resolutions they were restricted to, being employed in later work to investigate more specific, small-scale, phenomena. Operating at 10,000 fps also had the advantage, as mentioned, of running the laser at its optimum frequency. This meant its output power did not degrade during testing, thus providing the highest levels of illumination at all times.
For high-resolution single shot imaging an Olympus E-330 Digital SLR camera was used. This was selected primarily for its “live view” feature, which displayed a continuous live preview of the camera image on its rear LCD screen. This provided a safe means of framing images without the need to look through the viewfinder, which would otherwise directly expose the user’s eye to the high power laser illumination; at the time of purchase, this camera was the only model available with this particular feature. Its 7.5 megapixel (effective pixels) NMOS sensor offered a significant increase in image resolution over the 1 megapixel (max) chip of the high-speed camera and would therefore provide more detailed images of the cavitating flow structures.

4.4 IMAGE CAPTURE AND SYNCHRONISATION

The complexity of the image capture varied depending on the mode of injection and the image capture method being employed. The synchronisation of the capture to a single injection event presented different challenges for the high speed video and the single shot imaging.

To aid the synchronisation of external equipment to the transient injections, the “injector fire” signal from a second injector was taken from the ECM and converted
to a TTL pulse. The signal from this second injector would normally have been associated with a different engine cylinder, and as a result, a constant “crank angle” offset between the firing of the rig injector and the generation of the TTL pulse was produced. A time offset that was dependent on engine speed was therefore created, and had to be accounted for when setting the trigger timing.

Additionally, under transient conditions the injector was fired repeatedly, at a rate dependent on the simulated engine speed. When the user was ready to capture data it was necessary to ‘arm’ the trigger, with the subsequent TTL pulse triggering the data capture of the corresponding injection event.

As the experimental study proceeded, steps were taken to simplify the data capture triggering. This was achieved alongside the ongoing experimental work and thus, the early stages of testing saw the development of tools and techniques that allowed data to be captured using the rig features described above. These will now be discussed and their evolution to the techniques employed toward the end of the study will be described.

### 4.4.1 LabVIEW and Data-Logging Hardware

A LabVIEW software program was primarily written to log the rate tube pressure data. However, the desire to synchronise this data capture with the triggering of the image acquisition lead to a merging of the two tasks into a single program. The resulting code, in combination with National Instruments (NI) hardware, was able to log the rate tube pressure data and trigger the image capture of a single injection. A NI PCI-6110 data acquisition (DAQ) card and a NI BNC-2120 I/O connector block allowed data capture rates up to 5 Mega samples per second per channel (MS.s\(^{-1}.ch^{-1}\)) on 4 channels simultaneously, and provided all the necessary input and output connections to both capture data and handle multiple trigger signals.

Two channels of pressure data from the rate tube - absolute and gauge - were logged at 1 MS.s\(^{-1}\) for 5000 samples. This resulted in a real-time logging of 5 ms of data from each channel. The time delays in the LabVIEW code were set to begin logging
4.4 Image Capture and Synchronisation

Data 1 ms before the 2 ms injection event. The additional 2 ms after the expected end of injection ensured that the complete pressure trace of the injection event was captured.

4.4.2 LabVIEW for Synchronised High-speed Video

A signal generator was used to provide the external user trigger-input. The “signal-high” generated by the pulse satisfied an IF statement, which prompted the code to advance, and use the next Injector TTL signal to initiate the data capture. At the same time that this was initiated, the software generated a TTL pulse of its own to trigger the image acquisition. A delay was set after the Injector TTL pulse was received, to coincide the initiation of the data capture and image acquisition with the firing of the rig injector. The length of this delay was determined by observing the Injector TTL pulse and the pressure-trace output from the rate tube on an oscilloscope. The time difference between the rising edge of the TTL pulse and the start of the pressure rise in the rate tube was measured as 33.8 ms, for a desired engine speed of 1200 rpm. This engine speed setting was used for the majority of the early testing and meant that the injector fired at a rate of approximately 10 Hz. It was realised, however, that through exposure to numerous fuel injection events that were not recorded, undue stress was placed on the optical components, reducing their effective lifetime. Before the “fire-on-demand” system described in Section 3.3.2 was set up, this problem was mitigated by reducing the simulated engine speed to 120 rpm. Due to an apparent non-linear input/output speed relationship below the production idle-speed of 800 rpm, the ECM output an injector firing rate equivalent to 190 rpm; this appeared to be close to a minimum output speed, and was considered sufficient for the purpose of reducing component fatigue.

The resulting delay between the Injector TTL and the rig injector firing was measured as 210 ms for a desired engine speed of 120 rpm. as shown in Figure 4.5. This engine speed and corresponding delay were used for the remainder of the testing program.
4.4 Image Capture and Synchronisation

Figure 4.4 below, describes the operation of the code in a basic flow-chart

Figure 4.4 Flow chart overview of code processes to initiate pressure data logging and Image capture
Figure 4.5 Oscilloscope screen image showing time difference $t_{\text{TTL-SOI}}$ between the Injector TTL pulse (top) and the rate tube pressure trace (bottom), measured as 210 ms at 120 rpm desired engine speed.

To aid the user in setting the nozzle back pressure, a pressure gauge had been fitted to the main fuel return pipe, which was common to both the transient and steady state injection set-ups. Initially the scale on the analogue gauge display was found to have insufficient resolution to accurately set the desired back pressure intervals during testing. This was overcome by replacing the analogue gauge with a digital readout, which displayed the pressure with a resolution of 0.1 bar. Greater convenience and functionality was obtained, however, when the piezo-resistive pressure transducer was fitted to the rate tube. This allowed the absolute value of back pressure to be read into the LabVIEW code, where it could be displayed and logged. The graphical front-end of the code, shown in Figure 4.6, thus acted as a display for the back pressure, and the piezo-resistive pressure channel was logged simultaneously with the piezo-capacitive channel.

Figure 4.6 shows the difference in the outputs from the two transducers when a constant back pressure was present in the rate tube. The piezo-capacitive read-out, reacting to pressure changes only, shows a zero value on the graph, while the numerical display relays the value of the absolute pressure from the piezo resistive transducer. This was advantageous when it came to the results analysis, as a record of the actual back pressure in the rate tube at the time of injection was logged during the fuel injection event.
Once the desired number of data samples had been logged the code prompted a file-save dialog box. The recorded channels were saved as comma separated values (*.csv) files, and the code reset to await the next test point.

This basic control program thus allowed the simultaneous capture of pressure data and high-speed video, where the results obtained from one method could be directly compared to those of the other for any given fuel injection event.

**4.4.3 LabVIEW for Single-Shot Image Capture**

The synchronised capture of data using the high-speed camera was greatly simplified by the fact that the camera frame-rate drove the pulse frequency of the laser. This meant that the camera and laser could be considered as a single entity, which only required a single trigger signal to begin capturing optical data.
In the case of single shot imaging, the SLR camera and the laser operated independently during image capture. Therefore this required the triggering of the laser to also be integrated into the LabVIEW code; synchronising it to the camera and the pressure data capture.

Application of the single-shot technique was further complicated by the restricted operating characteristics of the laser. The copper vapour laser produced a beam of constant pulse frequency at wavelengths of 511 and 578 nm, with the repeated high-voltage discharges in the laser cavity increasing, and then maintaining, the laser power to usable levels. Upon pumping the vapour medium with an initial high-voltage pulse, up to ten further pulses were required for the energy of the emitted light to reach satisfactory levels. Further to this, the vapour medium required pumping at a sufficient frequency (ideally the optimum frequency of the capacitor set employed; in this case 10 kHz) to affect a step-wise increase in the energy of each successive pulse.

As a result, production of a single pulse on demand was not possible, and it was necessary to engineer a way of avoiding multiple image exposures through the need to constantly pulse the laser.

Greater control of the laser was achieved by integrating the manufacturer’s optional laser control unit into the system. The “N” Shot Controller provided the ability to “burst” the laser, producing a specified number of laser pulses, at a specified frequency, on demand. The timing diagram shown in Figure 4.7 provides an overview of its operation in “external trigger mode”. The control unit provided the laser with a constant run-frequency signal, \( f_r \), by means of its “laser drive” output. Upon receiving an external trigger, this frequency, and the laser output, was set to zero. The unit then output a “ready to fire” TTL pulse and remained idle until a further external signal was received. Upon receiving the signal, the laser drive output delivered a burst of ‘n’ pulses to the laser at a burst frequency, \( f_b \). The laser was then returned to its zero state, before reverting back to the run-frequency after a user defined period of time.
Integration with the camera required the ‘n’ pulse to be delivered when the camera shutter was open, with these two events then needing to be synchronised to the fuel injection event.

This was achieved by using the flash signal from the camera’s ‘hot shoe’ as the initial trigger. The “laser fire” signal would then be generated to coincide with the firing of the injector by the LabVIEW code. As the time between the initial trigger and the firing of the injector was variable, the camera was set to a long exposure that could capture the injection whenever it occurred.

To prevent over-exposure of the frame, a second, normally-closed shutter was placed between the camera and the optical nozzle assembly. The opening of this shutter was triggered to coincide with the burst of the ‘n’ laser pulses, in such a way that \( n - 1 \) of the pulses were blocked by the shutter; only exposing the camera to the \( n \)th pulse.

To achieve this, a shutter capable of opening within the space of two laser pulses, \( \Delta t \), was required. \( \Delta t \) was increased by reducing \( f_b \); relaxing the time constraint placed on the shutter. \( f_b \) was limited to 2 kHz by the N Shot Controller, but to reduce this further still a frequency divider was placed on its laser drive output. This reduced the output frequency by a factor of four so \( f_b \) became 500 Hz, increasing \( \Delta t \) to 2 ms.
$f_r$ was set to 40 kHz on the N Shot Controller, maintaining the laser drive frequency of 10 kHz at all other times.

The timing diagram shown in Figure 4.8 illustrates the sequence of events that takes place to synchronise the camera, laser, external shutter and fuel injection event.

Opening the camera shutter (1) generated the ‘hot shoe’ flash pulse (2), which was amplified to 5 volts and acted as the ‘trigger in’ signal for the N Shot Controller; setting the laser output (6) to zero and generating the ‘ready to fire’ pulse (3). The ‘ready to fire’ pulse then prompted the LabVIEW code to trigger from the next Injector TTL pulse (4) it received. To ensure that the refresh rate of the LabVIEW code picked up the ‘ready to fire’ signal every time, the pulse length was increased by a signal generator from 1 ms to 20 ms.

The code was adjusted to produce two output pulses. The first activated the ‘laser fire’ input (5) on the N Shot Controller, initiating the burst of laser pulses from the laser drive output (6). The second activated the external shutter (7), coinciding its opening with the final pulse of the laser burst. A delay, $t_s$, was placed between the two pulses to account for the activation time of the shutter, $t_a$, and the duration of the laser burst $t_b$, such that the shutter opening (8) took place between the final two pulses of the burst sequence.
A photo-detector was used to confirm the effectiveness of the set up, with the oscilloscope images in Figure 4.9 showing the external shutter blocking the \( n - 1 \) pulses of the laser burst.

![Figure 4.9 Oscilloscope images showing laser fire (top) laser drive (middle) and photo-detector (bottom) channels, without shutter (left) and with shutter (right). A single pulse is detected when the shutter is used.](image-url)
To enable different stages of the injection to be imaged, a separate user input allowed the initial delay $t_f$ between the injector TTL pulse and generation of the ‘laser fire’ pulse to be adjusted in microsecond increments. This delay was independent of $t_s$ and thus moved the end of the laser burst and the activation of the shutter to the desired point during the injection. Examples of this are shown in Figure 4.10 below, which shows the shutter activation in blue (low = open), the laser pulses in orange and the injection pressure trace from the rate tube in purple. The left hand case would expose an image 1 ms after SOI and the right hand image would do likewise at the end of injection.

Figure 4.10 oscilloscope images showing the final laser pulse of the burst being timed to coincide with different stages of the fuel injection event. Middle of injection (left) and end of injection (right). Shutter opening is also shown to show its synchronization with the final pulse.

4.4.3.1 External Shutter Design

It was critical to achieving single exposure images that a fast acting shutter was employed to open within the space of two laser pulses. Frequency dividing the ‘laser drive’ signal had increased this time constraint from 500 µs to 2 ms. The initial design utilised the high speed actuator from a standard hard drive, and is shown in Figure 4.11. The hard drive spindle and platters were removed and body was cut away to leave a single actuator arm exposed. To the end of this arm an arc of shim steel was attached, such that it was the same height as the optical nozzle viewing region. As the actuator arm rotated, the arc continued to block the view of the nozzle until just before the extent of the arm’s motion was reached. This allowed for the initial acceleration of the arm and meant that the trailing edge of the arc exposed the nozzle in the shortest time possible.
The activation pulse generated by the LabVIEW code was amplified to 12 volts to increase the energy imparted to the actuator magnet – increasing the rotational velocity of the actuator arm. The remaining hard drive body was attached to a base plate, which accurately located the shutter in front of the cassette bridge allowing for easily repeatable placement.

This initial design was later superseded by an off-the-shelf ThorLabs SH05 hi-speed shutter and control unit, which provided a more robust construction and greater functionality; the shutter activation signal seen in Figure 4.10 was output by the shutter unit. Only minor adjustment to the code was required to take account of the different (shorter) activation time of the new shutter mechanism.

Towards the end of the study, a separate investigation into the application of PIV to the nozzle flow required the use of a Nd:YAG laser. This laser was capable of producing a single high energy (100 mJ) laser pulse of approximately 8 ns duration, removing the need for an external shutter. Adaptation of the PIV Illumination set up saw this laser source pumping a fluorescein dye cell, placed behind the nozzle assembly, as shown in Figure 4.11.
Figure 4.12, providing uniform back lighting to the optical components. Its ability to produce a single laser pulse on demand, coupled with the new fire-on-demand control of the rig injector, negated the need for additional (LabVIEW) software to synchronise the hardware components. Instead, the ‘hot shoe’ pulse was sent to the signal generator controlling the relay-switch between the speed timing box and the ECM, thus initiating a single injection. The laser was then triggered a time, $t_{\text{TTL-SOI}}$, after the ensuing Injector TTL pulse was produced. As before, the camera shutter was held open for the duration of the sequence as the exact timing of the injection remained ambiguous. Similarly, the location of the image exposure during the injection event was now easily controlled by the delay placed on the triggering of the laser pulse.

A final advance came by using a ‘current probe’, clamped around the positive input wire connected to the rig injector solenoid. This produced a voltage output proportional to the current that fired the injector, and was used to trigger the image capture for both imaging techniques without the need to apply a delay.

The N Shot Controller was however, primarily designed to enable high-speed video capture at frame rates that exceeded the optimum frequency of the laser; bursting the laser at the same frequency as the higher frame rate. In practice, it was only possible
to burst the laser at frequencies up to 42 kHz to stay within the safe operating limits of the laser; synchronizing the burst (‘laser fire’ pulse) and the start of image capture with the fuel injection event. At these higher frequencies the laser was only capable of running at useable power levels for a finite number of pulses, so the laser was burst to the high frequency for a period consisting of tens of pulses, before returning to its optimum frequency. Physically synchronising the laser to the camera frame rate was greatly complicated due to the N Shot Controller needing to drive the laser frequency. As a result the camera frame rate was set to equal the burst frequency and ran independently of the laser. The short duration of the laser pulse (10-30 ns) in relation to the camera frame exposure time (1/frame rate) meant any frequency difference that may cause the two to drift out of phase, resulting in unexposed frames, would only occur briefly and not frequently.

In summary, the development of the associated hardware and software that allowed the optical and pressure data to be successfully recorded was driven by the need to overcome the operating characteristics of the available hardware. Less complex solutions were employed as they became available or a specific problem was better understood. This resulted in a continual development of the experimental techniques employed, whilst allowing useful data to be captured throughout the life of the project.

**4.5 ILLUMINATION**

Until now we have discussed the development of the methods used to capture images of the nozzle flow, but to achieve the highest quality optical data the means of illuminating the optical nozzle required similar attention. The microscopic flow features of the two phase flow and the small geometry of the nozzle, coupled with the refractive index differences between the optical materials and the working fluid, presented a significant challenge to creating high-quality lighting conditions. Illumination optimisation was ongoing throughout the experimental study and this will be reflected in the progressive quality of the images shown in the results discussion chapters that follow.
4.5 Illumination

4.5.1 Single Fibre Illumination

Initial work used a fibre delivered continuous white light source, which served as a basic illumination method for familiarising the author with the operation of the rig. It became apparent from this early work that the use of a diffuser would be necessary when using the laser, to eliminate the glare generated by the internal surfaces of the optical assembly. As such, acrylic diffusers with differing grades of opacity were shaped to fit around the optical cassette, and along with the fibre-optic placement, provided the first means of optimising the laser illumination quality.

The high refractive index of the sapphire \((n = 1.77\) (Irving et al. 1958)) and the small diameters of the nozzle geometry combined to produce a lensing effect that partially obscured the internal flow of the nozzle. As can be seen in Figure 4.13, low levels of diffusion resulted in roughly two-thirds of the hole being obscured, with only the centre-line of the nozzle, where the light passed through at small angles of incidence, being visible; Increasing the amount of diffusion i.e. increasing the opacity of the diffuser, brought about a reduction in this effect, increasing the viewable flow area. Doing so, however, had the counter-effect of reducing the definition of the visible flow features and so a compromise was reached to limit both effects to acceptable levels.

![Figure 4.13 Comparison of low diffusion (left) and high diffusion (right) lighting regimes for imaging the single-hole sapphire nozzle](image)

Increasing the amount of diffusion inherently reduced the amount of light that could be captured by the camera. To reduce the effect of this, the delivery optics were adapted to produce a horizontal light sheet, which was sufficiently thick to illuminate the optical section of the nozzle assembly.
By doing so, light was concentrated in the desired (vertical) region of the diffuser, while the broad projection of light onto the diffuser’s curved surface in the horizontal plane served to reduced the unidirectional nature of the light source; thus reducing the amount of required diffusion. This effect is illustrated in Figure 4.14. We will discuss shortly how this effect was of limited benefit in the case of the sapphire components.

This optical set up was fixed into a housing, which met the laser safety requirements by enclosing the optics and accurately located them behind the nozzle assembly, producing repeatable lighting conditions between tests.

4.5.2 Multi-Fibre Illumination

Greater flexibility was given to the illumination set up with the acquisition of a ‘laser energy-share’. This apparatus enabled the single 25 mm diameter copper vapour laser beam to be split down three optical fibres, with the light level from each being independently adjustable. This allowed the laser light to be delivered from multiple angles and thus provided greater opportunity for optimising the lighting conditions.

The three fibres were held in post holders clamped to an optical breadboard base that located the fibres behind the nozzle assembly as shown in Figure 4.15. Translating
4.5 Illumination

posts, provided fine height adjustment, and the clamps allowed free location of the post holders to position the fibres where necessary behind the assembly.

![Image of the multi-fibre illumination setup behind the cassette directing light onto the diffuser](image.png)

Figure 4.15 Image of the multi-fibre illumination setup behind the cassette directing light onto the diffuser (laser is off in image) using three fibres.

The higher refractive index of the sapphire required the light to be more diffuse than that for the borosilicate components. As a result, satisfactory lighting conditions were far harder to achieve for imaging sapphire nozzles and the benefits of using the multi-fibre set-up were not as great as they were for the borosilicate. Directing light at off-(camera) axis angles had no perceivable benefit as the light was refracted away from the camera. More appreciable results were gained when imaging borosilicate, where the refractive index of the material \(n = 1.51\) (Tearney et al. 1995) more closely matched that of the diesel \(n = 1.46\) (Dombrovskya et al. 2002), and multi-fibre illumination was able to produce high quality lighting conditions.

