Particle image velocimetry in gas turbine combustor flow fields

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PARTICLE IMAGE VELOCIMETRY
IN
GAS TURINE COMBUSTOR FLOW FIELDS

David Hollis

Submitted in partial fulfilment of the requirements for the award of
Doctor of Philosophy of Loughborough University
Department of Aeronautical and Automotive Engineering
June 2004

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Abstract

Current and future legislation demands ever decreasing levels of pollution from gas turbine engines, and with combustor performance playing a critical role in resultant emissions, a need exists to develop a greater appreciation of the fundamental causes of unsteadiness. Particle Image Velocimetry (PIV) provides a platform to enable such investigations. This thesis presents the development of PIV measurement methodologies for highly turbulent flows. An appraisal of these techniques applied to gas turbine combustors is then given, finally allowing a description of the increased understanding of the underlying fluid dynamic processes within combustors to be provided.

Through the development of best practice optimisation procedures and correction techniques for the effects of sub-grid filtering, high quality PIV data has been obtained. Time average statistical data at high spatial resolution has been collected and presented for generic and actual combustor geometry providing detailed validation of the turbulence correction methods developed, validation data for computational studies, and increased understanding of flow mechanisms. These data include information not previously available such as turbulent length scales.

Methodologies developed for the analysis of instantaneous PIV data have also allowed the identification of transient flow structures not seen previously because they are invisible in the time average. Application of a new ‘PDF conditioning’ technique has aided the explanation of calculated correlation functions: for example, bimodal primary zone recirculation behaviour and jet misalignments were explained using these techniques. Decomposition of the velocity fields has also identified structures present such as jet shear layer vortices, and through-port swirling motion. All of these phenomena are potentially degrading to combustor performance and may result in flame instability, incomplete combustion, increased noise and increased emissions.

Keywords: Particle Image Velocimetry, Gas Turbine Engine, Combustor Aerodynamics, Sub-Grid Filtering, Conditional Averaging, Flow Instabilities, Vortices.
Acknowledgements

Thanks go to all members, past and present, of the Rolls-Royce University Technology Centre in the Department of Aeronautical and Automotive Engineering at Loughborough University. Special acknowledgement goes to the members of the tea school for the technical, and not so technical discussions – both of which never failing to initiate some sort of thought process.

Special thanks go to Adrian Spencer for his continual support, advice, and direction. Thanks to Jon Carrotte and Jim McGuirk for their invaluable suggestions and encouragement. And to the technicians, without whom, experimental projects like this could not work. I would also like to thank EPSRC and Rolls-Royce for supporting this project.

Thank you to my friends and family for believing in me.

And most of all to my wife, Eloise, who is always there for me.

Dave Hollis
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<td>a</td>
<td>acceleration</td>
</tr>
<tr>
<td>B</td>
<td>bleed ratio ($B = \frac{\dot{m}_b}{\dot{m}_m}$)</td>
</tr>
<tr>
<td>C</td>
<td>confidence level</td>
</tr>
<tr>
<td>$\text{cov}_{ij}$</td>
<td>two point velocity covariance between components in the i and j direction</td>
</tr>
<tr>
<td>d</td>
<td>diameter</td>
</tr>
<tr>
<td>$d_{\text{diff}}$</td>
<td>diffraction limited particle image diameter</td>
</tr>
<tr>
<td>$d_p$</td>
<td>particle diameter</td>
</tr>
<tr>
<td>d_{i}</td>
<td>particle image diameter</td>
</tr>
<tr>
<td>ds</td>
<td>absolute particle image shift</td>
</tr>
<tr>
<td>$C_d$</td>
<td>discharge coefficient</td>
</tr>
<tr>
<td>D</td>
<td>diameter</td>
</tr>
<tr>
<td>f'</td>
<td>lens F-number (ratio of focal length to lens aperture diameter)</td>
</tr>
<tr>
<td>G</td>
<td>filter function</td>
</tr>
<tr>
<td>H</td>
<td>boundary layer shape parameter ($H = \delta^*/\theta$)</td>
</tr>
<tr>
<td>k</td>
<td>turbulent kinetic energy</td>
</tr>
<tr>
<td>K</td>
<td>r.m.s. coefficient in PDF conditioning</td>
</tr>
<tr>
<td>$L$</td>
<td>characteristic length scale of the physical flow geometry</td>
</tr>
<tr>
<td>$L_c$</td>
<td>characteristic ‘turbulence’ length scale ($L_c = k^{3/2} \varepsilon$)</td>
</tr>
<tr>
<td>$L_{ij}$</td>
<td>integral lengthscale (if $k = i = j$ longitudinal scale, if $k \neq i = j$ transverse scale)</td>
</tr>
<tr>
<td>$L_L$</td>
<td>(longitudinal) integral lengthscale (shortened version in general discussion)</td>
</tr>
<tr>
<td>$L_g$</td>
<td>lateral integral lengthscale</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>M</td>
<td>magnification</td>
</tr>
<tr>
<td>N</td>
<td>number of samples</td>
</tr>
<tr>
<td>N</td>
<td>Filter matrix size</td>
</tr>
<tr>
<td>p</td>
<td>static pressure</td>
</tr>
<tr>
<td>q</td>
<td>volumetric flow rate</td>
</tr>
<tr>
<td>Q</td>
<td>Q-ratio (PIV signal to noise indicator)</td>
</tr>
<tr>
<td>r</td>
<td>polar co-ordinate in radial direction ($i = 2$ in $x_i$)</td>
</tr>
<tr>
<td>r</td>
<td>separation vector</td>
</tr>
<tr>
<td>R</td>
<td>jet to cross flow ratio ($R =</td>
</tr>
<tr>
<td>$R_{ij}$</td>
<td>two point velocity correlation between components in the i and j direction</td>
</tr>
<tr>
<td>$Re_{\lambda}$</td>
<td>Taylor scale Reynolds number ($Re_{\lambda} = u' \lambda /\nu$)</td>
</tr>
<tr>
<td>$Re_j$</td>
<td>Jet Reynolds number ($Re_j = \overline{U}_j D /\nu$) (where $D$ = port diameter)</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number ($Re = U_{ref} L /\nu$)</td>
</tr>
<tr>
<td>$Re_{L_c}$</td>
<td>turbulence Reynolds number ($Re_{L_c} = k^2 / (\varepsilon \nu)$)</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>interframe time</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>time interval (between successive measurements; $\Delta T = 1/f$)</td>
</tr>
<tr>
<td>u</td>
<td>fluctuating velocity vector (Reynolds decomposed component)</td>
</tr>
<tr>
<td>$u'$</td>
<td>root mean square velocity vector</td>
</tr>
<tr>
<td>$\tilde{u}$</td>
<td>small scale filtered velocity vector</td>
</tr>
<tr>
<td>$u_{\text{type}}$</td>
<td>velocity scale (‘type’ being applicable as for timescale, $\tau$)</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>$\mathbf{U}$</td>
<td>instantaneous velocity in x (axial) direction ($i=1$ in $U_i$)</td>
</tr>
<tr>
<td>$U_i$</td>
<td>$i$ component of instantaneous velocity</td>
</tr>
<tr>
<td>$U_s$</td>
<td>particle velocity lag</td>
</tr>
<tr>
<td>$\mathbf{U}$</td>
<td>instantaneous velocity vector</td>
</tr>
<tr>
<td>$\mathbf{U}$</td>
<td>time average velocity vector</td>
</tr>
<tr>
<td>$\tilde{\mathbf{U}}$</td>
<td>large scale filtered velocity vector</td>
</tr>
<tr>
<td>$\langle U_i \rangle$</td>
<td>ensemble average of $U_i$</td>
</tr>
<tr>
<td>$\langle U_i</td>
<td>X \rangle$</td>
</tr>
<tr>
<td>$\mathbf{V}$</td>
<td>instantaneous velocity in radial or normal direction ($i=2$ in $U_i$)</td>
</tr>
<tr>
<td>$\mathbf{W}$</td>
<td>instantaneous velocity in transverse or out of plane direction ($i=3$ in $U_i$)</td>
</tr>
<tr>
<td>$\mathbf{x}$</td>
<td>cartesian co-ordinate vector</td>
</tr>
<tr>
<td>$x_i$</td>
<td>cartesian co-ordinate axis</td>
</tr>
<tr>
<td>$\mathbf{x}$</td>
<td>cartesian co-ordinate in the primary (streamwise) direction ($i=1$ in $x_i$)</td>
</tr>
<tr>
<td>$x$</td>
<td>distance in image plane</td>
</tr>
<tr>
<td>$X$</td>
<td>distance in object plane</td>
</tr>
<tr>
<td>$\Delta X$</td>
<td>dimension of interrogation cell</td>
</tr>
<tr>
<td>$y$</td>
<td>cartesian co-ordinate in normal direction ($i=2$ in $x_i$)</td>
</tr>
<tr>
<td>$z$</td>
<td>cartesian co-ordinate in transverse direction ($i=3$ in $x_i$)</td>
</tr>
<tr>
<td>$\delta z$</td>
<td>confidence interval coefficient</td>
</tr>
<tr>
<td>$\Delta Z_0$</td>
<td>light sheet thickness</td>
</tr>
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**Greek Symbols**

<table>
<thead>
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<th>Symbol</th>
<th>Definition</th>
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<tr>
<td>$\alpha$</td>
<td>jet injection angle</td>
</tr>
<tr>
<td>$\delta$</td>
<td>boundary layer thickness</td>
</tr>
<tr>
<td>$\delta'$</td>
<td>boundary layer displacement thickness</td>
</tr>
<tr>
<td>$\Delta$</td>
<td>vector grid spacing</td>
</tr>
<tr>
<td>$\tilde{\Delta}$</td>
<td>filter width</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>rate of dissipation of turbulent kinetic energy</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>error band</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>wavelength of incident (laser) light</td>
</tr>
<tr>
<td>$\lambda_s$</td>
<td>swirl strength</td>
</tr>
<tr>
<td>$\lambda_r$</td>
<td>longitudinal (axial, streamwise) Taylor micro scale</td>
</tr>
<tr>
<td>$\lambda_k$</td>
<td>lateral Taylor micro scale</td>
</tr>
<tr>
<td>$\lambda_{ij}$</td>
<td>Taylor micro scale</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>standard deviation</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Kolmogorov length scale</td>
</tr>
<tr>
<td>$\theta$</td>
<td>azimuthal plane in polar co-ordinate system</td>
</tr>
<tr>
<td>$\theta$</td>
<td>boundary layer momentum thickness</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density</td>
</tr>
<tr>
<td>$\tau$</td>
<td>temporal offset (time shift)</td>
</tr>
<tr>
<td>$\tau_p$</td>
<td>particle response time</td>
</tr>
<tr>
<td>$\tau_{type}$</td>
<td>timescale ('type' refers to the particular scale, e.g. $\tau_n$)</td>
</tr>
<tr>
<td>$\omega$</td>
<td>out of plane vorticity</td>
</tr>
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Subscripts