This resulted in two different lighting regimes being used: Imaging the borosilicate nozzle assemblies used all three fibres to provide illumination at multiple angles. For sapphire imaging, only the fibre placed on axis with the camera was utilised and the energy share was adjusted to direct 100% of the laser light down this fibre. A laser service had increased the laser output, so the use of a light sheet to provide sufficient
brightness was found to be unnecessary. This meant that the energy share could cater for both cases with minimal positional adjustment of the fibres.

4.5.3 Nd:YAG laser for Single Shot Illumination

A new illumination technique was introduced with the use of the Nd:YAG laser for single-shot imaging. Its higher energy, in the region of 100 mJ per pulse, meant it was able to provide sufficient illumination by pumping a fluorescein dye cell placed behind the optical nozzle as shown in Figure 4.16. The beam was angled perpendicular to the camera axis and passed through the dye cell at the same height as the optical assembly. The diffuse nature of the fluorescent light emitted allowed the dye cell to be placed directly behind the cassette without the need for an acrylic diffuser. Light intensity was controlled by adjusting the laser power and fine adjustment of the height at which the beam passed through the dye cell.

![Figure 4.16 Dye cell illumination using the Nd:YAG laser. The dye cell is placed directly behind the optical cassette to back-illuminate the nozzle assembly. Laser beam at 90° to camera axis.](image)

This technique produced uniform and easily adjustable lighting conditions and was by far the preferred technique for single shot imaging. The low pulse energy of the copper vapour laser by comparison, in the region of 2 mJ per pulse, did not provide adequate lighting levels to illuminate the high speed video by a similar method. This method therefore became interchangeable with the multi-fibre technique, dependent
4.6 Concluding Remarks

on the image capture method being used, resulting in two distinct illumination techniques being employed.

As mentioned previously in Section 3.4.2 the redesign of the optical plates, to include a polished flat on their front face, brought about a dramatic increase in image quality. This change above all was responsible for the greatest improvement in image quality seen during the study.

4.6 CONCLUDING REMARKS

The development of the image and pressure data capture techniques has been described in this chapter. The following chapters will discuss the results obtained using these methods and refer to the rig features described in chapter 3. Further examples of the techniques described in this chapter will be given in the context of the results being presented. Chapter 5 will now discuss the cavitation phenomena observed in the nozzle holes of the different optical nozzle designs.
CHAPTER 5

NOZZLE-HOLE CAVITATION PHENOMENA
5.1 INTRODUCTION

The experimental results presented in this chapter focus on the development of cavitation in the nozzle holes, and as such represent the low pressure phase of testing where the full rig capabilities were not required to achieved the necessary data. Testing below fuel injection pressures of 1000 bar was found to be more than sufficient to study the complete development of geometric cavitation, from inception at low cavitation numbers, to full development, at higher cavitation numbers. The high pressure (>1000 bar) capabilities of the rig will be briefly discussed in reference to certain results, however, a full discussion of the results achieved in this higher pressure range will be included in Chapter 6.

Results for all three nozzle types discussed in Chapter 3 are presented here, with the simplest, single-hole geometry presented first. This is in keeping with the chronological order of the experiments, which saw the single-hole geometry used initially to explore the rig’s experimental capabilities. With similar nozzle geometries being the focus of numerous prior studies this approach acted as a well documented starting point from which to progress the research, and allowed the author to familiarise himself with the basic behaviour of cavitation before the more complex flow conditions of the multi-hole nozzle were considered.

5.2 SINGLE-HOLE NOZZLE FLOW

As described in Chapter 3, the single-hole assembly consisted of two optical plates, with the 2 mm diameter hole of the sac plate and the 300 μm diameter hole of the nozzle-hole plate aligned axisymmetrically to produce a simple nozzle-flow geometry. This type of nozzle has been studied on various scales in both 2D and 3D forms by a number of research groups since the original optical study by Bergwerk (1959). The simple axisymmetric design reduces the additional flow effects introduced by more complex geometries, which may obfuscate the flow conditions within the nozzle, and allows the fundamental behaviour of a cavitating nozzle to be investigated.
Only one type of single-hole nozzle was tested as part of this study, with the effect of inlet radius and convergence/divergence, which have been the focus of previous studies, being neglected in favour of establishing a comprehensive dataset, mapping the behaviour of the nozzle across a wide range of cavitation and Reynolds numbers.

The nozzle hole was designed to have a sharp inlet to increase the likelihood of cavitation and simplify the design specification. A scanning electron microscope image of the nozzle-hole inlet, shown in Figure 1 below, revealed that a negligible inlet radius was present, and as such the flow transition from the sac volume to the nozzle-hole would not be smoothed by inlet radius effects.

![Figure 5.1(a) Scanning electron microscope plan view image of the 300 µm single-hole inlet edge, showing the orifice as black to left and the sapphire optical plate top surface to the right. (b) Cut-away side view of nozzle assembly including 2 mm diameter sac volume shown for reference.](image)

**5.3 TEST MATRIX**

To characterise the nozzle flow, the test rig was first configured to run under transient injection conditions. Downstream of the nozzle the fuel injection rate tube was fitted, allowing the discharge coefficient ($C_d$) of each injection to be measured and compared to the corresponding optical data for each fuel injection event.
5.3 Test matrix

With no *a priori* knowledge of the nozzle’s specific cavitation characteristics, a test matrix was designed to provide a structured means of exploring the development of cavitation in the nozzle. Preliminary testing of the rig hardware during commissioning had established the nozzle opening pressure (NOP) of the injector to be approximately 250 bar. To avoid the inclusion of anomalous fuel injection characteristics close to this pressure, the test matrix was limited to a minimum injection pressure of 300 bar.

As previously discussed in Section 3.7 a notable pressure drop was observed between the point of rail-pressure measurement and the optical sac volume downstream of the injector. For clarity of presentation the desired rail pressure will be used when referring to specific test conditions, with the corrected average value being used to calculate any subsequent results. Table 5.1 shows the rail pressures, \( P_{\text{rail}} \), used during testing and their corresponding corrected average sac pressure values, \( \bar{P}_1 \).

<table>
<thead>
<tr>
<th>Rail Pressure ( P_{\text{rail}} ) (bar)</th>
<th>Sac Pressure ( \bar{P}_1 ) (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>270</td>
</tr>
<tr>
<td>400</td>
<td>374</td>
</tr>
<tr>
<td>500</td>
<td>474</td>
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<tr>
<td>600</td>
<td>573</td>
</tr>
<tr>
<td>700</td>
<td>670</td>
</tr>
<tr>
<td>800</td>
<td>763</td>
</tr>
<tr>
<td>900</td>
<td>857</td>
</tr>
<tr>
<td>1000</td>
<td>952</td>
</tr>
</tbody>
</table>

Thus for a desired rail pressure of \( P_{\text{rail}} = 300 \) bar, the corrected average value of \( \bar{P}_1 = 270 \) bar will be used to calculate the injection average cavitation number, \( \sigma_{\text{av}} \), and the injection average Reynolds number, \( R_{\text{eav}} \).

From the minimum value of \( P_{\text{rail}} = 300 \) bar, the injection pressure was increased in increments of 100 bar up to a maximum pressure of \( P_{\text{rail}} = 1000 \) bar. At each desired rail pressure the nozzle back-pressure, \( P_2 \), in the rate tube was incremented in steps
of 10 bar, first increasing from a minimum value of 1 bar (atmosphere) to 10 bar then increasing regularly up to a maximum of 150 bar.

For safety reasons the test matrix was always run with an increasing back pressure, so as to gradually approach the maximum pressure of \( P_2 = 150 \) bar. Doing so meant that the higher cavitation numbers were achieved at the start of a particular test run, and cavitation was suppressed as the run progressed. For the purposes of discussion it is more suitable to talk in terms of cavitation development within the nozzle, so the results of each test will be discussed in order of decreasing back pressure for this reason.

During the first test run, multiple injections were observed at each test condition to confirm that the repeatability of the CR200 injector, seen in Figure 3.40, was replicated in the observable results. Under conditions where cavitation did not extend the full length of the nozzle-hole i.e. from initial inception to full development, no notable differences in the extent of cavitating flow region were seen between like injections. It was possible therefore to capture a single fuel injection event at each test condition, and use the collected data with a high degree of confidence in its representation of the flow at that particular condition.

### 5.4 RESULTS ANALYSIS

#### 5.4.1 Cavitation Development

Starting at a rail pressure of 300 bar, a back pressure of 150 bar produced the lowest achievable cavitation number (\( \sigma_{av} = 0.8 \)) with the transient experimental set up. Under these conditions there was no cavitation observed in the nozzle-hole entrance region and the image of the diesel-filled nozzle remained unchanged throughout the injection event.

Cavitation inception was observed to occur when \( \sigma_{av} = 0.9 \) at a back pressure of 140 bar. A faint region of vapour was seen to form in the hole-inlet region extending no further than 150 \( \mu \)m down the nozzle length. Its presence was made more obvious
5.4 Results analysis

when viewed as a video sequence, and is presented in Figure 5.2 next to an image of the nozzle where no cavitation is present to highlight its location.

![Cavitation inception](image)

Figure 5.2 Image of cavitation inception in the 300 µm single-hole nozzle (b), shown with a non-cavitating flow (a) to highlight its location.

The following image sequences in Figure 5.3, captured at 10,000 fps, show the rapid development of the cavitating flow region as the cavitation number was increased. It can be seen that the small increase in $\sigma_{av}$ from 1.1 to 1.3, brought about by the reduction in back pressure from 130 to 120 bar, was sufficient to see the cavitating region abruptly extend toward the exit of the nozzle.

![Image sequences](image)

Figure 5.3 Image sequences showing the development of cavitation within the nozzle-hole flow with increasing $\sigma_{av}$. Frames taken from high-speed video captured at 10,000 fps.
A further increase in the cavitation number saw the point that cavitation filled the nozzle-hole move closer to the start of injection. This result fitted well with the known sac-pressure profile for a single fuel-injection event, where the sac pressure did not remain constant, but instead rose steadily as the injection event progressed. The resulting transient pressure difference across the nozzle created an analogous transient value of σ. The instantaneous value of σ thus defined the tendency of the flow to cavitate during the injection, with the switch to predominantly fully developed cavitation taking place abruptly due to the square-edge of the hole-inlet.

Figure 5.4 illustrates this effect by plotting the sac pressure and the instantaneous values of σ for the test conditions shown in Figure 5.3 during a single fuel injection event. Sac pressure is plotted against the left x-axis for reference.

Figure 5.4 illustrates this effect by plotting the sac pressure and the instantaneous values of σ for the test conditions shown in Figure 5.3. For lower back-pressures the instantaneous value of σ reached the critical values for inception, σᵢ, and full development, σᵢ, at progressively earlier stages of the injection event; correlating well with the imaged results. Consistent values of σᵢ and σᵢ were observed to exist across injections with differing values of σᵢ; the values of σᵢ for the relevant 300 and 400 bar rail pressure cases are shown in Figure 5.5.
5.4 Results analysis

It should be noted that this analysis is not definitive, as the sac pressure trace used to calculate the instantaneous values of $\sigma$ was obtained with a back pressure of atmosphere, and variations in the trace profile caused by back pressure, were unknown. However, the close correlation of these results suggests that these effects were limited.

This highlighted that characterisation of the nozzle’s cavitating behaviour using transient injections must be qualified with the transient characteristics of the injection event. To use a single value of $\sigma$ (i.e. $\sigma_{av}$) would be misleading for injections that included both non-cavitating and cavitating flow. For example, having determined that fully developed cavitation occurred for $\sigma \geq 1.35$, the value of $\sigma_{av} = 1.3$, calculated for the case shown in Figure 5.3(b), suggests that the cavitation would not have become fully developed under these conditions. This was clearly not the case, with nearly one third of the injection duration containing significant amounts of cavitation; which could lead to potentially unexpected injection characteristics. Unlike calculating an average Reynolds number across the injection, $\sigma_{av}$, in this case, does not account for the presence of two discrete flow regimes during the injection.

The use of a square-edged hole-inlet had the effect of reducing the number of test conditions where this transitional flow was observed and also lowered the value of $\sigma_f$. So for this study, the impact of the discrepancy in the flow information provided...
by $\sigma_{av}$ was minimised. For $\sigma_{av} \geq 2$, the onset of fully developed cavitation was not discernable from the start of injection. Therefore, it could be said that for injections where $\sigma_{av} \geq 2$ the nozzle-hole would contain fully developed cavitating flow for the duration of the injection, giving a substantive validity to the use of $\sigma_{av}$ to describe the nature of the flow.

In real fuel injectors a certain amount of hole-inlet radius and irregularity is to be expected, which would have the effect of smoothing the transition from non-cavitating to fully developed cavitating flow and thus raise the value of $\sigma_{f}$, possibly to within the working range of the injector. Unless a “square” injection profile could be produced, the issue of variance in $\sigma$ through the injection would need to be considered in the design or characterisation of a fuel system.

Proceeding with the test matrix, there were a limited number of test conditions where $\sigma_{av} \leq 2$. For $P_{rail} = 300$ bar, back pressures in the range of 100-150 bar there was perceivable development of the cavitating flow during the injection; this range subsequently reduced to 130-150 bar when $P_{rail}$ was increased to 400 bar. Beyond these test cases, the fully developed state of the cavitation prevented optical data from adding further value to the existing data, other than to prove that its appearance remained unaltered as the value of $P_{rail}$ was increased to 1000 bar. As a result, the additional points of the test matrix were not required for model validation and the additional data were captured in order to complete the preliminary phase of testing.

Upon completion and review, the region where $\sigma_{av} \leq 2$ was revisited to provide a more comprehensive data set for this condition. Back pressure was incremented in steps of 5 bar to provide interim data points between those already collected. The second iteration of the optical plate design, incorporating the flat front face, was now available and was used for this additional phase of testing.

The improvement in image quality, apparent in Figure 5.6, gave improved insight into the nature of the cavitating flow structure, with flow structures resolvable to approximately 10 µm. Fine vapour filaments and individual bubbles were now
visible at the extent of the cavitating flow region, which itself had film-like appearance (Figure 5.6 (b)).

Figure 5.6 Images showing (a) the full extent of cavitation for, $P_{\text{rail}} = 300$ bar, $P_2 = 125$ and (b) The ‘film-like’ appearance of the cavitating flow at $P_{\text{rail}} = 300$ bar, $P_2 = 105$ bar

Figure 5.6 (a) shows the maximum extent of the cavitating flow region for $P_{\text{rail}} = 300$ bar, $P_2 = 125$ bar, capturing a condition between the two previous test points that subtended the switch to a fully developed flow. Here cavitation was restricted to 1.5 mm of the nozzle-hole’s 2 mm length, having a maximum instantaneous value of $\sigma = 1.31$, which correlated well with value of $\sigma_f = 1.35$ previously calculated.

### 5.4.2 Discharge Coefficient

Figure 5.7 shows the rate tube pressure traces for the non-cavitating $P_{\text{rail}} = 300$ bar, $P_2 = 150$ bar and the $P_{\text{rail}} = 300$ bar, $P_2 = 120$ bar case, where we see cavitation development part way through the injection. As described in Chapter 3, the area under these plots was integrated between the start and end of injection to calculate the fuel mass delivered during the injection event.

Using this data and the average pressure drop across the nozzle ($\bar{P}_1 - P_2$), the average discharge coefficient, $\bar{C}_d$, of the nozzle for a particular injection condition can then be calculated.

Before calculation, the influence of cavitation on the nozzle mass flow rate was apparent in the two data plots. The known rise in pressure toward the end of injection
at $P_{\text{rail}} = 300$ bar was matched by a similar rise in mass flow rate in the non-cavitating nozzle. In the case where cavitation was known to be present this rise was somewhat suppressed, illustrating that the presence of cavitation was restricting the mass flow through the nozzle.

Figure 5.7 Rate tube pressure traces for (a) $P_{\text{rail}} = 300$ bar, $P_2 = 150$ bar and (b) $P_{\text{rail}} = 300$ bar, $P_2 = 120$ bar

To show the related change in the discharge coefficient of the nozzle, instantaneous values of $C_d$ are plotted in Figure 5.8 for the two injections. Values of $C_d$ remained consistently above 0.8 for the non-cavitating flow, but were seen to decrease by around 5% at a point that coincided with the development of cavitation in the nozzle. The change was not as abrupt as the appearance of the cavitating flow itself, but decreased at a rate that appeared proportional to the increase in cavitation number.

Figure 5.8 Instantaneous discharge coefficient plotted for (a) $P_{\text{rail}} = 300$ bar, $P_2 = 150$ bar and (b) $P_{\text{rail}} = 300$ bar, $P_2 = 120$ bar. Sac pressure is plotted for reference.

Given that the $\sigma_{av} \leq 2$ condition was of the greatest interest for this preliminary work, in terms of the observable changes in flow behaviour and the validation data it provided, the data from the remainder of the original test matrix was analysed
5.4 Results analysis

primarily as a means of mapping the relationship between Reynolds number, cavitation number and discharge coefficient, across a broader set of results.

Figure 5.9 plots the average discharge coefficient of each injection against cavitation number (σ_{av}) for each point on the test matrix. The dependency of the discharge coefficient on the cavitation number is clear, reducing significantly with the onset and continued development of cavitation. \( \bar{C}_d \) tends towards the value of the vena contracta coefficient, \( C_c \), which represents the limit of the flow area through the nozzle. Under these conditions there is no pressure recovery along the nozzle length and the discharge coefficient represents the fraction of the flow area occupied by the liquid jet (Schmidt and Corradini, 2001).

![Figure 5.9 Graph of average discharge coefficient \( \bar{C}_d \) against average cavitation number \( \sigma_{av} \) for the complete test matrix](image)

For completeness the discharge coefficient is also plotted against Reynolds number in Figure 5.10 to confirm its independence from this parameter under cavitating flow conditions.
5.4 Results analysis

Figure 5.10 Average discharge coefficient $\bar{C}_d$ plotted against average Reynolds number $Re_{av}$ for the complete test matrix

5.4.3 Summary

The study of the single hole geometry proved to be a useful exercise in testing the capabilities of the rig. In completing the test matrix, imaging at rail pressures of 1000 bar was achieved and the limitations of certain experimental approaches and equipment were highlighted. The exercise also provided a useful initial data set, from which City University were able to commence the validation of their cavitation model sub-routines. The main focus of the study now moved on to the multi-hole nozzle geometries to enable investigation of more complex flow phenomena.
5.5 Multi-Hole Nozzle Flow

5.5 MULTI-HOLE NOZZLE FLOW

The experimental programme for the multi-hole nozzle testing was split into a low and a high pressure data set. The borosilicate multi-hole nozzles, shown in Figure 5.11, were used for the low pressure work, whilst the sapphire nozzles were reserved for the high pressure testing. The results of the high pressure testing led directly into the experimental data discussed in the following chapter, and as such will be addressed separately toward the end of this chapter.

FEA analysis carried out by Caterpillar suggested that a maximum injection pressure of 500 bar could be achieved using the borosilicate nozzles. Although significantly less than the 1000 bar cut-off for the low pressure single hole work, this pressure range still provided adequate scope to produce a useful data set.

5.5.1 Large Multi-hole Nozzle flow

Analysis of the 300 µm multi-hole nozzle plate showed several deviations from the design drawings. Irregularities were visible on the inlet edges of the holes and there was a varying degree of eccentricity in the circumference of each hole; the extent of this deviation can be seen in Figure 5.12.

Figure 5.11: Images of the (a) large, 300 µm multi-hole and (b) small, 110 µm multi-hole borosilicate nozzle plates. OD of plates = 10 mm
Additionally there was an anti-clockwise rotation of the two rear holes of approximately 10° about the centre of the plate, causing them to not align perpendicularly with the flats on the sides of the plate. This meant that the rear-right hole (ref: Figure 5.13) was positioned further underneath the obstruction than the rear-left hole. From an experimental point of view, these physical attributes did not affect the results gathering process and were simply taken into account when describing certain observations.