1  Primary flow direction (axial or streamwise, x)
2  Secondary flow direction (normal or radial, y or r)
3  Third flow direction (spanwise, out-of plane or pseudo circumferential, z)
11 Quantity relating to a correlation performed between the streamwise velocity (u) at the origin, and the streamwise velocity at a different point.
22 Quantity relating to a correlation performed between the normal velocity (v) at the origin, and the normal velocity at a different point.
12 Quantity relating to a correlation performed between the streamwise velocity (u) at the origin, and the normal velocity (v) at a different point.
a  annulus
b  bleed
c  core
h  hole, port
j  jet
o  outlet
ref  reference value
th  theta (0)

Superscripts

1  Primary flow direction (axial or streamwise, x)
2  Secondary flow direction (normal or radial, y or r)
3  Third flow direction (spanwise, out-of plane or pseudo circumferential, z)

Abbreviations

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<tr>
<td>ACF</td>
<td>temporal AutoCorrelation Function</td>
</tr>
<tr>
<td>CCD</td>
<td>Charged Coupled Device</td>
</tr>
<tr>
<td>CCF</td>
<td>space-time Cross Correlation Function</td>
</tr>
<tr>
<td>CV</td>
<td>Control Volume</td>
</tr>
<tr>
<td>DNS</td>
<td>Direct Numerical Simulation</td>
</tr>
<tr>
<td>Frame A / B</td>
<td>PIV image frames separated by time Δt</td>
</tr>
<tr>
<td>FoV</td>
<td>Field of View</td>
</tr>
<tr>
<td>HeNe</td>
<td>Helium Neon (laser)</td>
</tr>
<tr>
<td>HMN</td>
<td>Hoest-Madsen and Nielsen SGF compensation method</td>
</tr>
<tr>
<td>HWA</td>
<td>Hot Wire Anemometry</td>
</tr>
<tr>
<td>LDA</td>
<td>Laser Doppler Anemometry</td>
</tr>
<tr>
<td>LES</td>
<td>Large Eddy Simulation</td>
</tr>
<tr>
<td>Nd:YAG</td>
<td>Neodym Yttrium Aluminium Garnet (laser)</td>
</tr>
<tr>
<td>PDF</td>
<td>Probability Density Function</td>
</tr>
<tr>
<td>PIV</td>
<td>Particle Image Velocimetry</td>
</tr>
<tr>
<td>POD</td>
<td>Proper Orthogonal Decomposition</td>
</tr>
<tr>
<td>PSD</td>
<td>Power Spectrum Density</td>
</tr>
<tr>
<td>PTV</td>
<td>Particle Tracking Velocimetry</td>
</tr>
<tr>
<td>r.m.s.</td>
<td>root mean square</td>
</tr>
<tr>
<td>RANS</td>
<td>Reynolds-Averaged Navier-Stokes</td>
</tr>
<tr>
<td>SGF</td>
<td>Sub-Grid Filtering</td>
</tr>
<tr>
<td>SVC</td>
<td>Spatial Velocity Correlation</td>
</tr>
<tr>
<td>TKE</td>
<td>Turbulent Kinetic Energy</td>
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Chapter 1

INTRODUCTION
This thesis presents the applications and limitations of particle image velocimetry in the study of gas turbine combustor aerodynamics. The combustor is one of the main components of a gas turbine engine, and like many other great inventions, the development of the gas turbine engine has been a lengthy one in order to arrive at today's compact and efficient designs. Despite individuals, industry and research organisations having contributed immensely to the development of the gas turbine, we collectively strive for constant improvement. Today those areas for advancement are heavily targeted at fuel efficiency, noise reduction, and reduced emissions of polluting gases. These topics were primary drivers behind this experimental project because the fluid dynamic characteristics of the combustor play a critical role in determining gas turbine operation, performance and efficiency. In order to develop knowledge of fluid dynamic processes within the combustor, we must build upon the understanding of time average information collected to date by investigating the instantaneous behaviour. Particle Image Velocimetry (PIV) allows us to do that by capturing series of instantaneous planar velocity data, hence resolving spatial structures in the flow. This project therefore set out to develop PIV methodologies for use in real engineering flows in order to increase the understanding of fundamental unsteadiness in gas turbine combustion systems.

1.1 Gas Turbine Engines

"Inventing it was easy. Making it work was not!" – Frank Whittle

The concept of jet propulsion is said to have been inspired by Hero, who lived in Alexandria, Egypt, about 150 B.C. He invented a novelty toy driven by steam called the Aeolipile but found no industrial application for his discovery. Much later in 1550 Leonardo da Vinci sketched a device, pertaining to the theme of gas turbine propulsion, which could be placed in a chimney where the upward movement of hot gases would turn a spit for roasting meat. Ideas were slowly researched and advanced subsequent to da Vinci’s proposal over the following three centuries (Giovanni Branca, 1629; John Barber, 1791; John Bumbell, 1808; F. Stolze, 1872) however, inefficiencies prevented successful practical application of the land based designs. It wasn’t until 1903 that the first successful gas turbine engine was built in Paris, albeit with an overall efficiency of only 3%. It consisted of a three-cylinder, multistage reciprocating compressor, a combustion
Introduction

chamber, and an impulse turbine. At a similar time, Aegidus Elling, a Norwegian, started investigating the possibilities of gas turbine power and constructed an engine producing 11hp (Encyclopaedia Britannica, 2001).