The deviation from the ‘ideal’ case did however, mean that any simulation work based on ideal geometries, was susceptible to similar deviations from the experimental results. For this reason, a metrological study of the plates was undertaken and the results were used to build realistic model geometries for simulation purposes. Doing so also highlighted the shift in the –ve y-axis direction of the rear holes, positioning the rear-right hole closer to the centreline of the plate.
Additionally, the obstruction plate was measured, as manufacturing limitations had also caused it to have a non-ideal profile. A coordinate measuring machine (CMM) analysis of its edge profile and protrusion height allowed realistic representation of its geometry in the simulation. The metrological data was passed to City University and Caterpillar who developed the following volume mesh in Gambit for use in their simulations.

![Simulation mesh of large multi-hole nozzle assembly consisting of 500,000 volume cells. Image provided by City University, London – Giannadakis (2005)](image)

**5.5.1.1 Test Matrix Analysis – String Cavitation**

A structured test matrix was again undertaken to cover the operating pressure range of the nozzle. The rig was set up in its transient flow configuration, injecting into the rate tube. Due to the reduced rail pressure range that was achievable with the borosilicate nozzles, rail pressures were incremented by 50 bar instead of 100 bar, as done previously, to increase the size of the data set.

The pressure drop observed in the sac pressure measurements, with reference to $P_{\text{rail}}$, was more significant with the large multi-hole geometry than for that of the single hole. As shown in Section 3.7, peak sac pressure fell approximately 50 bar below the desired value of $P_{\text{rail}}$, with the average value of $P_1$ for the injection duration falling approximately 100 bar below $P_{\text{rail}}$. This difference was consistent across the range of rail pressures tested, and as before, the desired values are used here to simplify the presentation of results whilst the measured values were used for calculation and to describe particular flow conditions where necessary.
Again, the back pressure \( P_2 \) was incremented in 10 bar intervals from 10 bar up to 150 bar. Injection duration was set at 2 ms and high speed imaging was used to capture the cavitating flow development at each point of the test matrix.

Unlike the single-hole sapphire nozzle, cavitation development was far more protracted for the multi-hole nozzle as a result of the asymmetric flow geometry and the irregular profile of the hole-inlets. This allowed the development of the cavitating flow features to be observed over a broader range of flow conditions, with the initiation of particular features still observable at the maximum rail pressure of \( P_{\text{rail}} = 500 \) bar.

As will be shown in this discussion, certain predictable patterns of development existed in the formation of cavitation in the nozzle. However, these existed alongside highly transient flow features that showed a marked dependency on the instantaneous flow conditions within localised regions of the flow geometry. The additional asymmetry of the flow path, introduced by the misalignment of the holes, also served to highlight the sensitivity of these features’ formation to the location of the physical flow boundaries within the nozzle geometry.

Figure 5.15 illustrates the nozzle orientation, such that when imaged, the two rear holes located under the obstruction appeared either side of the front hole.
5.5 Multi-Hole Nozzle Flow

Imperfections in the borosilicate appeared as permanent dark features in the image, with the outline of the holes otherwise indistinguishable due to the close refractive index match of the borosilicate to the diesel (1.51:1.46).

As with the previous analysis of the single-hole results, the discussion will follow the process of cavitation development within the nozzle and thus will begin by looking at the high back pressure cases; starting at a back pressure of 150 bar.

For rail pressures below 500 bar, geometric cavitation was suppressed in all of the 150 bar back pressure cases. However, visible regions of two-phase flow were present under these conditions in the form of vapour-containing vortices. For $P_{\text{rail}} = 350$ bar an intermittent vortical flow structure was observed in the right hand hole, extending into the hole entrance and occasionally anchoring itself to the underside of the obstruction plate.

Figure 5.16, taken from the high speed video, shows the very faint vortex extending across the hole entrance.

![Vortex structure extending across hole entrance](image)

Figure 5.16 Frame grab from high speed imaging showing vortex in hole entrance region of the right hand hole. $P_{\text{rail}} = 350$ bar, $P_2 = 150$ bar

The presence of this vortex was only apparent as a result of the vapour phase at its core. The origin of this vapour was not clearly defined and could be perceived to be the result of either cavitation in the low pressure core region, or the entrainment of
free-stream vapour bubbles into the vortex. In both cases it was the entrainment of a nucleus of critical size or greater that likely initiated the growth and appearance of the vapour core, or cavitation string.

Figure 5.16 captures a moment where vapour is present in both regions of the nozzle; however, it was more common for vapour to be visible in either the sac volume or the hole entrance exclusively, as is shown in Figure 5.17. This implied that temporal changes in vortex intensity existed along the vortex axis, affecting an axial pressure gradient that governed the location of the visible region. Taking into account the known fluctuations in sac pressure, this implied that there was no preferential location for the intense region of the vortex within this area of the nozzle geometry.

Increasing the area of interest from the entrance region of the right hand hole, the asymmetry of the two rear holes had an evident affect on the observations that were made. The right hand hole’s position, further underneath the obstruction, appeared to lend itself to more intense vortex formation. The fact that no vortex was visible in the left hand hole provided a useful insight into the role of the obstruction in creating the vortical flow structure. Even though the left hand hole was ‘covered’ by the obstruction, its proximity to the obstruction’s front edge appeared sufficient to suppress formation of a vortex intense enough to produce a cavitation string under these conditions. An increase in nozzle flow-rate saw the emergence of a corresponding on-axis cavitation string in the left hand hole, as can be seen in Figure 5.17.
5.5 Multi-Hole Nozzle Flow

5.18, but its appearance, in terms of size, was predominantly retarded in comparison to that of the right hand hole, suggesting a likewise deficiency in vortex intensity.

Growth of the cavitation strings and the emergence of an additional vortex structure were observed with increasing cavitation number. This third, bridging vortex developed in the sac volume between the two rear holes and formed an arc whose ends terminated in the hole entrance regions. A reduction of just 10 bar in back pressure to $P_2 = 140$ bar for the $P_{rail} = 350$ bar case was sufficient to bring about its appearance, however, its dimensions and intermittent residence time during the injection were so small at this point, that it was only apparent in a moving image, where the changes between frames could be perceived.

Two consecutive frames are presented side-by-side in Figure 5.19, so that the appearance of the vortex in the second frame is more obvious.
The hole-to-hole interaction was consistent with the findings of Arcoumanis et al. (1999) and suggested the transport of vapour between the holes. For the purposes of discussion in this chapter the behaviour of this vortex will be examined in terms of its interaction with the on-axis vortices in the hole entrance regions. A more detailed discussion of its behaviour is carried out in Chapter 6.

Figure 5.20 captures the interaction of this vortex with the on-axis vortex in the entrance region of the right hand hole after a further drop in back pressure to 100 bar.

The opposing end of the bridging vortex could be seen to extend into the entrance region of the left hand hole, implying the presence of vortical flow in that region. The lack of a visible cavitation string however, highlighted the fact that without vapour being present in the vortex core, the presence of that particular flow structure could not be confirmed. The flow was evidently unstable and it was unclear as to whether the extent of this instability was being fully described by the observed cavitation strings, or that alternatively, a lack of free-stream vapour was preventing the observation of all active flow features.

The previous hypothesis suggested that free-stream bubble entrainment was a possible source of vapour for the vortex cores (Arndt and Maines, 2000), and it follows that the presence of greater quantities of vapour in the surrounding fluid would make their appearance more likely. This was found to be true to a certain extent, with the three dominant vortices becoming visible as the observable inception...
of geometric cavitation in the nozzle holes was approached. It was generally observed that increases in cavitation number brought about an increase in the prominence of the cavitation strings, and that increases in Reynolds number saw the extent to which the cavitation string reached into the nozzle hole increase; here, Reynolds number is used to express the flow velocity, where physically it may not be the right choice.

However, the development of geometric cavitation did not guarantee the formation of cavitation strings, and they remained highly transient whilst their maximum physical size appeared to stagnate. The disparity between the vortices in the right and left-hand holes remained, and the appearance of the bridging vortex seemed dependent on the presence of one or both of the on-axis hole-vortices.

**5.5.1.2 CFD Results**

Figure 5.22 and Figure 5.23 were kindly provided by Dr. E. Giannadakis and Dr. M. Gavaises of City University London to aid the discussion of the experimental results and should not be considered as the work of the author. Full details of the validated simulation model employed to achieve these results can be found in Giannadakis (2005) and Giannadakis *et al.* (2008).

The use of the CFD simulations to better understand the fundamental aspects of the flow proved to be a highly valuable resource. The additional insight into the complex...
flow behaviour of the nozzle complimented the experimental findings, producing a more robust description of the nozzle flow.

Figure 5.22 contains the velocity streamlines predicted by the model for $P_{rail} = 350$ bar, $P_2 = 40$ bar. The recirculating flow region under the obstruction and the ensuing rotational hole-flow, responsible for the observed vortex formation, were both evident, and provided an enhanced physical understanding of the mean flow structure.

The simulation also allowed determination of the rotational direction of the flow, which was not possible from the imaging alone due to the short residence time of the flow features and the directional ambiguity of the still images. It was shown that two counter-rotating vortices were created under the obstruction, driven by the geometry of the sac and the location of the holes. The outermost flow in these counter-rotating vortices then interacted with the incoming bulk flow re-circulation region under the centre of the obstruction and created the lateral re-circulation that led to the formation of the bridging vortex.
The ratio between the tangential and axial flow velocities defined the swirl of the nozzle flow, and was used as an indicator for regions of potential string formation. Figure 5.23 shows the iso-swirl surfaces for the $P_{\text{rail}} = 350$ bar, $P_2 = 40$ bar case revealing the regions of swirl in the sac volume above each rear-hole entrance, and the conservation of this swirl level along the length of the nozzle hole. Also present was the region of lateral swirl in the sac volume between the holes, correctly predicting the location of the bridging string. The combined insight provided by the experimental observations and the CFD simulations allowed the schematic of the sac-flow conditions shown in Figure 5.24 to be established.
5.5 Multi-Hole Nozzle Flow

5.5.1.3 Summary

The observation of string cavitation in the 300 µm multi-hole nozzle geometry preceded the formation of visible quantities of geometric cavitation in the nozzle-holes. There was however, a link to cavitation number, with the prominence of the cavitation strings increasing with increasing values of \( \sigma \). This suggested that the increase in free stream vapour, brought about by the increase in \( \sigma \), was the dominant source of vapour for the formation of cavitation strings. The onset of geometric cavitation did not bring about the steady appearance of the strings. Instead, the highly transient nature of their formation alluded to the presence of unsteady flow structures in the sac volume. These flow features showed a marked dependency on the instantaneous localised flow conditions, and sensitivity to their location with respect to the physical flow boundaries of the nozzle geometry. The interaction of these flow features with the development of geometric cavitation will now be examined.

5.5.1.4 Test Matrix Analysis - Geometric Cavitation

The inception of geometric cavitation was observed to occur in all three holes simultaneously, with the ensuing cavitation development proceeding more rapidly in the central hole; positioned on-axis with the incoming bulk flow from the upstream injector.

![Image](image.png)

Figure 5.25 Cavitation inception at \( \text{P}_{\text{rail}} = 500 \) bar, \( \text{P}_2 = 140 \) bar showing similar levels of development in each hole

As mentioned previously, the inception and development of cavitation was far more protracted than for the single hole case. Inception was observed at all rail pressures tested; appearing at \( \text{P}_2 = 140 \) bar for the \( \text{P}_{\text{rail}} = 500 \) bar case. It was noted that
incipient cavitation vapour was visible for the same period of the injection duration for the cases of $P_{\text{rail}} = 350$ bar, $P_2 = 90$ bar and $P_{\text{rail}} = 500$ bar, $P_2 = 140$ bar. Figure 5.26 shows the instantaneous values of $\sigma$ plotted for these two cases, and as with the single hole geometry, a good correlation between time of visible inception and a common value of $\sigma_i$ was found. This value was $\sigma_i = 1.85$.

![Figure 5.26 Values of cavitation number, $\sigma$, plotted for (a) $P_{\text{rail}} = 350$ bar, $P_2 = 90$ bar and (b) $P_{\text{rail}} = 500$ bar, $P_2 = 140$ bar. Inception observed at a common value of $\sigma_i = 1.85$, $t \approx 1$ ms ASOI.](image)

The development of cavitation for $\sigma \geq 1.85$ saw a difference in behaviour for the three nozzle holes. Development in the two rear holes was retarded with respect to the front hole, a typical example of which is shown in Figure 5.27. Here, 1.6 ms after SOI for $P_{\text{rail}} = 450$ bar, $P_2 = 80$ bar, $\sigma = 3.80$ and the development of the cavitation in the two rear holes was approximately half that of the front hole, which was almost fully developed.

![Figure 5.27 Frame grab showing retarded development of rear holes with respect to the front hole during a transient fuel injection event. $P_{\text{rail}} = 450$ bar, $P_2 = 80$ bar, $\sigma = 3.80$](image)
This behaviour can be explained by the difference in the flow characteristics of the nozzle-hole in-flow: The restriction in the flow path over the rear holes, created by the obstruction, would have had the effect of reducing the local fluid pressure under the obstruction, and hence reduce the effective cavitation number of the rear holes. It must also be considered that fluid entered the holes asymmetrically, imparting a tangential velocity component on the hole in-flow, which would effectively reduce the pressure drop experienced by the fluid as it accelerated into the holes.

During the test matrix analysis, the effect of swirl had no direct influence on the extent of the cavitation development. The presence of a visible core bore no correlation to the amount of cavitation produced in a particular hole, and the disparity seen in the levels of vorticity between the holes was not reflected in the development of cavitation. Furthermore, the presence of geometric cavitation obscured the view of any small scale in-hole vortices, preventing their visible detection.

Fluctuations in the extent of cavitation in the two holes were observed during development, but with no correlation to swirl-levels, these were more likely the result of turbulent flow variations in the sac volume affecting changes in the localised cavitation number. Figure 5.28 illustrates the differences that were present. In both images a vortex core, of similar dimensions, is visible above the right hand hole, whilst the extent of cavitation varies between like and opposing holes.

![Figure 5.28](image-url)
5.5 Multi-Hole Nozzle Flow

From this analysis, and that of the previously described string cavitation behaviour, it could be surmised that turbulent perturbations in the fluid-flow, beneath the obstruction, created highly dynamic pressure-field variations in the near-hole sac volume. Due to the flow geometry these led to the transient formation of coherent vortical flow structures, which were then convected in to the nozzle-hole flow, entraining free-stream vapour to produce a visible vapour core, or cavitation string. At the same time, under conditions where the nozzle had the propensity to produce geometric cavitation, the localised pressure variation produced an analogous change in the cavitation number, which independently affected each of the holes.

Fully developed cavitation formed in all three nozzle holes as cavitation number was increased. For the front hole this occurred at $\sigma = 4.2$, however, the full development of cavitation in the rear holes occurred at a value of $\sigma$ that increased with the decreasing back pressure as shown in Table 5.2.

<table>
<thead>
<tr>
<th>$P_2$ (bar)</th>
<th>$\sigma$ for front hole full development</th>
<th>$\sigma$ for rear holes full development</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>4.15</td>
<td>7.3</td>
</tr>
<tr>
<td>50</td>
<td>4.2</td>
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</tr>
<tr>
<td>70</td>
<td>4.1</td>
<td>4.9</td>
</tr>
</tbody>
</table>

This was conducive to the theory that the acceleration of the flow under the obstruction caused the local cavitation numbers of the rear holes to be less than that for the front hole. Figure 5.29 illustrates the deviation in pressure under the obstruction, by an amount $\Delta P$, below the main sac pressure, $P_1$, caused by the flow acceleration. Decreasing $P_2$ would further accelerate the flow and thus increase the value of $\Delta P$. Therefore, for a given value of $P_1$ the actual cavitation number in the rear holes would be less than that of the front hole by an amount dependent on the value of $P_2$. To achieve a given cavitation number in either of the holes, the value of $P_1$ must be higher for the rear holes than for the front. Therefore, calculating $\sigma$ for the front and rear holes by only using values of $P_1$, the rear holes would appear to require a larger cavitation number to replicate the conditions seen in the front hole. Similarly, as $P_2$ increased, the pressure drop was reduced and for lower cavitation
5.5 Multi-Hole Nozzle Flow

numbers, particularly where inception was observed, the three holes exhibited similar characteristics for a given value of $P_1$.

![Diagram of Multi-Hole Nozzle Flow](image)

Figure 5.29 Deviation in pressure from $P_1$ caused by flow acceleration under the obstruction. $P_2$ is the common back pressure for both nozzle-holes

In an extension of the test matrix, high-resolution imaging was used to capture the interaction between the vortex structures and the incipient geometric cavitation. Figure 5.30 shows a series of images of the right-hand hole, taken from separate injection events, where $P_{rail} = 400$, $P_2 = 90$ bar and incipient cavitation was present on the hole inlet region.

The detail of the vapour core surface can be seen in each image, surrounded by cavitation vapour bubbles that have become entrained in the rotating flow of the nozzle hole. The irregular and non-continuous nature of the visible core was indicative of significantly small-wavelength vortex unsteadiness, similar to that observed by Chang et al. (2007) in free-stream wing-tip vortices. This unsteadiness is consistent with the turbulent nature of the flow that leads to the formation of the vortex, but additionally shows the small-scale turbulent effects that lead to the transient location of the visible core.

Two separate cores are visible in the final two images (Figure 5.30 (g) and (h)). The left of these is the termination of the bridging vortex, however, in both cases the extent of the visible core is seen to terminate at the hole entrance, with no detectable continuation of the vortex into the sac volume. Similarly, the downstream extent of the core terminates before it is possible to ascertain fully whether or not the two vortices combine to common core. This would appear to be the case in the final image (h), though it is possible that the image perspective is creating this illusion.
Figure 5.30 (f) shows quite clearly the incipient cavitation being drawn into the vortex very close to the hole entrance. A large diameter core is visible in this region, which appears to dominate the vapour entrainment at this instant. Downstream, vapour filaments remain drawn to the centre of the hole implying a continuation of the flow rotation; however, these remained distinctly separated suggesting an abrupt breakdown of the more intense hole-entrance vortex.

Figure 5.30 Interaction of incipient cavitation with nozzle-hole vortex flow. Non-sequential images taken for $P_{\text{rail}} = 400$ bar, $P_2 = 90$ bar
In a further nozzle set up, erratic behaviour was observed in the right-hand hole of the nozzle. The following images, taken for $P_{\text{rail}} = 400$ bar, $P_2 = 30$ bar, show that the amount of geometric cavitation was highly-variant in the right hole, while levels in the left and front hole remained in a constant, fully developed, state. High levels of swirl were evidently present in the hole, though again a driving effect on the development of the geometric cavitation could not clearly be established. Instead the diameter of the string appeared to remain dependent on the quantity of geometric cavitation vapour present in the nozzle – less vapour pertaining to a narrower string as before – with the increased levels of vapour serving to better define the flow structure along the hole-length. This is particularly evident in Figure 5.32 where a distinct vortex is outlined as the geometric cavitation is drawn toward the centre of the rotating hole-flow. Figure 5.31, for the same test condition, exhibits the flow characteristics seen for low cavitation numbers in the test matrix (ref: Figure 5.21 (b)), indicating a substantial difference in the pressure boundary conditions of the two rear holes. This difference is eliminated in Figure 5.33, which shows a very large diameter vapour core in the sac volume of the right hole, and near-identical in-hole flow conditions for both holes. The appearance of the core in this instance matches that of cores that were sustained close to the end of injection, where flow-rate and rotation rapidly decreased. Therefore this feature of the core can be used here as an indicator to suggest that flow rotation had significantly reduced, whilst, nozzle flow-rate remained sufficiently high to sustain geometric cavitation.