The real breakthrough and principle of the modern design came in 1929, when Frank Whittle perceived the revolution in aerospace propulsion, applying for a patent in 1930. Frank Whittle and Hans von Ohain separately built and successfully ran jet engines in 1937, with Whittle being attributed with the first ever successful jet engine ground run on 12th April 1937. However, the backing of an aircraft manufacturer enabled von Ohain to build and fly an axial flow engine, the HeS 3a, in the Heinkel He178 research aircraft before Whittle on 27th August 1939 (Taylor, 1982). The Whittle W1 engine which powered the Gloster Meteor E28/39 aircraft produced 1,240lb thrust. To put that in perspective, the GE90-115B – today’s most powerful turbofan produces 120,316lb thrust, nearly 100 times that of the W1.

Since those first successful examples, the last sixty years has seen the development of the modern gas turbine engine progress significantly, although the overall basic operation has changed little since Whittle’s efforts. Indeed, all modern gas turbine engines can trace their origins back to the Whittle W1 engine. The application of the gas turbine engine on civil airliners has revolutionised air travel, providing the ability to fly faster, more economically, and more comfortably than in the days of only reciprocating engines. Today the gas turbine engine can be found not only powering aircraft, but also as an efficient means of powering marine craft, and generating electrical power. Clearly the gas turbine engine plays an important role in today’s society and economy.

The engine consists of the compressor, combustor, turbine and exhaust, and a typical modern gas turbine engine, in aircraft type guise, is shown in figure 1.1. Countless items are available in the literature regarding the overall operation of gas turbine engines and an excellent basic overview can be found in ‘The Jet Engine’ (Rolls Royce, 1995). The aerodynamic behaviour of the combustor is the focus of this project: In the most basic terms, it is within this component that the fuel and air are mixed together and burned, providing sufficient energy to drive the turbine and hence power for compressing the flow prior to the combustion system. The excess energy is then used for propulsion in aero applications.
Introduction

The basic combustor design is common to all gas turbines, whether the engine is driving a shaft (e.g. in helicopters, marine craft, propeller driven aircraft) or providing direct thrust. It is the combustion process, the way in which the fuel and air are mixed and burned, that plays an important role in gas turbine efficiency. Fuel efficiency remains of paramount importance, especially to the airborne users of gas turbines because the potential gains have many knock-on effects: a decrease in fuel consumption results in a weight saving, which can result in a smaller aircraft, and perhaps lower drag forces, or the ability to carry a greater payload. For many years efforts to increase fuel efficiency have been central to gas turbine development, but today the issue of emissions, both in terms of polluting gases and noise, are the main targets for improvement. This project sets out to provide methodologies to identify the underlying fluid dynamic characteristics of the combustion process, and generate quantitative data to provide insight into the impact on the resultant polluting gases.

1.2 Gas Turbine Combustors

The objective of the combustor is to efficiently burn the fuel and produce the lowest possible emissions across the entire operating range of the engine, whilst meeting environmental legislation and the relevant safety requirements. The ability to reliably relight the combustion products is also of paramount importance in the event of an engine 'flame-out'. A typical combustor is shown in figure 1.2. It is of the annular type,
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whereby circumferential core liners contain between 16 and 24 fuel injectors in modern designs thus creating a single circumferential combustion chamber, and circumferential annulus liners contain the combustor core. This modern arrangement provides the same volume but in a shorter and more efficient design than older designs. The older type can combustors consisted of a number of individual combustor cans (7 to 16, Lefebvre, 1999) positioned equidistant in a circumferential arrangement around the engine. The intermediate can-annular type combustor has also been used, where several separate core 'cans' were contained within a continuous annulus. Despite the differences in the combustor types, they all comprise similar basic geometries and 'zones' as illustrated in figure 1.3. It could be argued that combustor design has changed little in the last sixty years because, although much research has been made into combustor design, our understanding of the fundamental processes occurring within a combustor have not been developed sufficiently to make quantum leaps (Spencer, 1998). However, modern developments are beginning to propose more radical modifications to the basic architecture, such as introducing up to 70% of inlet mass flow through the dome region (compared to 20% in current designs). Despite the developments, and in keeping with the combustion systems studied within this thesis, only current combustor designs are described in detail here.

Fig. 1.2 An Annular Gas Turbine Combustor
The combustor has a number of tasks to fulfil in its operation, with the aerodynamic processes controlling or affecting the majority of these. It must firstly reduce the velocity of the approaching airflow from the compressor to prevent the flame being extinguished. This is achieved by providing a sheltered region within the flame tube liner. Some air (approximately 20%) passes through the dome region and is mixed with the injected fuel spray where it ignites. This is the primary zone, in which air also enters from the annulus via admission ports in the core liner creating a recirculating region to anchor the flame. Air is also introduced into the liner via wall cooling slots to produce a film of (relatively) cool air at the liner walls. Further air enters the liner through secondary (intermediate) and dilution ports downstream of the primary zone. Burning of the fuel in the rich primary zone continues in the secondary zone, where the mixture is leaner. The air admitted through the dilution ports (which cannot be seen in figure 1.3) serves to further mix out and control the high temperature spots within the combustor and cool the exiting air prior to the turbine. All of the aforementioned processes have to be achieved with the minimum possible pressure losses being incurred through the system.

![Combustor Operation](image)

**Fig. 1.3 Combustor Operation and Zones (Courtesy of Rolls Royce)**

Although the investigation looks at the overall aerodynamic characteristics of the combustor, a primary topic is the characteristics of the flow through the air admission ports, and coupling between annulus and core flow characteristics. The balance of the airflow splits through the admission ports is critical to the proper and efficient
performance of the combustor, and although our understanding of the time average fluid mechanics is arguably well developed, the fluctuating behaviour has so far received scarce attention mainly due to the lack of appropriate measurement techniques. This project aims to address those shortfalls.

1.3 Gas Turbine Emissions

An unfortunate bi-product of air travel is the pollution caused by the gas turbine engine, with a reported 750 million tonnes of pollutants emitted each year into the atmosphere from the global aircraft fleet (Secrett, 2001). Despite the adverse effects on the aerospace industry caused by the tragedies of September 11th 2001, air travel remains a growth industry. Prior to September 11th Airbus, Boeing and British Airways projected air travel to grow at 5-6% per year, a prediction echoed by Brasseur et al (1998). In the last ten years passenger traffic on scheduled airlines has increased by 60% and it should be remembered that air travel influences everyone's life whether they personally fly or not, via the carriage of freight and the use of military aircraft. Nearly four million people are employed by the aerospace industry worldwide, with a further eight million indirectly employed.

The economic implications of jet powered aircraft are clear, and what we must now concern ourselves with is developing gas turbines that are even more efficient and environmentally friendly. Some commentators on the subject (Green, 2001; Cumpsty, 2001) suggest that “the modern gas turbine engine in its current (traditional hydrocarbon burning) layout cannot be significantly improved”, and that “we can only expect a maximum of a 10% improvement [in efficiency] in the future.” Despite a pessimistic tone, such improvements would have massive economic impacts on air travel, for the reasons described earlier. Therefore those improvements must be sought, and additionally, it is a distinct possibility that the findings of such research into those improvements may spawn new innovative directions to pursue in an industry that “thrives when it is pushing technological barriers, rather than consolidating existing capabilities” (Gardner, 2003).
"Of course, the Holy Grail for the engine industry remains the clean powerplant that produces no emissions" – 'Full Throttle', Aerospace, August 2001

Unfortunately, the only current hopes of ‘zero’ emissions are nuclear or hydrogen powered solutions, which have obvious and serious safety implications. Such solutions would offer water vapour as the only ‘pollutant’ and thus appear extremely environmentally sound. However no research has been done into water vapour emissions in the upper atmosphere and while it sounds harmless, it is possible that such emissions could be worse than the cleanest hydrocarbon burning units. Although research is undoubtedly being carried out into these areas, efforts must be given to improve current conventional designs and reduce their emissions in order to comply not only with moral obligations, but to also with future legislation.

The previous section alluded to the impact of fuel-air mixing upon the fuel efficiency and emissions of a gas turbine. Incomplete combustion leads to unburnt hydrocarbons (UHCs) and carbon monoxide (CO) production, whereas combustion at very high temperatures produces nitrogen oxides (NOx). In addition to these pollutants, carbon dioxide (CO2) and water (H2O) are unavoidable by-products of the combustion process, but bear a linear relationship with the quantities of fuel burned, and hence improved fuel consumption leads to reduction of these emissions. Improvements in the overall efficiency of the gas turbine have generally reduced emissions, especially with the introduction of higher by-pass ratio units in the 1970’s and 1980’s. However, the exception to this statement is NOx emissions, which actually increased in the same period due to higher operating temperatures within the combustor.

NOx is known to be harmful at sea level to plants and animals, and is also understood to contribute to the depletion of the ozone layer at typical aircraft cruise altitudes. Hence particular focus is given to the reduction of nitrous oxides (NOx) nowadays. Oxides of nitrogen are produced via the Zeldovich chain reaction, when ordinary nitrogen and oxygen in the air comes into contact with the high temperatures. This reflects the fact that the Nitrogen is no longer inert at high temperatures such as those experienced within the combustor primary zone. This ‘thermal NOx’ emission is hence a strong function of combustor flame temperature and residence time. It has been mentioned that increases in
efficiency have came from increased temperatures within the combustor as a result of improved materials within the gas turbine, hence lowering the flame temperature would reverse those improvements. Therefore the way in which reduced NOx production is strived for is via the achievement of more homogeneous combustion by avoiding hot-spots and reducing residence times. By utilising PIV to investigate the unsteady instantaneous behaviour of the flow, this project aims to provide insight into the mechanisms by which residency times may be reduced.