The constant geometry of the nozzle throughout this series of images indicated the levels to which the localised flow conditions varied at the entrance to this particular hole. Variation in the left hand hole, by comparison, was not evident. All images were captured during the middle of the injection event (approx. 1 ms after SOI), and sufficient bulk nozzle flow-rate could be guaranteed by the presence of geometric cavitation in the other two holes.

From the alignment of the holes it could be seen that the assembly of the nozzle on this occasion had reduced the ‘hole-skew’ to a certain degree. The reason for this variation could thus be attributed to the altered location of the hole in relation to the
surrounding physical boundaries. Asymmetry of the flow behaviour had been observed in the test matrix, and these results represented a likely exaggeration of that effect. In contrast to the previous hypothesis, however, disparity between the rear holes had now increased with the right-hole positioned closer to the front edge of the obstruction. Therefore, simple correlation of the hole-flow characteristics to the x-axis position of the hole (ref. Figure 5.13) could not be substantiated, and brought into consideration the more complex interdependence that the 2D position of the holes, acting as sinks, played in determining the upstream flow structure in the sac volume.

Above all, these images provided evidence for the marked differences in nozzle-flow characteristics that the highly transient flow conditions in the sac volume could create. From an experimental point of view these results gave an interesting insight into the sensitivities of the flow to the hole position, but from a modelling perspective they presented a substantial challenge to correctly predict nozzle flow that varied so greatly with consistent boundary conditions.

Figure 5.31 Image of geometric cavitation suppression in the right hole; characteristic of a low local value of $\sigma$.

$P_{\text{rail}} = 400$ bar, $P_2 = 30$ bar
5.5 Multi-Hole Nozzle Flow

Figure 5.32 Incipient cavitation defining the presence of a strong vortical flow structure in the right nozzle-hole. \( P_{\text{rail}} = 400 \text{ bar}, P_2 = 30 \text{ bar} \)

Figure 5.33 Right hole exhibiting characteristics of low flow rotation and sufficient axial flow rate to sustain fully developed geometric cavitation. \( P_{\text{rail}} = 400 \text{ bar}, P_2 = 30 \text{ bar} \)
The findings of the study thus far corresponded well with the observations of Andriotis et al. (2008) that attributed variation in nozzle-hole flow-rate and cavitating flow structure to the intermittent presence of vortices attached to the hole-entrance. Furthermore, Gavaises and Andriotis (2009) established a link between the formation of string cavitation in the nozzle hole and an increase in the spray angle. The results of the current study suggest that the formation of string cavitation appeared as evidence of a coherent vortex structure within the sac volume and/or the nozzle-hole. The increase in tangential velocity, imparted on the nozzle-flow during the residence of the vortex, would thus manifest itself as an increase in spray angle as the swirling flow exited the nozzle-hole. Therefore, it could be inferred that the increase in spray angle was not the direct result of string cavitation, but rather both resulted from coherent flow rotation about the nozzle-hole axis.

Improved illumination of the nozzle flow allowed the fine surface details of the fully developed cavitating flow to be observed. Figure 5.34 shows two separate images of the cavitating flow structure in the front hole at slightly different stages of development during an injection for $P_{\text{rail}} = 400$ bar, $P_2 = 40$ bar. The flow here is dominated by a vapour film that can be seen to extend the entire length of nozzle hole in certain regions of the flow. Where flow turbulence disrupts the film structure, the cavitation breaks down to form a more bubbly flow, consisting of detached film filaments and individual bubbles; creating two distinct flow regimes within the nozzle flow. These different regimes of cavitating flow have previously been identified by Arcoumanis et al. (2001) in large scale nozzles, and as with those findings it could also be shown here that the upstream flow geometry had significant influence on the physical appearance of the cavitating flow.
In this study the effect of upstream geometry could be examined simultaneously by way of the established difference in the hole in-flow characteristics between the front and rear holes. Figure 5.35 shows all three holes in the fully developed state. The depth of field achievable with the lens system employed, prevented all three holes from being in focus at the same time. Therefore, image (a) is focussed on the rear holes and image (b) is focussed on the front hole.

The small-scale turbulent flow features, thought responsible for the irregular surface appearance of the cavitation strings, also had an evident effect on the structure of the developed geometric cavitation in the rear holes. The asymmetric hole in-flow and established unsteadiness of the flow within the holes prevented development of the smooth film-like structures of the front-hole cavitation. Instead the flow quickly
established the previously observed bubbly appearance seen at the breakdown of the film-stage.

A further feature of the flow noted in this particular nozzle assembly was that of secondary vorticity in the hole flow. This manifested itself as distinct ligaments of cavitation extending from the main body of cavitating flow. Again, referring to Chang et al. (2007), this flow feature could be perceived as the result of two interacting vortices in the confines of the nozzle hole. Previous examples of twin vortices within a single nozzle-hole have been presented and it is therefore possible, considering the transient nature of the flow, for this interaction to take place. Alternatively, these features could have been the result of vapour interacting with regular eddy shedding from the hole inlet edge. However, as with many of the observations made in this study, the transient nature of the flow dictated that the more subtle flow features were hard to replicate, and so conclusive evidence of their formation mechanism was not forthcoming during this investigation.
5.5 Multi-Hole Nozzle Flow

5.5.1.5 CFD Results

Once again images from the CFD simulations of the nozzle flow were provided by City University to aid the results discussion and demonstrate the correlation between the model and the experimental results. Figure 5.37 shows cavitation inception, illustrated by vapour content of the flow, occurring at $\sigma = 1.73$. This appears slightly in advance of the experimental results where $\sigma = 1.85$. However, it must be considered that the results are presented here as an iso-surface of 3% vapour, which effectively amplifies the presence of the vapour in the simulation where it would otherwise be undetectable using the purely optical experimental approach.
Figure 5.37 Comparison of CFD simulation with experimental data for $\sigma = 1.73$. Iso-surface of 3% vapour used in CFD image. Image courtesy of City University – Giannadakis (2005)

Figure 5.38 Comparison of CFD simulation with experimental data for $\sigma = 2.18$. Iso-surfaces of 10%, 20% and 30% vapour used in CFD image. Image courtesy of City University – Giannadakis (2005)

Figure 5.38 shows simulation results for $\sigma = 2.18$ alongside the equivalent experimental condition. Here, iso-surfaces of 10%, 20% and 30% vapour are shown, which again has the effect of exaggerating the visible vapour content of the nozzle flow. The model was accurate in showing similar levels of cavitation development in each of the holes and capturing the entrainment of cavitation into the swirling flow of the nozzle-hole. Iso-surface values of >15% seemed to best capture the location of visible vapour in the experimental case; a level which is confirmed by the results shown in Figure 5.39. Deviations or imperfections in the hole-edge profile showed a
tendency to be strong sources for cavitation development and thus highlighted the model’s sensitivity to such features.

For the observed fully developed cavitating flow conditions, the model successfully demonstrates the reduced levels of cavitation vapour in the rear holes. This is consistent with the observed results during cavitation development (ref. Table 5.2) and shows a continuation of this trend once the nozzle-hole flow is obscured by the visible vapour.

Further proof of this trend’s continuation could have been provided by comparing the discharge coefficients of each hole, however, the experimental apparatus did not allow for the average discharge coefficient, $\bar{C}_d$, of the individual holes to be measured. Taking a single value of $\bar{C}_d$ for the entire nozzle would have provided limited information about the three holes, with each exhibiting their own flow characteristics. Gaining an insight into the effect of observable differences between the holes on their individual discharge coefficients required significant adaptation of the rate tube and downstream flow geometry, and as such was considered the subject of further work.
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5.5.1.6 Summary

The development of geometric cavitation in the nozzle-holes was subject to the same turbulent perturbations that had been previously observed in the formation of string cavitation. The localised sac pressure variation produced an analogous change in the local cavitation number, which independently affected each of the holes. The magnitude of this effect was highly sensitive to hole position, and the interdependence that the hole position shared with the structure of the upstream sac flow. These turbulent perturbations were intrinsically linked to the transient formation of coherent vortex structures, which were the primary mechanism of string cavitation formation. Therefore, it could be said that string cavitation was a result of the change in flow structure, and not a cause of it.

5.5.2 Small Multi-Hole Nozzle Flow

An identical test matrix to the one undertaken for the 300 μm multi-hole nozzle was planned for the 110 μm design to provide a set of directly comparable results. The metrological analysis of the plate, shown in Figure 5.40, revealed similar deviations in both position and eccentricity of the holes to those of the 300 μm design. Again, this data was passed to City University and Caterpillar to create a realistic model geometry for the simulations (Figure 5.41).

Figure 5.40 Plot of hole-edge coordinate string for small borosilicate multi-hole nozzle plate. 0,0 denotes the centre of the plate
5.5 Multi-Hole Nozzle Flow

Testing began at a rail pressure of \( P_{\text{rail}} = 300 \) bar and a back pressure of \( P_2 = 10 \) bar, from which \( P_2 \) was increased in steps of 10 bar. Upon reaching a value of \( P_2 = 40 \) bar there was a complete mechanical failure of the nozzle and no further results were obtainable. Repeat attempts to obtain data at greater pressures (of both \( P_{\text{rail}} \) and \( P_2 \)) were unsuccessful and the test matrix was abandoned as a result.

Captured video showed a similar difference in appearance between the front and rear holes as was observed in the 300 µm nozzle. The front hole showed a more film-like cavitating flow, evident by its transparent on-axis appearance, whilst the rear holes maintained the opaque characteristic of a bubbly two-phase flow (Figure 5.42).
For the results captured, the cavitating flow remained fully developed for the duration of the injection such was the magnitude of the achievable cavitation numbers. One important observation noted was the absence of any vortex structures in the nozzle. For equivalent (fully developed) nozzle-flow conditions the appearance of cavitation strings had been shown to be intermittent but common in the larger nozzle geometry. To expand the experimental range and better explore this observation the rig was converted to operate under steady state conditions. This provided a number of advantages over the transient configuration of the rig in terms of the stresses placed on the optical components: the removal of the injector eliminated the lower pressure limit of the rigs operation created by the injector’s NOP. This allowed pressures below 300 bar to be applied to the nozzle with the added advantage of eliminating the rapid pressure transient created at the start of injection. Imaging could also be carried out without the need for synchronisation to a specific injection event.

$p_{\text{rail}}$ was reduced to 100 bar, which was sufficient to examine the transition to cavitating flow in the nozzle-holes, and allowed the rig to operate for significant periods of time, which substantially reduced the risk of mechanical failure. Figure 5.43 shows a sequence of frames from the high-speed video. Under steady-state conditions the appearance of the flow remained similar to that seen under transient conditions. The reduced risk of nozzle failure meant that the new plates (polished flat front face), which were limited in number, could be used to achieve higher quality images of the internal flow.

Film-like cavitation is seen in the entrance regions of all three holes, which breaks down to a bubbly flow after approximately one third of the hole length in the rear holes. By contrast, the flow in the front hole remains in the film-like stage for the duration of its transition through the hole, having an appearance close to that of a flow that has undergone hydraulic flip. Again it can be surmised that the turbulent flow conditions and the presence of swirl in the rear holes aided the break down of the film-stage to the more the bubbly flow that was observed. Instances of similar break-down were also observed in the front hole (Figure 5.43 (a)), albeit on a smaller scale and occurring in the final third of the nozzle length.
For a given cavitation number the flow features appeared largely static, in particular the film-like cavitation stage where extent and vapour density remained effectively constant. Vapour shedding at the transition to bubbly flow was inherently dynamic, but the flow regions where this occurred also remained constant.

The relative stability of the flow enabled the particular flow features to be imaged with greater ease. The frame rate of the high-speed imaging was increased in an attempt to obtain smooth frame-to-frame image transitions of the bubbly flow formation as it developed from the film stage. Increasing the frame rate to a maximum of 30,000 fps (limited by the laser) remained insufficient to resolve the flow motion. Figure 5.44 shows a sequence of consecutive frames focussed on the rear-right hole, captured at 30,000 fps. The individual frames exhibited no frame-to-frame continuity in the bubbly flow region, showing the residence time of the flow features to be of the order of < 33 µs.
The use of high-speed video presented no further advantages to imaging the steady-state flow, therefore the higher resolution offered by the single-shot imaging approach was used to investigate the cavitation development sequence.

\( P_{\text{rail}} \) was maintained at 100 bar and back pressure was varied to observe the development sequence. Tests were run with both increasing and decreasing back pressure trends to quantify the hysteresis effects present in the cavitating flow. Figure 5.45 depicts the six distinct stages of cavitation development that were observed during the first test run, where the back pressure \( P_2 \) was decreased, producing an increase in \( \sigma \).

Figure 5.45 Cavitation inception sequence for 110 µm multi-hole nozzle. \( P_{\text{rail}} = 100 \) bar

(a) \( \sigma = 4.65, P_2 = 17.7 \) bar
(b) \( \sigma = 5.60, P_2 = 15.0 \) bar
(c) \( \sigma = 5.85, P_2 = 14.6 \) bar
(d) \( \sigma = 6.24, P_2 = 13.8 \) bar
(e) \( \sigma = 17.20, P_2 = 5.5 \) bar
(f) \( \sigma = 5.54, P_2 = 15.3 \) bar
In contrast to the previous test matrices, this test was event driven; back pressure was altered until a change in the flow was brought about. From the outset, changes to the developed state of cavitation in each hole occurred abruptly. The flow would only exist in the six discrete states shown in Figure 5.45, despite efforts to adjust the back pressure as carefully as possible. The most significant change was shown to occur in (a), where upon reaching $P_2 = 17.7$ bar, the film-like vapour instantly reached the extent of the nozzle-hole length. Attempts to reach a state of partial development were unsuccessful. Hysteresis effects were revealed during these attempts with the fully developed state persisting until $P_2 = 19.2$ bar was reached with increasing back pressure.

Further reduction of $P_2$ brought about the condition depicted in (b). Again progressive development was not achievable, and at $P_2 = 15.0$ bar the film-like stage of cavitation instantly extended to the length shown. A further minor decrease in $P_2$, to $P_2 = 14.6$ bar (c), saw a similar abrupt development of the film stage in the left-rear hole. At this point the bubbly-flow phase had also appeared in the right-rear hole, extending off the previously established film-like cavitation. This development also occurred in the left hole at $P_2 = 13.8$ bar (d), at which, the bubbly flow in the right hole had extended further down the nozzle length. Development of the bubbly flow in both rear holes exhibited a more linear relationship with the decrease in back pressure. Full development in all three holes was reached at $P_2 = 5.5$ bar.

The distinct pressure conditions that lead to the development of the film stage and the static appearance of the vapour suggest that they are the result of flow detachment at the hole-inlet, forming a flow structure similar to that of sheet cavitation. As with sheet cavitation (Foeth et al. 2006) it can be surmised that the positive pressure gradient created at the point of reattachment induces a re-entrant jet into the vapour region, which impinges upstream on the vapour-fluid boundary and detaches a section of the sheet cavity. Subsequent reattachment of the cavity and repetition of the process leads to the formation of the bubbly flow downstream of the sheet region.
Increasing the cavitation number resulted in an expected increase in the persistence of the bubbly flow along the nozzle-hole length, but no similar increase in the extent of the film-like region. This suggested that the location of the separated-flow region was dominated by the flow geometry and viscous effects rather than the Reynolds or cavitation number of the flow, as might be expected for flow development within a nozzle hole.

The final image (f) of Figure 5.45 shows how the amount of hysteresis varied in each hole. An increase in back pressure suppressed the cavitation in each hole; however, the sequence of suppression did not match (the reverse of) that of development, with the film cavitation persisting in the rear-left hole after its suppression in the right-hole at \( P_2 = 15.3 \) bar. The window of \( P_2 \) in which this difference was present was \(< 0.1 \) bar as suppression of the left-hole occurred at \( P_2 = 15.4 \) bar, leaving only the front hole in its fully developed state. Nevertheless this showed that the left-hole exhibited higher levels of hysteresis than the right, but significantly less than the front hole.

Possible reasons for this difference were further complicated upon repeating the test some hours later. It is important to note that the exact same nozzle was used, having been left clamped-up in the rig since the previous test.

On this occasion the development sequence occurred in a different order to that seen previously. Figure 5.46 shows the three stages of development that were observed. Inception was first seen this time in the rear-right hole, again developing abruptly to the condition shown in (a) at \( P_2 = 14.5 \) bar. This was followed by inception in the left-hole at \( P_2 = 13.8 \) bar. In direct contrast to the first experiment, inception in the front-hole was last to develop at \( P_2 = 11.0 \) bar. Thus, the cavitation numbers for inception and the order in which the sequence progressed differed from the previous experiment; the largest deviation being that of the front hole.
Increasing the back pressure from the condition shown in Figure 5.46(c) revealed significant differences in the nozzle behaviour in terms of hysteresis effects. Suppression of cavitation in the left-hole occurred at $P_2 = 14.7$ bar followed by the right-hole at 16.1 bar. The front hole exhibited a far greater amount of hysteresis, with cavitation suppression finally occurring at 16.5 bar.

Table 5.3 summarises the results of the two experiments. It could be reasoned from the disparity in these results that the nozzle exhibited the characteristics of a chaotic system, whereby the evolution of the flow sensitively depended on the initial flow conditions. This property implied that two result trajectories emerging from two distinct but similar initial conditions would separate exponentially over the course of time (Boccaletti et al. 2000). In the case of the nozzle flow, minute perturbations in the upstream flow could thus lead to vastly different bulk flow structures in the sac volume. Similarities could also be drawn with the behaviour of the 300 µm multi-
hole nozzle, seen in Figures 5.31 - 5.33, and the erratic formation of cavitation strings, which also appear to be governed by changes in the flow structure of the sac volume.

Table 5.3 Comparison of back pressures $P_2$ required to achieve cavitation inception and suppression for each experiment, $P_{inj}=100$ bar

<table>
<thead>
<tr>
<th>Experiment</th>
<th>Inception (bar)</th>
<th>Suppression (bar)</th>
<th>Hysteresis (bar)</th>
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<td>1.5</td>
</tr>
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<td>15.4</td>
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</tr>
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<td>Right</td>
<td>15.0</td>
<td>15.3</td>
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</tbody>
</table>

Due to their critical dependence on the initial conditions, and due to the fact that experimental initial conditions could not be perfectly described, the intrinsic unpredictability of the nozzle-hole behaviour made modelling such changes too complex an undertaking for the scope of the project. Instead, the flow simulations produced the mean-flow characteristics of the nozzle, based on fixed initial boundary conditions.

Steady state experimentation with the 300 µm multi-hole nozzle did not reveal the same discrepancies in inception sequence or hysteresis, suggesting that the increased nozzle flow area made it less sensitive to small scale perturbations in the upstream flow.

5.5.2.1 CFD Results

The simulation work of Giannadakis and Gavaises (Giannadakis, 2005) was used to compliment the experimental discussion, and again provided useful insight into the mean-flow structure that accounted for a number of the observed flow features. Their work also provided a more insightful basis from which to discuss the absence of any form of visible vortices; a feature of the flow which proved to be common across all tests using the small nozzles.
Figure 5.48 shows the predicted velocity streamlines present in the nozzle geometry for $P_{\text{rail}} = 350$ bar, $P_2 = 40$ bar, which again displayed the counter-rotating vortices under the obstruction and corresponding rotational hole-flow. The predicted swirl levels within the rear holes, however, were significantly reduced in comparison to the 300 $\mu$m design, as illustrated in Figure 5.49.