Due to the impact of oxides of Nitrogen upon the environment, strict permissible levels of NOx (and other pollutant) emissions have been internationally agreed and set by the Committee on Aviation Environmental Protection (CAEP) - a branch of the International Civil Aviation Organisation (ICAO). In addition to CAEP, the main regulatory frameworks have been defined by the following:

- The 1944 Chicago Convention
- Rio Declaration (1992)
- UN Framework Convention on Climate Change (1992)
- Kyoto Protocol (1997)

All of the above were developed in their entirety, or in part, with a view to reducing the environmental impact of aircraft emissions. CAEP have been tackling aviation environment issues since 1968 and are internationally recognized as the global instrument for developed countries to pursue the limitation or reduction of emissions from international aviation. CAEP specifies levels of emissions permissible of the various pollutants mentioned earlier, and those permissible levels (in particular NOx) are continually being reduced. It is for this reason that engine manufacturers have shifted their major development attention from fuel efficiency to controlling and limiting emissions. The importance of the combustor’s role in achieving these reductions is evident by the recent developments:

- CFM56 Double Annular Combustor (DAC) in service early 1995 – Residence time at higher power settings reduced leading to lower NOx emissions. Reduced CO₂ emissions via decreased fuel consumption.
- GE low emissions combustor for CF6, 1995 – Reduced UHC, CO and NOx
• P&W TALON II (Technology for Advanced Low NOx), June 2001 – Reduced CO, UHC and NOx emissions. The latter being reduced by the fuel rich front end promoting rapid mixing in its quick quench zone, minimising the residence time.
• RR Low Emissions Phase 5 combustor, 2001 – Reduced NOx emissions by 40% by lowering peak combustion temperatures.

Rolls Royce is currently leading a European initiative to develop a next generation combustor in the ANTLE (Affordable Near-Term Low-Emissions) project. Based on a heavily modified version of the Trent 500 engine, it aims to reduce NOx emissions by 60%. In parallel (and in competition) to the European work, the aero engine industry in the United States is concentrating on their Efficient and Environmentally Friendly AeroEngine (EEFAE) programme. CFM are also reported to be working on ‘TECH56’ which aims to achieve a 5-7% reduction in fuel consumption. This includes a new Twin Annular Pre-Swirl (TAPS) combustor which has 50% below current NOx and 10% below current UHC output limits set by ICAO.

"Like washing powder, unleaded petrol and recycled paper, there could soon come a time when passengers may choose which airline to fly on the basis of which is the most environmentally responsible – in fact it is surprising that no airline has fully exploited this yet” – ‘Full Throttle’, Aerospace, August 2001

In addition to legislative obligations, recent proposals (Read, 2003) have included financial penalties related to aircraft pollution. The British Airports Authority (BAA) plan to introduce an emissions related element for landing charges in which the most polluting aircraft will pay more. Friends of the Earth would also like to see tax on aviation fuel, as is applied to other forms of transport. The UK government is reported to be keen to participate in a European-wide-emissions trading scheme to be introduced 2008. In this, there would be a limit on the total pollution permitted by airlines in which individual operators could buy and sell their shares. It is also envisaged that this buying and selling of greenhouse gas ‘permits’ could also operate between industry sectors.

It is thus clear that the improvement of gas turbine engines in terms of efficient and environmentally acceptable performance is essential to future air travel growth, and that the aerodynamic performance of the combustor is vitally important to this goal. Hence research continues to grow in an attempt to understand the totality of the aerodynamic
processes within the combustor, but elements of that understanding remain elusive at present. As mooted earlier, little is known about the unsteady aerodynamic behaviour within the combustor and the spatial flow structures present, i.e. the time variations of the flow evolution that makes up the (relatively well understood) time average picture. It is these mixing processes within the combustor that are crucial in achieving complete combustion, and the reduction of residence time.

1.4 Previous Work

This investigation follows on directly from those of Spencer (1998) and Griffiths (2000). The former performed an investigation in a water analogy facility, the geometry of which represented a section of simplified gas turbine combustor. The work concentrated on annulus-core interaction for a variety of mass flow splits typical of primary and dilution port flows. The experimental method employed was Laser Doppler Anemometry (LDA), which gives temporally resolved measurements at a point within a flow field. Moving the LDA control volume allowed time averaged data fields to be generated. However, taking single point LDA measurements does not allow the quantification of structures or spatial correlations within the flow field. This was not the aim of collecting that LDA data; it was acquired in the simplified combustor geometry to validate a time average CFD simulation also reported in Spencer (1998). The CFD technique was subsequently applied to realistic combustor geometry, similar to that studied experimentally by Griffiths (2000). This later investigation was performed on an isothermal airflow test rig directly representative of a sector of a production gas turbine engine, with the flame tube being an actual production item and the Perspex casing replicating the combustor casings (including dump diffuser and exhaust sections). LDA was again the primary measurement technique employed to map out the mean and turbulent characteristics of the internal flow field. Additionally Hot Wire Anemometry (HWA) and pressure probes were utilised in order to acquire data in the regions external to the flame tube where there was a lack of optical access. As the work reported within this thesis follows on directly from the two aforementioned studies, extensive reference to these is necessary throughout. Wherever possible, replication of information is avoided, but inevitably it is essential in some sections in order to provide a complete description.
The following sections describing previous work are divided according to component or feature position working downstream from the combustor inlet. It should be noted that some studies described are not specific to gas turbine combustors, such as jets in cross flow. It should also be borne in mind that most experiments detailed are in isothermal test facilities (i.e. non-combusting), due to the vast increase in complexity and cost that arises when considering combusting flows. However, this is a legitimate approach and one adopted by many workers, due to the fact that the primary zone flow is controlled more by physical than chemical reaction (Bicen et al, 1989), and hence the main area of interest are the aerodynamic processes. Anacleto et al (1996) reinforce the difficulties of making measurements of the complete velocity field within real combustors, stating that isothermal constant density systems are much more viable. However, they point out that exact similarity between isothermal and reacting flows is clearly not possible, and state that in real combustors the primary recirculation zone is shorter and wider than that in the equivalent isothermal flow: The mass flow recirculated is smaller and the reverse velocities larger due to the decrease in density. In contrast to the generally accepted opinion that aerodynamic processes dominate the primary zone, Cameron et al (1989) report that the flowfield is significantly altered in the presence of reaction, and state that strong on-axis backmixing in the dome region present in isothermal flow is dissipated in the case of reaction. The swirl airflow dominates the dome region, and the primary jet flow does not interact with the swirl to enhance the strength or size of the recirculation zone. The turbulence levels are, however, increased with the higher velocities due to the primary jet injection. Therefore, although isothermal measurements are practical, some care is needed when correlating to hot conditions.

1.4.1 Diffuser

As the flow exits the (dump) diffuser it is bounded by the large stationary vortices in the dump cavity and the flame tube head. Flow either passes into the flame tube via the fuel injector swirler, or accelerates around the flame tube head, creating velocity profiles strongly biased toward the surface of the flame tube (Carrotte et al, 1993). However, due to the high levels of turbulence within the diffuser, outlet guide vane (OGV) wakes are mixed out as they pass through to the annuli (Barker et al, 1997). Upon exiting the compressor the highly three dimensional flow may exhibit residual swirl, due to over- or under-turning of the flow under different operating conditions. The decreased radius at
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the inner annulus means that, due to the conservation of tangential momentum, any swirl that is present at diffuser exit will be enhanced and may therefore affect the jet characteristics and discharge coefficient (Carrotte et al, 1995). Spencer (1998) emulated residual swirl in the annulus and found it to have a significant effect upon the port flow exit and subsequent core flow characteristics (described later).

1.4.2 Fuel Injector Swirler

It is well recognised that in addition to the air admission ports, the flame tube is heavily influenced by the flows entering through the fuel injector swirler. When sufficient rotation is imparted to the flow, a toroidal flow reversal is created within the swirler exit ‘cone’ due to the low static pressure (Lefebvre, 1999). The swirl number, S, is the ratio of axial flux of swirl momentum to the axial flux of axial momentum, multiplied by the swirler radius. This parameter essentially describes the amount of swirl, and Lilley (1977) found that for S=0.64 (in the absence of admission jets), the reverse flow region had a length equal to four swirler diameters. The recirculation also extended with increasing swirl number; a finding echoed by Escudier and Keller (1985). Ahmed et al (1992) found that significant anisotropy existed in the normal stresses with the highest values existing at the high velocity gradients in the shear layers at the swirler exit.

Anacleto et al (2003) presented data relating flow pulsations caused by hydrodynamic instabilities in the form of a precessing vortex core (PVC) with the downstream combusting flow in a lean premixed model combustor. They state that flow regimes that may lead to low NOx emissions (i.e high swirl number) are prone to increased flow oscillations (in the form of a PVC), whereas the target for combustors with lean premixed flames would be a regime with a steady recirculation zone and stabilized combustion in order to avoid large amplitude velocity and pressure fluctuations causing undesirable effects such as loud noise and vibrations. For the geometry studied, it is found that stable performance with low emissions is achieved for S=1.05. Midgley et al (2004) also find unsteadiness in their PIV study on a radially fed swirler, reporting evidence of highly turbulent mixing in the swirling shear layer leading to the identification of energetic discrete vortex structures, the origin of which being traced back to a suggested vortex breakdown and separation process inside the fuel injector and causing a twin vortex pattern to form.
1.4.3 Annulus-Port Flows

The velocity profile entering each annulus is strongly biased, as stated in Section 1.4.1, but this may mix out prior to the ports (Fishenden and Stevens, 1977). However rapid switching of the bias towards the casing has also been reported (Carrotte et al, 1995b). Spencer (1998) found that the axial momentum of the flow in the annulus was passed onto the port flow and jet, causing it to enter the flame tube at an angle. Due to the simplified geometry and lack of upstream combustor representative diffusing section, relatively low turbulence intensities in the annulus and jet cores were found. However, in agreement with Carrotte and Wray (1991), Griffiths (2000) found high turbulence intensities of up to 35% in the annulus of the engine representative geometry and found that these levels were transmitted into the jets themselves.

Manners (1987) performed an early computational study of coupled annulus-port flows, finding that significant changes in jet characteristics occur when changing the geometry from plain to plunged. In contrast, Spencer (1998) tested several port shapes and found that flow conditions affected the jet characteristics more than port shape. Additionally, in the later and more advanced computational study, Spencer (1998) found that the calculations poorly predicted the separation seen in the experimental data at the upstream edge of the port, including those with radiused inlets.