This was considered the primary reason for the absence of visible vortices, as the physical flow structure did not appear to be present to sustain their formation.
Furthermore, it appeared from the experimental results that viscous forces dominated the small-hole entrance-flow, preventing the entrainment of vapour into any vortical structure that may have been present; which similarly implied that any vortical structures were not strong enough to overcome these viscous forces. Therefore, it could be concluded that the reduced levels of swirl in the nozzle-holes and absence of free-stream vapour prevented the formation of cavitation string structures.

Simulation of the cavitating flow deviated from the experimental results such that it failed to predict the formation of cavitation in each of the three holes under all test conditions. Two different inception criteria were applied in an attempt to replicate the experimental conditions; one based on the established method of local fluid pressure, the other on fluid stress analysis. This second, more advanced method, assumed that cavitation initiated when the highest tensile principal stress in the fluid exceeded the negative of vapour pressure. Neither method, however, was successful in predicting cavitation inception for this particular nozzle geometry.

This discrepancy was most likely the result of insufficient resolution in the model geometry, which failed to capture sites that were preferential for cavitation formation. The observed cavitation appeared to be governed by such geometric features, and also suggested that it was these distinct edge features, preventing a smooth drop in pressure in the hole-inlet, that brought about the abrupt nature of its formation. Leger and Ceccio (1998) report on previous studies that show the formation of attached cavitation to be strongly linked to surface discontinuities. Additionally, Arakeri (1973) describes two distinct types of cavitation separation; viscous laminar and nucleate. The former of these possesses a smooth glassy surface cavity at the point of separation as opposed to the macroscopic bubble formation of the latter type. The close approximation of the observed cavitation to the viscous laminar separation mechanism, leading to the formation of attached cavitation, implies that the criteria used for cavitation inception by the model do not account for this type of cavitation formation.
5.5 Multi-Hole Nozzle Flow

5.5.2.2 Summary

The cavitating flow in the 110 µm nozzle holes showed markedly different characteristics to those previously observed in the 300 µm multi-hole geometry. Attached cavitation formed from imperfections in the hole inlet radius, producing a static sheet of vapour that broke down to form a transient bubbly flow upon reattaching to the hole wall. The inconsistent hole-order in which cavitation inception and development was observed exhibited the characteristics of a chaotic system. An apparent deficiency in swirl levels within the nozzle-holes suppressed the formation of cavitation strings at all cavitation numbers tested. The deficiency in swirl, compared to the 300 µm multi-hole geometry, was correctly predicted by the model, however, the complexities of the abnormal inception characteristics meant correlation with the cavitating flow conditions was not forthcoming.

5.5.3 Sapphire Multi-Hole Nozzle Flow

The next stage of testing progressed to investigate high pressure flows up to P_{rail} = 2000 bar. This required the use of the sapphire optical plates to form an optical nozzle that could withstand the increased fluid pressures. Again, large and small-hole variations of the multi-hole design were used. As described in Chapter 3, the large-hole design contained 300 µm diameter holes, whilst manufacturing limitations restricted the small-hole design to minimum hole diameter of 160 µm. Both plates were manufactured to a thickness of 1 mm.

As with the previous nozzle-hole plates, a metrological analysis was carried out in order to generate an accurate simulation model geometry. The irregularities seen in the hole profiles of the borosilicate plates were not present in either of the sapphire plates, reducing the possible discrepancies between the model and the real nozzle geometry.
The results obtained from the 300 µm multi-hole sapphire nozzle led directly onto the work discussed in the following chapter, and as such will be addressed in the introduction to Chapter 6. Here, the 160 µm multi-hole results will be discussed and the findings referenced to those of the 110 µm borosilicate nozzle.

5.5.3.1 Small Multi-hole

High pressure testing of the 160 µm multi-hole nozzle was undertaken after that of the 300 µm design. The steady-state nozzle flow-rates reached by the 300 µm design at rail pressures above 300 bar impacted the ability to control the back pressure within the rate tube. Therefore, the rate tube was replaced by the nitrogen-filled injection chamber downstream of the nozzle assembly, which accommodated the increased flow rates and allowed the full test-pressure range to be explored.

At the high rail pressures tested (P_{rail} > 500 bar), the cavitating flow within the 160 µm nozzle-holes remained fully developed for all back pressures. Unlike the 110 µm borosilicate nozzle, the flow in the 160 µm sapphire nozzle was notably more dynamic and opaque. This factor, combined with the strong lenses effect of the small diameter holes, greatly restricted the optical data that could be obtained from the nozzle-flow. Ultimately, this drove the research to look outside of the nozzle-hole flow and more closely investigate the string cavitation formation in the sac volume, which will form the focus of the following chapter.

Figure 5.50 Plots of hole-edge coordinate strings for (a) large and small multi-hole sapphire nozzle plates. 0,0 denotes the centre of centre plate
As with the 110 µm borosilicate multi-hole nozzle, there was an absence of any form of visible vortex structures in the 160 µm nozzle geometry. The dynamic and opaque nature of the observable nozzle-hole cavitation alluded to the presence of sufficient free-stream vapour in the expected regions of vortex formation. Consequently, it could be assumed that there were insufficient levels of swirl generated by the nozzle geometry to affect the formation of cavitation strings. This also implied that it was the lack of swirl, rather than the lack of free-stream vapour that was the dominant factor in the absence of cavitation strings in the 110 µm nozzle geometry.

Fuel injection pressures were increased up to $P_{\text{rail}} = 1400$ bar before mechanical failure of the nozzle-hole plate prevented further testing from being undertaken. Figure 5.51 shows an image of the nozzle flow for $P_{\text{rail}} = 1400$ bar, $P_2 = 20$ bar, $\sigma = 69$. The appearance of the nozzle flow at this condition was identical to that observed at the starting pressure of $P_{\text{rail}} = 500$ bar, exhibiting fully developed cavitation in the hole-flow, and an absence of cavitation strings.

The simulation results of Giannadakis and Gavaises (Giannadakis, 2005) were able to successfully predict the formation of the more conventional, opaque, bubbly cavitation observed in the nozzle-holes for this geometry. Results also showed the absence of swirl in the rear holes.
5.6 BioDiesel Testing

5.5.3.2 Summary

Cavitation was successfully observed in the nozzle holes of the 160 µm sapphire multi-hole nozzle geometry up to pressure of $P_{\text{rail}} = 1400$ bar. Again, the formation of cavitation strings was suppressed by the reduced swirl levels present in the nozzle geometry at all pressures tested. This concluded the investigation into the nozzle-hole flow characteristics for the multi-hole nozzles using standard diesel fuel. The fully developed nature of the flow acted as a limit to the optical data that could be obtained from the nozzle and, as mentioned, the research progressed to study the formation of string cavitation in more detail.

5.6 BIODIESEL TESTING

Experimental testing was also undertaken to investigate the nozzle-hole cavitation characteristics of two blends of bio-diesel, and thus provide a direct comparison to the observations made during the regular diesel fuel testing.

Bio-diesel is a diesel replacement fuel derived from natural and renewable sources, such as vegetable oils and animal fats. It has properties close to that of petroleum diesel, which allow it to be used in diesel engines with little or no modification to the engine hardware. The use of bio-diesel offers environmental advantages over petroleum diesel, both in terms of its renewable production source and the reduction
in particulate and associated emissions that can be achieved through its use (Balat and Balat, 2008; Murugesan et al., 2009).

Biodiesels can be blended to any level with petroleum diesel to create a bio-diesel blend. At biodiesel concentrations of up to 5% by volume (B5), the mixture will meet the ASTM D975 diesel fuel specification and thus can be used in any application as if it were pure petroleum diesel. Blends of B6 to B20 can be used in many diesel fuel applications, achieving similar engine performance with minor or no modification to the engine hardware. For higher blend levels, up to B100, special handling and modification of equipment is usually required and therefore these higher blends can only be used in applications that have been designed or adapted for their use (NREL, 2009).

The fuel properties of biodiesel such as cetane number, heat of combustion, specific gravity, and kinematic viscosity differ from those of petroleum diesel, inherently altering both the fuel injection and combustion characteristics of a blended fuel in comparison to pure petroleum diesel (Canaki et al., 2006). Of these, the increased viscosity of biodiesels is of greatest importance as it affects the performance of the FIE (Demirbas, 2009; Balat and Balat, 2008; Ejim, 2007), which plays a significant role in governing the quality of the ensuing combustion reaction. Therefore, research into the effects of biodiesel on injection characteristics is essential if engine systems are to be optimised for their use.

Limited literature is currently available on this specific subject, with the majority of studies, including those already cited in this section, considering the performance and emission of different fuel types for a complete engine system. Ejim (2007) studied the effects of viscosity and surface tension on the atomisation of bio-diesels in a typical diesel fuel injector. The study observed an increase of 41% in sauter mean diameter (SMD) for B100 RME bio-diesels over Type 2 Petroleum Diesel; closely matching the findings of Allen and Watts (2000). This effect was significantly reduced for blended fuels, with B20 and B5 blends of RME exhibiting a 7.2% and negligible increase in SMD respectively. This is in keeping with the assertion that
blends of B5 and below retain the characteristics of petroleum diesel, and that those up to B20 may require minor recalibration of FIE operation. In emissions studies the increased oxygen content of bio-diesel appears to mitigate this increase in SMD by enabling more complete combustion in fuel-rich regions. Therefore, higher blend levels exhibit reduced PM levels, contrary to expected results of poorer atomisation (Lapuerta et al., 2007; Demirbas, 2009).

The increased viscosity, density and bulk modulus of biodiesel is often cited as a source of potential difference in the engine performance characteristics when higher level blends are used. Lapuerta et al. (2007) suggest though, that these differences are more evident for pump-line-nozzle systems, where pressure changes are more rapidly propagated through the fuel-lines, resulting in earlier needle opening and an advancing of the combustion with respect to petroleum diesel. Despite this, changes to the flow behaviour within the injector nozzle, particularly for common rail systems, have received little attention, and furthermore, literature that investigates the cavitating behaviour of bio-diesel was found to be very limited. Park et al. (2007) investigated soy-bean derived biodiesel in large-scale 2D nozzles and found negligible differences in the inception conditions and appearance of the cavitating flow, but reported similar trends in SMD when using bio-diesel versus petroleum diesel.

5.6.1 Results Analysis

This section aims to expand the current knowledge through a comparative study between biodiesel and petroleum diesel, using the same real-scale nozzle geometry at pressures up to 2000 bar. For this investigation two biodiesel blends will be studied: B20 and B100, containing 20% and 100% rapeseed-oil methyl ester (RME) biodiesel by volume respectively. The properties of the B100 made available are compared to those of the petroleum diesel in Table 5.4
5.6 BioDiesel Testing

<table>
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<tr>
<th></th>
<th>Petroleum diesel</th>
<th>B100 Biodiesel</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Density @ 20°C, (kg.m⁻³)</strong></td>
<td>829.9</td>
<td>880.0</td>
</tr>
<tr>
<td><strong>Kinematic Viscosity @ 40°C (m².s⁻¹)</strong></td>
<td>2.65 x 10⁻⁶ (est.)</td>
<td>4.0 x 10⁻⁶</td>
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The same test matrix used for the 300 µm multi-hole borosilicate nozzle was repeated for both biodiesel blends; P_{rail} increased to 500 bar in 50 bar steps, P₂ increased to 150 bar in 10 bar steps. Overall, the trends for cavitation development within the three nozzle holes remained similar to those seen previously. Both blends saw cavitation development initiate in all three holes simultaneously, then advance more rapidly in the front hole with increasing cavitation number. However, a difference in the value of σ for cavitation inception in each fuel type was observed. Figure 5.53 illustrates how, for the same fuel injection condition, P_{rail} = 500 bar, P₂ = 140 bar, σ = 1.85, the three fuels exhibited different stages of cavitation development. The amount of development does not follow the trend of blend level, as can clearly be seen by the extent of cavitation in the B20 blend compared to its isolated components. This suggests that the propensity to cavitate was not governed by a property that was unique to either of the two unblended fuels, such as viscosity, but rather it was a resultant of the blending process. The inception of cavitation is highly dependent on the presence of nuclei in the free-stream. These can take the form of small particles or micro-bubbles present in the bulk of the liquid, which reduce the tensile strength of the liquid and allow cavitation bubble growth to initiate in regions of low fluid pressure. The mixing of two separate fluids is likely to entrain air, especially in the case of “splash mixing” (NREL, 2009), increasing the dissolved gas content of blend. Therefore, it could be reasoned that if creating the blend of two fuels increased the gas content of the resulting blend, this would manifest itself as an increased propensity to cavitate with respect to unblended fuels. Without performing a gas content analysis of the three fluids this hypothesis could not be confirmed, and time scales prevented such an analysis from being undertaken. Therefore it must remain a viable explanation of the observation and become the subject of further study.
The observed difference in the value of $\sigma_i$ for the three fuels only had appreciable consequences for fuel injection conditions that generated partially developed cavitating flow. Therefore, it could not be assumed that a diesel fuel injector that typically suppressed cavitation would do so as effectively when a percentage of the fuel was replaced with bio-diesel.

For cases where the flow was known to cavitate, the characteristics of the cavitating bio-diesel flow remained identical to those of petroleum diesel. The cavitating flow in the front hole maintained a film-like appearance (shown previously in Figure 5.34); with a more bubbly opaque flow again evident in the rear holes, Figure 5.54 (d). The nozzle-flow was susceptible to the same apparent fluctuations in the sac volume pressure-field (Figure 5.54 (b) and (c)), which again produced the intermittent appearance of cavitation strings in the rear nozzle-holes and sac volume, Figure 5.54 (a), (b) and (d).
5.6 BioDiesel Testing

Similarly, tests in the smaller diameter holes, shown in Figure 5.55, produced results that differed very little from those of the petroleum diesel. The 110 µm borosilicate nozzle exhibited the same detached flow characteristics as previously observed, developing to a more dynamic structure with the increased and more uniform hole diameter profile of the 160 µm sapphire nozzle.

High pressure testing was also undertaken, successfully reaching $P_{\text{rail}} = 2000$ bar in the 300 µm sapphire multi-hole nozzle. The results obtained for both blends in this case will be discussed alongside the petroleum diesel results in the next chapter;
however, for the purposes of this discussion it is necessary to mention that again, no differences in flow characteristics were observed between the three fuel types. Testing of all three fuel types was also successfully achieved at pressures of $P_{\text{rail}} = 2000$ bar in the 160 µm sapphire nozzle using the ‘fire on demand’ approach. No further development of the flow was seen beyond that observed at the $P_{\text{rail}} = 1400$ bar case.

5.6.1.1 Summary

The investigation concluded that the differing physical properties of the three fuel types were not sufficient to bring about a significant change in the cavitating-flow characteristics. The minor difference seen in the value of $\sigma$ highlighted the potential for cavitation to form more readily in blended fuels. This effect could potentially impact injector performance under conditions where cavitation was only marginally suppressed using standard fuel, however, this would need to be assessed on a case-by-case basis. To fully quantify the reasons for this difference, a more comprehensive analysis of the fuel properties would be necessary, regarding in particular, the dissolved gas content.

5.7 CONCLUDING REMARKS

This chapter has examined the nozzle-hole cavitation phenomena for five different nozzle geometries using both standard diesel and two different blends of biodiesel. The appearance of both string and geometric cavitation was found to be highly dependent on turbulent perturbations in the sac volume, which brought about the intermittent formation of coherent vortex structures, and affected localised changes in cavitation number for each of the holes. The next chapter will now discuss the flow behaviour in the sac volume in greater detail, paying particular attention to the factors that govern the formation of string cavitation.
CHAPTER 6

SAC-VOLUME CAVITATION PHENOMENA
6.1 Introduction

In Chapter 5 the behaviour and characteristics of the cavitating flow in five different nozzle geometries was examined. Results showed that the turbulent nature of the flow in the sac volume and sensitivities to the flow geometry produced highly transient flow features in both the sac volume and the nozzle holes. As fuel injection pressures increased, the full development of cavitation in the nozzle-hole limited the optical data that could be obtained from this region of the flow. Whilst in general, the increase in the amount of cavitation followed established trends – governed by the cavitation number - the formation of cavitation strings remained highly transient and unpredictable. This chapter focuses on establishing the parameters that govern cavitation string formation, thus providing a means to control and potentially exploit their influence.

6.2 LARGE SAPPHIRE MULTI-HOLE NOZZLE

In section 5.5.3 of the previous chapter the 300 µm sapphire multi-hole nozzle geometry was introduced. The characteristics of the nozzle flow in this geometry were similar to those observed in the 300 µm borosilicate multi-hole nozzle; the geometric cavitation in the nozzle holes was opaque and dynamic, and cavitation string formation was evident in the sac volume, on axis with the rear holes. As with the 160 µm multi-hole geometry, once fully developed, no changes were perceived in the appearance of the cavitating hole-flow as rail pressures were increased.

Under steady-state flow conditions, injecting into nitrogen, the fuel rail pressure ($P_{rail}$) was increased from 500 bar in increments of 200 bar. The target pressure of 2000 bar was steadily approached in a series of attempts, with each attempt aiming to achieve a maximum pressure 200 bar greater than the last. It became evident through this approach that the sapphire optical plates possessed a finite lifetime of exposure to pressures of this magnitude. In certain cases the pressure at point-of-failure, for a particular optical plate, was less than the maximum pressure it had achieved on previous test runs. For this reason the attempt to reach 2000 bar was made using a brand new set of sapphire optical plates, and data was not captured at the intervening pressure points so as to minimise exposure time to the increased pressures.
The fuel pump speed was steadily increased to provide sufficient fuel flow-rate to maintain the desired rail pressure. The pump’s designed upper capability of 3200 rpm had to be surpassed by 50 rpm in order to achieve pressures above 1900 bar. Operating at the flow-rate limit of the rig meant that the final increase in pressure, from 1900 to 2000 bar, brought about by the increase in pump speed, was very gradual. After 10 seconds of running at >1900 bar, data was recorded at a rail pressure of $P_{\text{rail}} = 1963$ bar. Approximately 5 seconds after data capture was triggered, the nozzle failed preventing any further attempts to realise a 2000 bar flow condition.

Figure 6.1 is taken from the high-speed video captured during this test. The 300 µm multi-hole plate used on this occasion was of the ‘old design’ type, having a curved front surface. By contrast the sac plate included the flat front face and thus provided a clearer image of the sac volume compared to the nozzle-hole flow. Additionally the magnifying effect of the curved surface on the multi-hole plate caused the images of the two flow regions to become misaligned, positioning the rear holes out of frame, and also enlarging and de-focussing the front hole.

The image in Figure 6.1 is representative of the flow conditions that were observed at all rail pressures applied to the nozzle assembly. On-axis cavitation strings formed intermittently, but regularly, in the sac volume above each of the rear holes, and formation of the bridging string was suppressed.
The same observations were made during the biodiesel testing. Figure 6.2 shows the results for the B20 and B100 blends of biodiesel that were introduced in the previous chapter. Both images were taken during transient fuel injection events at \( P_{\text{rail}} = 2000 \text{ bar}, P_2 = 50 \text{ bar} \). For these tests, which were undertaken late in the test program, achieving the target pressure of 2000 bar was made easier by using transient injections and an improved design for the sapphire plates - the details of which will be discussed in Section 6.3.4.

![Figure 6.2 Frame grabs of nozzle flow conditions during a transient fuel injection events at \( P_{\text{rail}} = 2000 \text{ bar}, P_2 = 50 \text{ bar} \) for (a) B20 and (b) B100 biodiesel blends.](image)

The plate design also included the flat front face, which provided improved imaging of the cavitating hole-flow, showing the same differences in appearance between the cavitating flow of the front and rear holes as seen previously in the 300 \( \mu \text{m} \) borosilicate nozzle; namely the more transparent appearance of the front hole. Again, no differences or further development of the flow structures were observed over the tested pressure range of \( P_{\text{rail}} = 500 \) to 2000 bar.

### 6.3 THE EFFECT OF HOLE SPACING ON STRING CAVITATION FORMATION

The suppression of the bridging string was also consistent across all tested pressures. Experimental observations made using the 300 \( \mu \text{m} \) borosilicate multi-hole geometry, supported by the flow simulation results, implied that the formation of the bridging vortex was the result of interaction between the two counter-rotating vortex structures and the bulk-flow recirculation under the centre of the obstruction. The on-
axis cavitation strings confirmed the presence of the counter-rotating vortices in the sapphire geometry, and given the previously observed sensitivities of the visible flow structure to the nozzle geometry, it could be inferred that a geometric effect had limited the interaction of these flow regions.

A comparison of the sapphire and borosilicate multi-hole plate hole-profiles revealed a significant difference in the location of the rear holes for each of the plates. Figure 6.3 shows the two sets of coordinate strings plotted on the same axis.

The asymmetry of the rear holes in the borosilicate plate - caused by the misplacement of the right-hole – had resulted in a reduced distance between hole-centres along the x-axis compared to that of the sapphire plate (see Figure 6.4). This difference, with reference to the observed results, suggested that the formation of the bridging vortex critically relied on the spacing of the rear holes. For the sapphire plate this dimension appeared to exceed the limit at which interaction between the two counter-rotating vortices was able to sustain a coherent flow structure in the flow region between them.
6.3 The Effect of Hole Spacing on String Cavitation Formation

6.3.1 Multi-hole Plate Design

To investigate this further, a set of new multi-hole plates were designed that included three different hole spacing specifications. Narrow, standard and wide designs were developed to provide a range of hole layouts that would allow the influence of hole spacing to be investigated. With the flow-area of interest now reduced to the sac volume under the obstruction it was no longer necessary to observe to the nozzle hole-flow, therefore, the plates were machined from mild steel, which significantly reduced manufacturing time and expense. The optical nozzle assembly was thus adapted to only include an optical sac volume.

Figure 6.5 shows the coordinate plots of the hole layouts for each of the three designs. Each was based on the original hole layout design, containing three 300 µm diameter holes, with only the x-axis separation of the rear holes varying from plate-to-plate. The ‘wide’ design was closely based on the spacing of the sapphire plate, with each rear hole specified to a distance of 0.45 ±0.05 mm from the y-axis. The ‘standard’ and ‘narrow’ spaced holes followed a stepwise reduction from this value, specified to distances of 0.35 ±0.05 mm and 0.25 ±0.05 mm from the y-axis respectively. The ‘standard’ spacing represented the nominal distance between hole centres for the borosilicate plate. Machining in steel, however, significantly reduced the asymmetry of their layout and thus allowed the influence of that geometric feature to be investigated also. The narrow design was included to qualify the trend, and determine whether a lower spacing-distance limit existed.
The plates were laser machined in 1 mm thick mild steel by L4 Laser, of Leicester UK and the final hole spacing for each plate met the nominal design specification to within ± 0.01 mm.

Figure 6.5 Hole profiles for the three metal multi-hole plate designs used for hole spacing study. (a) narrow, (b) standard and (c) wide

### 6.3.2 Results Analysis

The first stage of the investigation continued to use the rig in its steady-state configuration. The apparent continuity of the string formation across all values of $P_{\text{rail}}$ tested, given a suitably high cavitation number, meant that reasonable conclusions could be drawn from the results obtained at lower pressures. Doing so
allowed the borosilicate optical sac plate to be used, which provided superior image clarity to that of the sapphire plate. Additionally, the lifetime of the optical plates was increased as a result of using lower pressures, allowing testing to undertaken with a minimised risk of plate failure. Testing was undertaken with fuel injection conditions of $P_{\text{rail}} = 150$ bar, $P_2 = 20$ bar. Figures 6.6 - 6.8 show representative results for each of the hole spacing configurations.

Figure 6.6 Frame grab of steady state injection for the narrow hole-spacing geometry. $P_{\text{rail}} = 150$ bar, $P_2 = 20$ bar

Figure 6.7 Frame grab of steady state injection for the standard hole-spacing geometry. $P_{\text{rail}} = 150$ bar, $P_2 = 20$ bar

Figure 6.8 Frame grab of steady state injection for the wide hole-spacing geometry. $P_{\text{rail}} = 150$ bar, $P_2 = 20$ bar

Figures 6.7 and 6.8, for the standard and wide spacing respectively, show results consistent with the previous observations made in their optical multi-hole plate equivalents. The formation of the bridging string in the narrow-spacing case, and its absence in the wide case, confirmed that the formation of the bridging vortex was a
6.3 The Effect of Hole Spacing on String Cavitation Formation

function of the hole spacing, and the direct result of interaction between the two counter-rotating vortices.

In both the narrow and wide cases, the fully developed flow structure differed little from the examples given in the above figures. Variation in vortex intensity led to analogous changes in the string diameters, but their location with respect to the hole edge (on the x-axis) remained largely constant. The appearance of the bridging string in the narrow case was intermittent but regular, and the increased vortex interaction that led to its formation seemed to affect the transient formation of the two vertical strings. This effect was made more notable by the relative stability of the strings in the wide case, where the reduced interaction between the two vortices meant a vertical cavitation string was present in both holes in almost every frame of the video.

The standard case, however, exhibited significant flow instabilities. Figure 6.9 provides examples of the variation in appearance of the cavitation strings, giving insight into the complex nature of the sac-volume flow for this particular geometry. In Figure 6.9 (a), the difference in string diameter illustrated an instantaneous disparity in vortex intensity between the vortices of each hole. In both (b) and (c), two vertical strings are visible in a single hole, confirming the presence of two exclusive vortices in the hole entrance regions. The irresolvable residence time of these features prevented their origin from being ascertained, so it remained unclear as to whether their formation was the results of, or led to the formation of, a bridging string; as seen in (d) and (e). Figure 6.9 (d) and (e) also provide further examples of the complex vortex interaction that was observed in the hole entrance regions. In these cases it was apparent that the bridging vortex existed in close proximity to, and in the case of (d) was able to wrap around, the vertical vortex, creating two vortices that were perpendicular to one another.
6.3 The Effect of Hole Spacing on String Cavitation Formation

Figure 6.9 Frame grabs from steady-state injection for the standard hole-spacing geometry showing (a) variation in string diameter, (b) twin vortices in the right hole, (c) twin vortices in the left hole, (d) bridging string ‘wrapping’ around the vertical string in the left hole (e) bridging string exhibiting close interaction with the vertical string in the left hole, and (f) complex vortex interaction and three vortices in the right hole

Figure 6.9 (f) represents the most extreme deviation from the flow structure shown in Figure 6.7. Here, three vortices were present in the hole entrance of the right hole, and the flow structure in the left hole had formed a string that possessed an abrupt change in orientation.

Similar features were captured using high-resolution single-shot imaging. The images in Figure 6.10 were taken from separate fuel injection events using the reverse mounted lens and Nd:YAG laser illumination set-up described in Chapter 4. Heightened detail of the cavitation string surface can be observed, and the reduced depth of field created by the lens system (of the order of tens of microns) suggests a variation in location of certain features along the y-axis. This is particularly evident in images (c), (d) and (f) where the bridging string and vertical string(s) appear in different focal planes. In image (d) it can seen that the bridging string appears behind the vertical string of the left hole and in-front of the vertical string in the right hole.
6.3 The Effect of Hole Spacing on String Cavitation Formation

Figure 6.10 High resolution images of cavitation string variation in the standard hole-spacing geometry.

\[ P_{\text{rail}} = 150 \text{ bar}, \ P_2 = 20 \text{ bar} \]
6.3 The Effect of Hole Spacing on String Cavitation Formation

The highly transient behaviour exhibited by this nozzle geometry, in comparison to the relative stability of the other two variations, again served to highlight the sensitivity of the flow to geometry changes. As the hole spacing increased so the flow between the holes became increasingly influenced by the incoming bulk flow rather than the recirculating flow of the counter-rotating vortices (see Figure 6.11). Similarly, as the spacing was reduced to that of the narrow-spacing case the flow in this region would become dominated by the vortex flow around each hole. Between this, the condition observed in the standard-spacing case existed, where the mutual interactions of the vortex flow were sufficiently influenced by the incoming bulk flow so as to reduce the flow stability in this region.

![Diagram](image)

Figure 6.11 Schematic of the dominant flow regimes predicted to exist in the sac volume

6.3.3 CFD Results

The dependency of the formation of the bridging string on the hole spacing was modelled by City University. Figure 6.12 contains images, provided by Giannadakis (2005), of the iso-swirl surfaces (swirl = 0.05) produced by the simulations of the three geometries. Regions of swirl were predicted between the two rear holes in all three cases. The predicted swirl in the wide-spacing case suggested a deficiency in purely using swirl as indicator for string cavitation formation.

Increasing the value of the swirl iso-surface to swirl = 0.1, as shown in Figure 6.13 (a), produced a result that was more indicative of the experimental observations. However, applying this swirl level to the other geometries reduced the apparent prominence of the swirl in this region, making the correct interpretation of the results difficult, without prior knowledge of the real flow.
6.3 The Effect of Hole Spacing on String Cavitation Formation

Figure 6.12 Iso-swirl surfaces of 0.05 for (a) narrow spacing, (b) standard spacing, and (c) wide spacing geometries. $P_{\text{in}} = 150$ bar, $P_2 = 20$ bar. Image provided by City University, London – (Giannadakis, 2005)

Figure 6.13 Iso-swirl surfaces of 0.1 for (a) wide hole-spacing and (b) standard hole-spacing geometries. $P_{\text{in}} = 150$ bar, $P_2 = 20$ bar. Image provided by City University, London – (Giannadakis, 2005)

6.3.4 High-Pressure Testing

Due to the geometry of the sapphire multi-hole plate and the pressure limitations of the borosilicate, the bridging string had so far not been studied at pressures above 500 bar. Advances made to the transient operation of the rig, which allowed it to function as a fire-on-demand system (detailed in Section 3.3.2), meant testing under transient conditions became more feasible at high pressures. Additionally, a new batch of sapphire sac plates had been manufactured by Agate Products Ltd. that had been annealed to increase their resilience at high pressures, and machined with the $c$-axis aligned with the viewing axis. This alignment eliminated the birefringent property of sapphire and ensured the best optical quality.
6.3 The Effect of Hole Spacing on String Cavitation Formation

Testing commenced at a rail pressure of $P_{\text{rail}} = 500$ bar and was advanced up to 2000 bar in steps of 500 bar. Any change in the observed flow characteristics was investigated further by taking results at intermediate points in an iterative manner. This approach ensured that data was collected economically, and minimised the running time of the test rig at high pressures.

To maximise the likelihood of string cavitation formation, the back pressure was minimised to $P_2 = 1$ bar for the majority of testing i.e. pressurised nitrogen was not applied to the injection chamber. This ensured the highest cavitation number possible was applied to each case. Higher back pressures were then subsequently applied to ascertain their effect on the trends observed for changes in $P_{\text{rail}}$.

6.3.4.1 Standard Spacing Geometry

The observations made using the sapphire multi-hole plate, and to a certain extent the borosilicate plate, indicated that the physical appearance of the fully developed cavitation strings remained independent of $P_{\text{rail}}$ – their development depending instead on the Reynolds and cavitation numbers of the flow before some threshold limit of each was reached. For the standard-spacing geometry this trend held true. Figure 6.14 shows example frame grabs from the high-speed video captured at $P_{\text{rail}} = 500$, 1000, 1500 and 2000 bar.

![Frame grabs of the cavitation string formation in the sac volume during a 2 ms transient injection event for (a) $P_{\text{rail}} = 500$ bar, (b) $P_{\text{rail}} = 1000$ bar, (c) $P_{\text{rail}} = 1500$ bar and (d) $P_{\text{rail}} = 2000$ bar. $P_2 = 1$ bar.](image)
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At each pressure, all three of the previously observed dominant string cavitation structures were visible, with no discernable change in their physical appearance as rail pressure increased. The most notable difference observed was the apparent increase in the stability of the string formation during the transient injections compared to the results seen for the steady-state flow. The erratic formation of the complex flow features, seen in Figures 6.9 and 6.10, was not present in any of the video captured at all values of $P_{rail}$. Instead, the bridging string and each vertical string appeared consistently; although still evidently affected by the turbulent nature of the flow. This is illustrated by Figure 6.15 where five consecutive frames starting 1000 µs after the start of injection (ASOI) for $P_{rail} = 500$ bar are shown.

![Figure 6.15 Consecutive frames taken from video shot at 10,000 fps of transient injection for $P_{rail} = 500$ bar, $P_2 = 1$ bar](image)

Video was captured at 10,000 fps, which still yielded abrupt changes in the frame-to-frame images. As a result, it was not possible to determine whether the observed cavitation strings possessed sufficient residence time to exist in consecutive frames,
6.3 The Effect of Hole Spacing on String Cavitation Formation

or if instead, each frame captured an instantaneous occurrence of string formation. In an attempt to achieve smooth frame-to-frame transitions the upper pulse-frequency limit of the laser, which had been increased after a service, was used to obtain video at 42,000 fps.

The consecutive frames shown in Figure 6.16 are taken from an injection for $P_{\text{rail}} = 2000$ bar, $P_2 = 1$ bar, captured at 42,000 fps

![Figure 6.16 Consecutive frames taken from video captured at 42,000 fps of a transient injection for $P_{\text{rail}} = 2000$ bar, $P_2 = 1$ bar](image)

From these images it is clear that the increased frame-rate was still not sufficient to resolve the transient behaviour of the strings, and therefore it could only be concluded that the strings were influenced by flow perturbations of the order of < 23.8 $\mu$s.

The increased frequency of their occurrence under transient fuel injection conditions, however, did allow the more successful application of the single-shot imaging technique. The following images were captured during subsequent testing to determine the influence of increased back pressure on the observed flow structure. Images of the dominant string structures, captured at a fuel injection pressure of $P_{\text{rail}} = 2050$ bar, $P_2 = 90$ bar, show their formation still persists with a significant back pressure applied to the nozzle.
The exact reason for the increased stability of the string formation during the transient injections is unclear, but it infers an increase in the dominance of the vortices in the sac volume, resulting in an overall stabilisation of the flow. This would suggest a direct dependence on the Reynolds number of the flow (used here as a measure of flow velocity). A dependence on Reynolds number had been observed during the 300 µm multi-hole borosilicate testing, where an increase in the Reynolds number resulted in an increase in the extent of the cavitation strings. The asymmetry of the hole geometry, however, appeared to impede the formation of the symmetrical flow structure that was so frequently observed in the more symmetrical geometry of the metal plates. Thus, the influence of Reynolds number on the formation of cavitation strings could be better assessed using the metal plates, where asymmetric geometry effects were minimised.

The effect of Reynolds number was isolated from the cavitation number by maintaining a constant value of $\sigma_{av}$ for varying values of $Re_{av}$. At $P_{rail} = 350$ bar the back pressure was steadily increased until a notable reduction in the formation of the...
strings was observed during the fuel injection event. This condition, at $P_2 = 82$ bar, was taken as the starting point of the study. $P_{\text{rail}}$ and $P_2$ were then increased simultaneously to maintain a constant value of $\sigma_{av} = 3.3$ and observe the effects of increasing $Re_{av}$. The results of the initial condition and the subsequent tests at $P_{\text{rail}} = 400$ bar, $P_2 = 93$ bar and $P_{\text{rail}} = 450$ bar, $P_2 = 105$ bar are shown in figures 6.19 - 6.21. Consecutive frames are taken from the same point during each injection.

Figure 6.19 Consecutive frames from fuel injection at $P_{\text{rail}} = 350$ bar, $P_2 = 82$ bar

Figure 6.20 Consecutive frames from fuel injection at $P_{\text{rail}} = 400$ bar, $P_2 = 93$ bar
6.3 The Effect of Hole Spacing on String Cavitation Formation

The increase in Reynolds number resulted in the formation of the bridging string, which was not present for injections at the initial condition shown in Figure 6.19. The previously described limit, beyond which no further development of the cavitating flow is perceived, was also observed. From these observations it can be concluded that the nozzle flow-rate is also critical to the initial formation of cavitation strings, causing the counter-rotating vortices to possess sufficient angular momentum to interact in the flow region between the holes.

This dependence on Reynolds number was investigated further during the testing of the wide-spacing geometry.

6.3.4.2 Wide Spacing Geometry

The 300 µm sapphire multi-hole geometry, which was almost identical to the wide-spacing metal plate geometry, had previously been tested up to P_{rail} = 2000 bar under steady-state conditions, and exhibited no difference in its observed string formation characteristics across all tested pressures.

The results for the metal plate under transient conditions, however, were significantly different. Figure 6.22 shows example images from the different fuel rail pressures that were tested for this geometry. The initial increases in P_{rail}, from
6.3 The Effect of Hole Spacing on String Cavitation Formation

500 bar to 1000 bar, saw negligible difference in the physical appearance of the cavitation strings. A slight increase in the frequency of string formation was noticeable at $P_{rail} = 1000$ bar, which again suggested Reynolds number effects were driving a more stable flow structure. At $P_{rail} = 1500$ bar, density fluctuations, visible as the transient refraction of light through the sac volume, became prominent in the region where the bridging string was expected to form, appearing laterally between the two vertical strings. The formation of a bridging string was not observed at this pressure however. In marked contrast to the previous results, obtained from the 300 $\mu$m sapphire multi-hole geometry, the formation of the bridging string became clearly visible at $P_{rail} = 2000$ bar, shown in Figure 6.23(d).

![Figure 6.22 Frame grabs of the cavitation string formation in the sac volume for the wide hole-spacing geometry for (a) $P_{rail} = 500$ bar, (b) $P_{rail} = 1000$ bar, (c) $P_{rail} = 1500$ bar and (d) $P_{rail} = 2000$ bar. $P_2 = 1$ bar.](image)

The formation of the bridging string occurred frequently during the 2 ms injection, suggesting that the flow conditions at $P_{rail} = 2000$ bar were somewhat in advance of those required to initiate its formation. Therefore, an intermediate test point of $P_{rail} = 1700$ bar was selected as the first iterative step in defining the rail pressure required to initiate its formation. To achieve $P_{rail} = 1700$ bar in the least time possible, it was necessary to ‘dump’ the accumulated pressure in the rail, and rebuild the pressure back up to 1700 bar. An example frame from the fuel injection events captured at $P_{rail} = 1700$ bar is shown in Figure 6.23(a). The presence of the density fluctuations was prominent, and the formation of the bridging string appeared largely suppressed. However, the presence of a bridging string was captured in a single frame in two of the five fuel injection events recorded, shown in Figure 6.23(a). The isolated occurrences of the string implied that the flow conditions were at the limit of inception. This was confirmed by tests carried out at $P_{rail} = 1600$ bar where no
bridging string formation was observed. $P_{\text{rail}}$ was then increased to 1900 bar, which yielded no further increase in the frequency of occurrence of the bridging string. An example frame from the $P_{\text{rail}} = 1900$ bar test is shown in Figure 6.23(c) and the bridging string observed is shown in Figure 6.23(d).

From these results it was concluded that the consistent formation of the bridging string was brought about by flow conditions that existed in close proximity to the pressure limit of $P_{\text{rail}} = 2000$ bar, exhibiting a similar transition to that seen in the results of the Reynolds number study described at the end of Section 6.3.4.1.

However, it was also important to consider that in addition to pressure effects, the flow conditions were subject to the significant increase in fuel temperature that was generated by the pressurisation of the fuel. To assess the impact that the increased fuel temperature, and the ensuing increased temperature of the FIE, had on the cavitating behaviour of the fuel, a further experiment was undertaken to determine if consistent formation of the bridging string could be induced at lower values of $P_{\text{rail}}$ by increasing the fuel and FIE temperature.

Initial tests were undertaken by rapidly building the rail pressure to 1700 bar and performing a series of injections under ‘cold’ conditions, where the temperature of the FIE remained close to room temperature; in particular the fuel injector and the optical nozzle. A thermocouple, fitted to the outside of the optical cassette, monitored the temperature of the nozzle assembly; for the ‘cold’ tests the
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temperature of the cassette did not rise above 30°C. These tests produced results similar to those seen previously at $P_{rail} = 1700$ bar, with isolated occurrences of the bridging string being observed. The test rig was then left to idle whilst maintaining a rail pressure of 1700 bar, which saw a steady increase in the temperature of the FIE. Once the temperature of the FIE had risen to 70°C, further injections were recorded.

No change in the frequency of the bridging string’s occurrence was observed during the higher temperature tests, implying that the formation of the string was not affected by temperature changes of this order of magnitude. Thus, the transition to more consistent formation of the bridging string remained a function of the fluid pressure.

The complex relationship between pressure, temperature, density and viscosity at high fluid pressures is investigated by Blair (2001) and Rodriguez-Anton et al. (2000). Both show the increased influence of pressure effects, over temperature effects, on fluid viscosity and density at pressures in the upper experimental range tested here. Therefore, it was possible that the formation of the bridging string was affected by the increases in these fluid properties combined with the increase in flow velocity, or angular momentum of the vortices. This somewhat obfuscates the exact influence of Reynolds number, in its true definition, and without more precise knowledge of the fuel properties at these elevated pressures and temperatures, the exact reasons for the observed change in flow structure could not be conclusively identified.

6.3.5 Angled-Hole Study

A further variation of the multi-hole plate was designed to investigate the effect of angling the two rear holes towards each other. By altering the orientation of the hole axis so that they were no longer parallel to one another, and maintaining the previous hole-spacing specifications, the experiment aimed to determine whether the angling of the holes affected the vortex formation, and produced a similar effect to reducing the hole spacing.
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Four plate designs were created to investigate the effect of this geometry change. Holes were angled at 20° and 30° to the perpendicular for both the standard and wide hole spacing to provide a range of conditions with which to establish the effect of the hole angle. Figure 6.24(a) shows a plan view of the hole layout for the wide hole-spacing with the holes angled at 30° to the perpendicular. This is clarified by Figure 6.24(b) which shows the side view of the same design.

![Figure 6.24 (a) plan view and (b) side view of the hole layout for the wide hole-spacing with 30° holes geometry. Dimensions in mm](image)

This particular design was found to be a reasonable match to the standard hole geometry of the CAT CR200 fuel injector used in the test rig, which possessed a similar angle and distance between adjacent holes (T. Langley, personal communication, February 19, 2009).

Testing was undertaken in the same manner as the previous high pressure tests; increasing $P_{\text{rail}}$ in increments of 500 bar to the target pressure of $P_{\text{rail}} = 2000$ bar. The results are presented here for each hole geometry, with a high-speed video frame sequence presented for each rail pressure.

Figures 6.25-6.28 show frame sequences for the standard hole spacing (0.7 mm between hole centres) with the rear holes angled 20° from the perpendicular. At $P_{\text{rail}} = 500$ bar, formation of all three dominant cavitation strings was observed. Formation of the bridging string appeared intermittent, but, forming less frequently than the two vertical strings. The change in orientation of the rear hole axis had an
evident effect on the vortex structure in the sac volume, where a notable difference in the appearance of the bridging string, compared to the vertical hole design, was observed. Its appearance now more closely resembled that of the bridging string formed in the narrow spacing test case, presented in Figure 6.6. This implied that the angling of the holes had the same effect as reducing the hole spacing. However, this difference was not a direct result of a change in hole-axis orientation, but rather it was a result of the hole-angle producing an elliptical hole entrance profile. This effect is illustrated in Figure 6.24, where the elliptical hole profile caused by the angle of the hole-axis, with respect to the plate surface, caused the distance between the two hole edges to be reduced; moving the distance-between-edges of the ‘standard’ spacing closer to that of the ‘narrow’ spacing. Therefore, it could be concluded that the interaction of the two counter-rotating vortices was governed by the distance between the hole edges; a result of the hole diameter and centre spacing, which had previously been established as governing factors of string formation.

Increasing $P_{rail}$ saw an increase in the frequency of the bridging string’s formation, until the appearance of the three dominant strings became consistent from frame-to-frame around $P_{rail} = 1500$ bar.

Figure 6.25 Consecutive frames for standard hole-spacing, 20° holes at $P_{rail} = 500$ bar
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Figure 6.26 Consecutive frames for standard hole-spacing, 20° holes at $P_{rail} = 1000$ bar

Figure 6.27 Consecutive frames for standard hole-spacing, 20° holes at $P_{rail} = 1500$ bar

Figure 6.28 Consecutive frames for standard hole-spacing, 20° holes at $P_{rail} = 2000$ bar
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The elliptical hole entrance had a more significant effect when the $20^\circ$ hole-angle was applied to the wide hole-spacing geometry. Figures 6.29-6.32 show how the consistent formation of all three strings was quickly established at $P_{\text{rail}} = 500$ bar; $1500$ bar less than for the vertical holes. This again highlighted the sensitivity of string formation to the geometry of the holes, and in particular their physical location within the sac volume, with hole edge spacing appearing critical to this.

Figure 6.29 Consecutive frames for wide hole-spacing, $20^\circ$ holes at $P_{\text{rail}} = 500$ bar

Figure 6.30 Consecutive frames for wide hole-spacing, $20^\circ$ holes at $P_{\text{rail}} = 1000$ bar

Figure 6.31 Consecutive frames for wide hole-spacing, $20^\circ$ holes at $P_{\text{rail}} = 1500$ bar
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The tests carried out with the 20° angled holes showed that the increase in hole angle had advanced the formation of cavitation strings in terms of rail pressure, and the change in orientation of the hole-axis, with respect to the axis of sac volume flow rotation, appeared to have little or no detrimental effect on the formation of the vortices in the sac volume. There was however an evident effect when the hole angle was increased to 30°.

Figures 6.33-6.36 show the development of the sac flow with increasing rail pressure for the standard spacing geometry with holes angled at 30° to the perpendicular. It can be seen that the flow exhibits a near constant suppression of the bridging string with only very faint examples of its formation being present; for example Figure 6.35(d) and Figure 6.36(c).
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The two vertical strings were consistently visible throughout, indicating the presence of the counter-rotating vortices. However, it can be seen that they are both angled slightly away from each other, signifying a change to the orientation of their vortex
6.3 The Effect of Hole Spacing on String Cavitation Formation

axis. This change of orientation, away from the hole axis, could be cited as a possible cause of the bridging string’s suppression, however, similar deviation can be observed in the previous 20° geometries, where no such suppression was forthcoming. This suggests a more complex reason for the suppression of the bridging string, particularly when similar suppression was not observed for the wide hole-spacing geometry.

It can be seen in the following sequence of figures that formation of the string first occurred at $P_{\text{rail}} = 500$ bar, as shown in Figure 6.37. Further increases in $P_{\text{rail}}$ saw the steady development of the string, forming consistently at $P_{\text{rail}} = 1000$ bar (Figure 6.38) and exhibiting an increase in prominence as the pressure increased through $P_{\text{rail}} = 1500$ bar (Figure 6.39) to $P_{\text{rail}} = 2000$ bar.

![Figure 6.37 Consecutive frames for wide hole-spacing, 30° holes at $P_{\text{rail}} = 500$ bar](image1)

![Figure 6.38 Consecutive frames for wide hole-spacing, 30° holes at $P_{\text{rail}} = 1000$ bar](image2)
6.3 The Effect of Hole Spacing on String Cavitation Formation

The consistent formation of the bridging string in the wide hole-spacing geometry, where it was otherwise not present in the standard geometry, implies that the relationship between hole angle and hole spacing is also of significant importance. It is clear that changes to the hole location and diameter inherently affect the vortex structure in the sac volume, and it has been observed through the deviation of the vertical strings axis that hole angle can affect this also. The differences observed between the standard and wide hole-spacing with hole angles of 30° could feasibly be another example of the sensitivity that the flow exhibits to geometry changes.

6.3.6 Summary

Observation of the bridging-string suppression in the 300 µm sapphire multi-hole geometry led to the study of hole spacing as a factor of string formation. The results of this study, using three different hole spacing geometries, showed that the formation of the bridging string was dependent on the distance between the adjacent holes. From these results it was concluded that the formation of the bridging string was governed by the interaction of the two counter-rotating vortices. Furthermore,
their interaction has been shown to be a function of both the geometry and the flow-rate of the nozzle, with higher flow rates required to achieve sufficient interaction as hole spacing was increased. This geometry dependence also included the hole diameter, an effect that was observed in the previous chapter, suppressing the formation of cavitation strings for hole diameters that do not permit sufficiently intense vortices to be established. Finally, altering the angle of the holes was also found to affect the appearance of the cavitation strings. Due to the elliptical nature of the hole inlets, the reasons for this were thought to be closely related to those of the hole-spacing, however, more complex effects were evident which saw an increased likelihood of bridging string formation with a wider hole spacing and increased hole angle.

In all cases, the changes to the flow structure have only been visible as a result of the formation, or suppression, of cavitation strings, signifying the presence of coherent vortex structures. Outside of these flow regions, the current optical analysis techniques failed to provide additional flow information that may have aided the explanation of the observed flow conditions. This chapter will now move on to discuss the application of a new optical technique, which aimed to obtain quantitative flow velocity data from the sac flow.

6.4 PARTICLE IMAGING FOR QUANTITATIVE FLOW DATA

Particle image velocimetry (PIV) is a non-invasive whole-field velocity measurement technique that allows quantitative information on the spatial structure of the velocity field to be obtained. In its simplest form, PIV measures two-dimensional velocity vectors in a two-dimensional plane at one instant in time (Adrian, 1997). More advanced adaptations of the technique allow for three-dimensional velocity and spatial measurement, and increased temporal resolution (Adrian, 2005).

The general principle of PIV is to illuminate tracer particles in the flow-field of interest and obtain two images of the flow field with a known time separation. The motion of the tracer particles between the two images determines the displacement field of the flow region, and by dividing this by the known time separation of the
images, the velocity field is obtained (Bolinder, 1999). In PIV the displacement field is determined as average displacements within so-called interrogation regions of the image plane, which are typically 32x32 pixels in size. Corresponding interrogation regions from the first and second images are cross-correlated to obtain the average displacement for that flow region; for a given interrogation region, the location of the highest correlation peak in the correlation plane corresponds to the most likely average particle displacement. Repeating this process for all interrogation regions yields the displacements across the entire imaged flow region.

Therefore, the use of PIV as a technique for obtaining quantitative flow velocity data from the optical nozzle was explored. Certain established modifications to the traditional technique were required to obtain images from the small-scale, restrictive flow volume of the optical nozzle. In most cases illumination of the particles is achieved through use of a short pulsed laser beam, formed into a sheet by a cylindrical lens to create the two dimensional measurement plane. Tracer particles are selected such that their size is small enough to follow the motion of the flow, but large enough to scatter sufficient light from the incident laser beam to be detectible by the imaging optics. Thus, the particle imaging relies on the Mie scattering of light from the particles. Application of this technique failed to be effective in the optical nozzle due to the interference of additional scattered laser light from the internal surfaces of the nozzle geometry. To eliminate this interference, the use of fluorescent particles and a long pass filter was employed.

Fluorescent tracer particles of 5 µm diameter with a peak excitation wavelength of 542 nm and a Stokes shift of 70 nm were used. When excited by the 532 nm wavelength light of a Nd:YAG laser, the particles fluoresced red, emitting light at 602 nm. A 550 nm longpass filter was then placed in front of the imaging optics to allow imaging of the particles without interference from the reflected laser light.
6.4 Particle Imaging for Quantitative Flow Data

6.4.1 Experimental Setup – Low-Pressure Rig

Introducing fluorescent tracer particles to the high pressure diesel flow of the main rig required adaptation of the rig design, and had the potential to affect future results gathering. For this reason, a separate low-pressure rig was designed and built to develop and prove the technique without risking the hardware of the main rig.

To ensure the practical transfer of the developed experimental approach, the new rig was designed to accommodate the optical cassette and nozzle assembly, along with use of the lens system employed on the high pressure rig. Figure 6.41 shows the new cassette holder, which clamped the assembly in position by way of threaded bars located at each corner of a bottom platform. This provided sufficient clamping force for the reduced fuel delivery pressure.

![Image of the cassette holder/nozzle assembly clamp for the low pressure rig. The moveable bottom platform provides clamping force by way of tightening a nut on the underside of the platform on each of the four threaded bars.](image)

Fuel pressure was limited to 6 bar by way of a 1 litre accumulator that worked off a compressed air line supply. Fuel delivery to the optical nozzle was controllable via solenoid valve that could provide transient injections of 10 ms duration. Diesel fuel was pre-seeded with fluorescent tracer particles and fed into the accumulator via a
pre-fill chamber, shown in Figure 6.42. This was then closed off before the compressed air supply valve was opened. Control of the injection was then governed by the solenoid valve, which was activated manually and synchronised to the PIV hardware. Injected fuel passed through the nozzle and into a collection vessel, where the particle seeding density could be altered if necessary, before the fuel was reintroduced upon exhaustion of the accumulator fuel supply.

The lens system, consisting of extension rings and the Nikon 200 mm Micro-Nikkor lens, was attached to a single 1000 x 1016 pixel 8-bit TSI PIVCAM 10-30, synchronised to a double-pulsed New Wave Nd:YAG Solo III laser. Processing and control of the PIV imaging was undertaken using TSI Insight PIV software. The 25 mm diameter 550 nm longpass filter was positioned close the nozzle assembly to minimise the amount of scattered laser light that reached the lens.

Initial attempts to achieve particle images followed the traditional approach to PIV, whereby a laser sheet was created to illuminate the flow region of interest. To achieve this, the cassette was re-machined to allow illumination at 90° to the camera view. Figure 6.43 shows the modified cassette with slot cut-out on the right side of the cassette body, in line with the mating surfaces of the sac and multi-hole plates.
Care was taken to machine the slot as wide as possible without compromising the lateral location that the internal bore of the cassette provided to the nozzle plates.

The level of fluorescent light emitted by the particles using a side illuminating light sheet was found to fall considerably below that required to achieve good quality particle images. Efforts to increase the levels of emitted light first looked to increase the energy density of the light sheet in the optical nozzle. Over a series of iterations the laser output was maximised to approximately 100 mJ.pulse\(^{-1}\) and the sheet was more tightly focussed into the sac volume. Neither of these adjustments produced satisfactory results, and furthermore, as shown in Figure 6.44, the increase in laser energy began to ablate the optical plates, rendering them unusable.

Therefore a new approach to illumination was taken, which drew upon the techniques of epifluorescent micro PIV (\(\mu\)-PIV) where the particles are broadly...
illuminated on-axis with the camera, and the reduced focal depth of the image is used to create the sheet effect (Meinhart et al., 1999).

To achieve this, the sheet optics were removed from the laser and the unfocussed beam was directed onto the front face of the nozzle assembly. Illumination on-axis with the camera was not possible due to the required proximity of the 550 nm longpass filter to the nozzle assembly. However, it was found that satisfactory particle images could be produced by directing the beam from an off-axis angle, as illustrated in Figure 6.45.

6.4.2 Low-Pressure Test Results

Limitations to the area that could be imaged still existed however, and efforts to increase the light emitted from particles located under the obstruction, in the flow region of greatest interest, proved unsuccessful. Therefore, particle imaging of the flow from this viewing angle was limited to the forward region of the sac volume, including the entrance to the front hole.

Figure 6.46 shows an image of the flow area (a) along with a single frame of the subsequent particle image (b) and the corresponding velocity profile (c) produced
through cross-correlation of the particle image pair. For a fuel injection pressure of 6 bar, a laser pulse separation ($\Delta t$) of 5 $\mu$s was used to best capture the higher velocity regions of the flow.

The acceleration of the flow into the front hole was captured and velocity data of the incoming sac flow is produced, however, changes to the experimental setup were required if the flow region under the obstruction were to be analysed. This was achieved by rotating the cassette through 90° and viewing the sac volume from the side, through the newly cut slot. The images in Figure 6.47 show a raw particle image and the corresponding velocity profile of this new view, which provided better illumination of the particles under the obstruction. The side profile of the obstruction can be seen on the right side of the image and the focus plane captures the flow entering the front hole and the flow acceleration under the front edge of the obstruction.
6.4 Particle Imaging for Quantitative Flow Data

Figure 6.47 (a) Particle image of the side view – the side profile of the obstruction can be seen in the right of the image, and (b) calculate velocity vector field for a 6 bar injection through the centre line of the geometry.

It was expected that the rotating flow that led to the formation of the bridging string might be discernible in this region; however, the magnification levels and the particle seeding density prevented any rotational flow from being resolved. Increases in particle seeding density only served to increase the amount of out of focus particles present, creating unacceptable levels of noise in the image.

Despite the limitations of the technique that were observed, completion of standard testing on the high pressure rig had allowed the required changes to be made to enable seeding of the high-pressure diesel flow with fluorescent particles. Therefore further development of the technique was carried on the high-pressure rig.

6.4.3 Experimental Setup – High-Pressure Rig

To seed the high pressure flow, the accumulator used on the low pressure rig was fitted in-series to the high pressure fuel line, upstream of the injector. Flexible high pressure hosing was used to simplify the attachment of the accumulator to the rig’s safety shielding framework, accommodating the movement of the injector during clamping/unclamping.
The accumulator was then filled with seeded fuel, the same way as previously, before the high-pressure hose was attached to its inlet, effectively filling the fuel line upstream of the injector with particle-seeded fuel. Additionally, the fuel return pipe was disconnected from the base outlet of the injection chamber, as was the injector’s low-pressure fuel return, allowing fuel from both to drain directly into collection vessels. This approach isolated the seeded fuel from the main rig hardware, significantly reduced the volume of fuel that required seeding, and allowed the vast majority of particles to be recovered upon exhaustion of the accumulator supply volume.

The introduction of the smaller accumulator restricted the maximum rail pressure to $P_{\text{rail}} = 600$ bar. However, using borosilicate plates had proved to provide superior particle image quality over the use of sapphire plates, and through their use the limit of 600 bar was more than sufficient to cover their working pressure range.

To accommodate the higher fluid velocities that resulted from the increased fluid pressures, the approach to PIV imaging also had to be adapted. Cross-correlation PIV required two separate images of the flow to be taken, separated by a time, $\Delta t$. A two order of magnitude increase in fluid pressure, over the $P_{\text{rail}} = 5$ bar low-pressure test
case, could be expected to produce fluid velocities an order of magnitude greater than those previously observed. Thus, a ∆t of <<1 µs was required to achieve similar particle displacement between image pairs. Such a value of ∆t was less than the time delay between successive frames of the PIVCAM, and thus it was necessary to switch to the auto-correlation imaging approach. This approach used the two laser pulses to produce a double exposure image on a single frame, with each interrogation region of the frame being correlated with itself to yield the displacement field. ∆t was thus limited by the pulse separation of the lasers, which was effectively limited by the laser pulse duration of 8-10 ns, which was more than sufficient for the requirements of this study.

For auto-correlation the use of a twin frame camera such as the PIVCAM 10-30 was no longer necessary, with any camera being able to capture the double-pulsed exposure to a single frame. This allowed the increased spatial resolution offered by the Olympus SLR camera to be utilised. Integration of this camera into the current PIV setup was not possible due to its incompatibility with the TSI triggering and synchronisation hardware. Instead, a manual system was developed around a Stanford Research Systems DG535 four channel digital pulse/delay generator. This was capable of receiving the trigger signal from the camera’s hot shoe adaptor and outputting the signals necessary to fire the injector and trigger the laser pulses. Each channel could be independently delayed - to a 5 ps resolution if necessary - allowing complete control of the imaging and fuel injection synchronisation.

Furthermore, the compact reverse-mounted lens system was fitted to the camera body (as detailed in Section 4.2). This placed the rear element group of the lens closest to the objective, where its smaller diameter was now able to fully accommodate the 550 nm longpass filter, ensuring all scattered laser light was filtered before entering the camera. Additionally, this lens setup exhibited a shallower depth of field to the Nikon lens system, meaning the effective sheet width was reduced and a more precise flow plane could be defined.
Figure 6.49 shows a schematic of the experimental setup for the high-pressure rig. The technique of illuminating the optical nozzle by the unfocussed laser beam was carried over from the work on the low pressure rig, as was the new favourable orientation of the nozzle assembly with respect to the camera. This was achieved by rotating the entire ‘injector pod’ through 90° and thus maintaining the 0.3 mm offset required to align the injector outlet with the D-shaped sac inlet.

Fuel injection pressures were set to $P_{\text{rail}} = 500$ bar, with the ‘fire on demand’ fuel injection control allowing higher pressures to be applied to the borosilicate with reduced component fatigue. For all cases, due to the modification to the injection chamber, back pressure was limited to atmosphere. Additionally the metal multi-hole plate with standard vertical hole spacing was used to provide the greatest probability of generating stable vortices in the sac volume.

A laser pulse separation of $\Delta t = 250$ ns was found to produce adequate separation of the particles between exposures. Figure 6.50 shows an image captured for $P_{\text{rail}} = 500$ bar.
Particle image pairs can be seen in the top left of the image, flowing down the front face of the obstruction, before flowing under the obstruction into the region above the right rear hole. Several issues were noted with the application of this technique to the high pressure rig. Firstly, the seeding density achievable was severely limited by the CR200 fuel injector, which was found to clog with fluorescent particles once the seeding density reached a certain level. The sparse seeding density made it impractical to perform an auto-correlation analysis of the image, instead requiring a technique better suited to low particle densities such as particle tracking velocimetry (PTV) (Ohmi and Li, 2000). For the seeding densities permitted by the injector it was even possible to perform a manual analysis to yield the displacement of each particle pair.

A further difficulty arose in determining the exact location of the focal plane relative to the internal nozzle geometry. Sparse seeding prevented the outline of the obstruction from being adequately defined, and thus it could not be guaranteed which geometric plane the in-focus particles were located on. To overcome this particular issue the camera was first focussed using back illumination, before the laser was realigned to provide front illumination. Back illumination was provided by a fluorescein dye cell, pumped by the unfocussed Nd:YAG laser beam. Figure 6.51 shows a schematic of the back illumination setup.
The fluorescent emission bandwidth of fluorescein, when pumped by the Nd:YAG laser, produced sufficient levels of light at wavelengths greater than 550 nm, which allowed for the longpass filter to remain attached to the lens during set-up. This feature allowed the camera to remain untouched once the required focus had been set.

Figure 6.52 shows an image of the flow where the focal plane has been set between the two rear holes, capturing a side view of the bridging string for the first time.

Figure 6.52 Image captured using back illumination using the fluorescein dye cell. The focal plane is located between the two rear holes showing a side view of the bridging string. Shadow images of particle pairs are also visible in the flow $\Delta t = 250$ ns, $P_{\text{rail}} = 500$ bar.
For the image shown in Figure 6.53, the focal plane has been brought forward to image to the rear right hole of the multi-hole geometry.

Figure 6.53 Image captured using back illumination using the fluorescein dye cell. The focal plane is located in line with the rear right hole, bringing the vertical string formed at this instant in to focus. Shadow images of particle pairs are again visible in the flow. $\Delta t = 250\,\text{ns}$, $P_{\text{init}} = 500\,\text{bar}$.

In both Figures, shadow images of particle pairs can be seen in the sac flow, with the particles that appeared in focus being located in the same geometric plane as the visible cavitation string, or vortex core. These images thus highlighted the possibility of combining flow visualisation with particle tracking, allowing the flow conditions at the instantaneous time of measurement to be more accurately described. By using particle images alone, one could not determine the presence or location of a cavitation string; therefore, any difference in the flow conditions that may exist during their residence could not be distinguished from the non-cavitating conditions.

By maintaining a low level of back lighting and reintroducing the front illumination, the contrast between the particle images and the flow environment was enhanced. This approach is similar to one demonstrated by Lindken and Merzkirch (2002) to observe multi-phase flows, but the technique here is better suited to the micro-scale high velocity flow being studied. Figure 6.54 shows a schematic of the optical setup used to achieve this ‘hybrid’ optical analysis.
An 80:20 beam splitter directed 20% of the laser energy away from the fluorescein dye cell, providing front illumination for the fluorescent particles. Before pumping the dye cell, the remaining 80% of the laser energy was further attenuated by a neutral density filter, ensuring that the particle images remained well contrasted to the back lighting. Increased confidence in the rig and component capabilities meant that the following images were captured from fuel injection events with a rail pressure of $P_{\text{rail}} = 600$ bar.

Figure 6.55 shows an example image achieved using the ‘hybrid’ technique. Of particular note in this image is the highlighted particle image pair positioned in front of the cavitation string. As a result of the shallow focal depth it can be said that this particle is located in close proximity to the cavitation string, and therefore its velocity is close to the rotational velocity of the vortex at this point. Based on a manual calculation of this particular displacement, a velocity of $109 \text{ m.s}^{-1}$ is measured.
6.4 Particle Imaging for Quantitative Flow Data

A similar result can be seen in Figure 6.56, with the nozzle returned to its original orientation, where the rotational velocity of the bridging vortex was measured to be 65 m.s$^{-1}$.

These velocities relied on the close approximation of the particles’ motion to the fluid motion. In the turbulent flow conditions observed here, Stoke’s drag law cannot be applied, therefore a discrepancy between the particle and fluid motion can always be argued. The use of smaller particles would reduce the discrepancy, and in this case may alleviate injector clogging to allow greater seeding density to be employed. A significant reduction in particle diameter could also be accommodated by the Olympus SLR camera and is therefore considered a high priority for any further work.
6.4.4 Summary

The technique outlined here is presented to its current stage of development, and evidently requires further work for it to provide more substantial quantitative data. However, the concept of simultaneous back and front lighting of the flow, using light filtering and fluorescent substances to achieve quantitative flow data, has been proven in a microscopic flow geometry at fuel injection pressures up to $P_{rail} = 600$ bar. The use of smaller diameter particles has been cited as a future development that could lead to improved data gathering capabilities, and as such, is considered here as a priority for further work.

6.5 CONCLUDING REMARKS

This chapter has examined the formation of string cavitation in the sac volume of the multi-hole nozzle geometry. Formation was found to be dependent on the hole geometry and nozzle flow rate, with both of these factors affecting the interaction of the two counter-rotating vortices in the sac volume; which governed the formation of the bridging string. To aid the explanation of certain observations, additional flow information was required beyond that offered by the occurrence of cavitation strings. Therefore, a technique for obtaining quantitative flow data has been developed and preliminary results obtained. Chapter 7 will now summarise the main conclusions that have been drawn from this work and discusses potential future work related to this research.
CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS FOR
FURTHER WORK
7.1 RESEARCH SUMMARY

The design and development of the patented high-pressure optical fuel injection test facility has been detailed. A range of experimental and analysis techniques has been employed to study the sac and nozzle flow characteristics in true-scale optical fuel injector geometries. New results have been obtained at, and beyond, the design limits of both the test rig and the diagnostic hardware, establishing a proven test facility that offers considerable flexibility for future study.

The research presented in this thesis has visualised cavitating flow phenomena at fuel injection pressures up to 2050 bar. String cavitation has been observed in the sac volume and nozzle-holes of optical multi-hole nozzle systems, and it was shown that its formation exhibited sensitivity to the nozzle geometry, flow-rate and cavitation number. Nozzle flow data has been presented for both transient and steady-state fuel injection conditions, injecting into liquid and gaseous back-pressures up to 150 bar. The data for twelve different nozzle-flow geometries has been presented using high-speed video imaging, high resolution single-shot photography and flow-pressure logging to establish the nozzle flow characteristics.

This research has provided a unique insight into the behaviour of true-scale fuel injector flows at high pressure, producing experimental data across the working pressure range of current diesel fuel injection systems. Data gathered during this investigation has been used to validate the advanced cavitation model sub-routines developed by City University London, which now form an integral part of the wider range of computer aided engineering (CAE)-based engine development tools at Caterpillar Inc.

7.2 CONCLUSIONS

The empirical findings of this study have led to an improved understanding of the geometric and flow-based factors that govern the formation of cavitation phenomena in true-scale fuel injector flows. The major conclusions that can be drawn from this work are:
1. Results obtained at fuel injection pressures up to 2050 bar have shown for the first time that cavitation strings observed at realistic fuel injection pressures exhibit the same physical characteristics as those observed at lower pressures. Sac and nozzle-flow features observed at fuel injection pressures of $P_{\text{rail}} = 150$ bar, $P_2 = 20$ bar were also observed at $P_{\text{rail}} = 2050$ bar, $P_2 = 90$ bar.

2. The formation of cavitation strings was found to be mutually dependent on nozzle geometry and flow rate. Hole diameter was of most significant importance, with hole diameters $\leq 160$ $\mu$m having been shown to suppress all cavitation string formation in this nozzle geometry.

3. The formation of a bridging string between nozzle-holes was the result of interaction between the two counter-rotating sac-volume vortices, which formed over the hole-inlets. As a result, hole-to-hole interaction was dependent on the hole-spacing between adjacent holes, and the vortex intensity being sufficient to induce interaction across that spacing. Spacing $\geq 0.9$ mm required fuel injection pressures $P_{\text{rail}} > 1700$ bar to induce interaction. The angle between holes also affected this interaction, having a varying effect dependent on hole spacing.

4. The formation of cavitation strings in a particular geometry became independent of fuel injection and back pressure once a threshold pressure drop across the nozzle had been reached. For the standard-spacing multi-hole geometry this threshold was $P_{\text{rail}} - P_2 = 307$ bar. Uncertainty over the fuel’s viscous properties at high pressure prevented the conclusive influence of Reynolds number from being established.

5. Cavitation strings are formed by the entrainment of free-stream vapour into the low pressure core of a coherent vortex, with their physical prominence being dependent on the amount of available free stream vapour. Therefore, when formed on-axis with a nozzle-hole, cavitation strings are evidence of rotating flow in the nozzle-hole, which may also affect changes in the spray angle due to increased tangential flow velocity.

6. Plotting the instantaneous cavitation number for a transient fuel injection event showed that the extent of geometric cavitation development was analogous to the transient fluid-pressure field in the sac volume. Consistent
values of cavitation number, \( \sigma_i = 0.97 \) and \( \sigma_f = 1.35 \) were shown to exist for fuel injections with different values of \( \sigma_{av} \) for the single hole geometry. Similar trends were observed for the multi-hole geometries. Therefore, \( \sigma_{av} \) does not account for the presence of two discrete non-cavitating and cavitating flow regimes during an injection. Thus, when using transient injections to characterise a nozzle’s cavitating behaviour, the transient characteristics of the injection event must be known to determine the temporal locations of \( \sigma_i \) and \( \sigma_f \).

7. Localised pressure changes, caused by turbulent perturbations in the sac volume of the multi-hole geometry, independently affected the geometric and string cavitation formation in each of the holes. The magnitude of this effect was highly sensitive to the hole position, and the interdependence that the hole position shared with the structure of the upstream sac flow.

8. The differing physical properties of two blends of bio-diesel were not sufficient to bring about a significant change in the cavitating-flow characteristics. The cavitation inception number \( \sigma_i \) was measured as being 0.35 less for B20 than for petroleum diesel, highlighting the potential for cavitation to form more readily in blended fuels. This effect could potentially impact injector performance under conditions where cavitation was only marginally suppressed using standard fuel; however, this would need to be assessed on a case-by-case basis.

9. Quantitative flow velocity data has been obtained from a microscopic flow geometry at pressures up to \( P_{\text{rail}} = 600 \) bar. Particle velocities of 109 and 65 m.s\(^{-1} \) were measured in close proximity to a vertical and bridging string respectively. The developed imaging technique allowed this data to be achieved whilst simultaneously capturing the visible flow characteristics at the time of measurement.

### 7.3 Recommendations for Further Work

The work presented here has provided new and interesting information on the nature of cavitation phenomena in high-pressure, real-scale fuel injector flows. During the
course of the investigation several areas worthy of further work have been identified; these are:

1. **Linking the observed internal nozzle-flow features to the ensuing spray characteristics.** From both an experimental and modelling point of view, the correlation between cavitation phenomena and nozzle spray behaviour is of great importance. The current test rig has the capability to accommodate a spray chamber under the optical nozzle assembly, enabling a transition to this area of study that requires limited modification to the current design. Simultaneous imaging of the nozzle flow and the spray would be supported by additional optical diagnostic techniques such as Phase Doppler Anemometry (PDA).

2. **Development of the ‘hybrid’ imaging technique to obtain substantial quantitative data from the nozzle flow.** The proof of concept has been presented, achieving some velocity data up to fuel injection pressures of $P_{\text{rail}} = 600$ bar. The use of smaller fluorescent particles and possible modification of the injector are seen as primary steps in the advancement of the technique. Additionally replacement of the accumulator with a standard fuel rail could allow fuel injection pressures up to $P_{\text{rail}} = 2000$ bar to be studied.

3. **New sac geometries.** Future work could take advantage of the rig’s flexibility to study a range of other sac geometries to simulate the effects of needle lift and seat angle for example.
REFERENCES


ADRIAN, R. J. 2005 Twenty years of particle image velocimetry. Exp. Fluids 39, 159-169


ARAKERI, V. H. 1975 Viscous effects on the position of cavitation separation from smooth bodies. J. Fluid Mech. 68, 779-799


BAE, C. et al. 2002 Effect of nozzle geometry on the common-rail diesel spray. SAE Paper 2002-02-1625

BAIR, S. 2001 The Variation of viscosity with temperature and pressure for various real lubricants. ASME J. Tribol. 123, 433–436


BAUMGARTEN, C. 2006 Mixture formation in internal combustion engines, Springer ISSN-1860-4846


BOSCH, W. 1966 The fuel rate indicator: a new instrument for display of the characteristic of individual injection. SAE paper 660749


CHANG N. A. et al. 2007 Cavitation visualization of vorticity bridging during the merger of co-rotating line vortices. Phys. Fluids 19, 058106


EUROPEAN COMMISSION COMMUNICATION 2007 COM(2007)19 Results of the review of the community strategy to reduce CO₂ emissions from passenger cars and light-commercial vehicles, Section 2.2.4

EUROPEAN COMMISSION COMMUNICATION, 2009 COM(2009)713 Monitoring the CO₂ emissions from new passenger cars in the EU: data for the year 2008


GAVAISES, M. *et al.* 2007 Link between cavitation development and erosion damage in diesel fuel injector nozzles. SAE Paper 2007-01-0246


GONEY, K. H. & CORRADINI, M. L. 2000 Isolated effects of ambient pressure, nozzle cavitation and hole inlet geometry on diesel injection spray characteristics. SAE Paper 2000-01-2043


References


LI, H. 1999 An Experimental Investigation of High Pressure Cavitating Atomizers, Ph.D. thesis, School of Aeronautics and Astronautics, Purdue University, West Lafayette, IN.


SALIBA, R. & CHAMPOUSIN, J.C., Influence of the Nozzle Geometry on the Cavitation and on the Spray Development in Diesel Injection, *Proc. ILASS–Europe*, Orleans, France


SOTERIOU, C., ANDREWS, R. J. & SMITH, M. 1995 Direct injection diesel sprays and the effect of cavitation and hydraulic flip on atomization. SAE Paper 950080


TEARNEY, G. J. *et al*. 1995 Determination of the refractive index of highly scattering human tissue by optical coherence tomography. Optics Letters **20**(21), 2258-2260

APPENDIX A

MITIGATION OF OPTICAL PLATE BREAKAGE

During the initial stages of testing, failure of the optical was common, often occurring at relatively low fuel injection pressures e.g. 300 bar. Additionally failures occurred during clamp-up and were evident when focussing the camera before testing commenced. Failure was suspected to be caused by imperfections in the obstruction plate so detailed metrological analysis of its surface profile was carried out using advanced measurement techniques.

Surface profiling revealed no distinct variations in the surface flatness. Imaging of the surface close the obstruction protrusion revealed a radius of 0.3 mm between the mating surface to the sac plate and the obstruction protrusion.

Figure A-2 measured radius of 0.3 mm between the obstruction plate mating surface and the obstruction protrusion
Initial attempts to remove this feature were unsuccessful due to its small diameter and close proximity to the obstruction protrusion. To mitigate its presence and bevel was machined into the upper edge of the sac volume to enable it to accommodate the radius.

![Figure A-3 Schematic of the bevel machined into the upper edge of the sac volume, accommodating the 0.3 mm radius of the obstruction plate.](image)

Machining the bevel left a rough inner surface, which created a large number of stress raiser sites. This had the effect of reducing the maximum operating pressure of the sac plates and breakages continued to occur.

![Figure A-4 images of nozzle failure with bevel machined into sac volume rim](image)

Subsequent re-grinding of the obstruction plate managed to reduce the radius sufficiently and also improved the mating surface between the obstruction plate and the sac volume.
Breakages did continue to occur, although these were more closely correlated to fatigue from high-pressure testing. Ensuring a clean mating surface between all plates was essential to maximising the operating lifetime of the plates. Borosilicate plates were clamped up with a force of 3-6 kN. This was sufficient to minimise leakage between the plates at pressures up to 600 bar. 10 mm diameter sapphire plates were clamped with a force of 12 kN; sufficient for testing in excess of 2000 bar.

Additional analysis of the optical plate surfaces revealed very minor periodic deformation of the plate surface. This was not considered to be a risk to the structural integrity of the nozzle assembly under pressure.

Figure A-5 Surface profiling of a sapphire sac plate. Deformation exaggerated.
APPENDIX B

TWIN-VIEW IMAGING

A novel imaging system was developed to allow simultaneous imaging of the front and side views of the sac volume and nozzle holes. Doing so enabled the 3D location of the strings to be established. A series of mirrored prisms were fixed to micrometer stages, which steered light from each view along identical path lengths to be focussed on a single camera chip.

![Figure B-1 schematic of the optical path layout.](image)

Prism pairs to the left and right could be independently adjusted to achieve a sharp focus of each view. Further adjustment was provided by the final prism stage which provided simultaneous focus control of each view.

The optical cassette was adapted to enable the side view of the sac/hole flow. Holes were machined through both sides of the cassette to allow imaging and back illumination. The holes were machined at the precise location that aligned with the mating surface between the sac plate and nozzle-hole plate, thus allowing simultaneous imaging of both flow regions.
Imaging using the set-up was successfully achieved, however results were limited in their use for this particular geometry. The location of the strings in the z-plane was established and their ‘in-line’ orientation meant that the side view offered limited benefit for the difficulty in aligning the images.
Figure B-4 Image of the two views side-by-side taken with the SLR camera, with side view to the left under no-flow conditions.

Figure B-5 Frame grab of the system being used under steady-state flow conditions. The side view is superimposed over the front view. Left and front nozzle-holes are cavitating and string is seen forming in both views above the left hole.

This apparatus was not used extensively during testing due to the limited benefit it provided by simultaneously viewing the images. Sufficient flow information was provided by the front view which allowed for greater magnification levels and increased flow detail.