Carrotte and Stevens (1990), Baker and McGuirk (1992), Doerr et al (1995) and Spencer (1998) observed instabilities through the ports in terms of vortical motion, resulting in a twisting or distorting of the jet in the core flow. These vortices were observed in the time averaged data under certain operating conditions: namely low annulus bleed ratios. The occurrence of instantaneous vortices as observed by flow visualisation were stabilised by increasing the bleed ratio or annulus depth. At high bleed ratios, the vortices did not occur. In these studies there are differences reported as to whether the presence of through port vortices actually reduces the port discharge coefficient, with Spencer (1998) observing no change. Spencer observed interaction in the annulus between the swirling motion through the ports in the form of contra rotating vortex pairs at adjacent ports. It was envisaged that separation from the annulus wall at the outer casing was related to the through port vortices. It was also found that this separation had a major effect upon the velocity profile issuing from the port. Carrotte and Stevens (1990) note that the vortices
are only present transitorily with one vortex always dominating. They state that plunged hole shapes are particularly susceptible to this effect, but that shapes such as D-shaped holes may provide corners on which the counter rotating vortices downstream of the hole can anchor, thus stabilising the vortex pair. Peterson and Plesniak (2002) investigated the effect of feeding the port flow with co-flowing, and counter-flowing plenums using PIV. Although both cases are effectively zero bleed, they find the dominant velocity direction in the plenum (relative to the cross flow) to have a strong effect on the resultant jet trajectory, with the co-flowing case (as is the situation in the annuli of gas turbine combustors) producing a steeper angle jet but reduced lateral spreading. In-hole velocity and vorticity data indicate that the co-flowing case has strong in-hole vortices with a sense of rotation that enhances the Counter rotating Vortex Pair (see following section), with the coherence aiding the increased jet trajectory.

1.4.4 Jets in Cross Flow

Jets in Cross Flow (JICF) are relevant to many fluid dynamic studies in addition to combustor flows, such as VSTOL aircraft in transitional flight and cooling tower effluent dispersal, and therefore there are many published works available (see Margasson et al, 1993, for a comprehensive review of the subject field). In the combustor, any flow into the flame tube through the air admission ports effectively results in a JICF type flow.

Early investigations of jets in cross flow tended to concentrate on determining the trajectory of the jet, for different jet to cross flow ratios (R). For example, Platten and Keffer (1971) investigated jet trajectory by tracking the velocity maxima, whereas Ramsey and Goldstein (1971) used temperature to measure the path of the jet. Srinivasan et al (1982, 1984, 1986) also used temperature as a means of studying a jet issuing into a confined crossflow. The temperature distribution on an axial downstream plane was measured; this is a particularly relevant parameter because it identifies whether complete mixing has occurred, thus describing the turbine inlet temperatures. Srinivasan et al found that jets issuing into a confined geometry were deflected more than the equivalent jets in infinite crossflows, due to the jet interacting with the opposing wall.
Andreopolous and Rodi (1984) described the jet as presenting a blockage to the cross flow, and therefore the force exerted on the jet causes it to bend over and eventually align with the cross flowing stream. Like the findings of Kamotani and Greber (1972), they observed a steady Counter-rotating Vortex Pair (CVP) being established in the jet itself in the far field, the result being the typical kidney shape cross section of the jet in cross flow. Carrotte and Stevens (1990) state that, in the case of multiple jet injection at dilution port like R ratios, the cross flow is the dominating motion, and the jet CVP decays rapidly. Smith and Mungal (1998) also identify the CVP structure, but find that it can become asymmetric. Additionally, in agreement with Fric and Roshko (1994), they describe a Karman-Like Vortex (KLV) street being shed from the jet column, like those from a cylinder in a cross flow. They originate at the base of the jet, and become entrained by it. Papaspyros et al (1997) studied a relatively high jet to cross flow ratio of 7, where the jet issued from a nozzle located well into the free stream of the cross flow, i.e. outside the boundary layer in the homogeneous region. They studied the vorticity in the jet wake and found that, whereas the KLV are formed when the jet issues at the 'floor' and into the boundary layer of the cross-flow, these are not formed when the jet issues into the region outside the boundary layer. This is stated to support the theory of Fric and Roshko (1994) who suggest that the KLV in the jet wake originates in the boundary layer, which in turn is in agreement with Haven and Kurosaka (1997). Kelso et al (1993) also suggest that this system originates in the boundary layer, and consists of a series of vortex loops that are lifted away from the flat wall as the boundary layer separates. This disagrees with the theory of McMahon et al (1971) who suggests the jet behaves like a solid cylinder, and so the KLV originates from the jet/cross flow interface, regardless of proximity to the boundary layer.

At the low jet to cross flow ratios studied, Andreopolous and Rodi (1984) observed shear layer vortices, which they describe as providing favourable mixing conditions for the dilution jet scenario. These shear layer vortices were also seen by Perry et al (1993) and Kelso et al (1996) who document them as forming as the shear layer separated from the jet orifice. They describe the vortices being formed with a Kelvin-Helmholtz like roll up, forming concentric rings with the jet. A further vortex system is seen in the form of a bound horseshoe vortex, identified by other workers including Krothapilli et al (1990). This system is analogous to vortices generated at wing-body junctures on aircraft.
One area relatively poorly understood is that of the evolution of the jet shear layer vortices. In one of the more recent experimental studies published, Lim et al (2001) study this aspect and find that the development of the large scale structures within the JICF are not as previously postulated by other researchers. They investigate the topic qualitatively by injecting dye into the jet at strategic locations and recording the fluid motion for a jet to cross flow ratio of 4.6, typical of combustor primary port flows (although the Reynolds number is lower at Re=1600). They find that the vortex rings which extend circumferentially around the jet column in free jets, are not present when the jet is subjected to a perpendicular cross-stream. Previous researchers (e.g. Perry et al, 1993) had supported the notion of these vortex rings in JICF, but the work of Lim et al clearly rules this out, and is supported by the Large Eddy Simulations of Yuan et al (1999). Their findings are linked to the observations in previous studies, adding credit to their explanation, and they point out that cross-sectional views of JICF can give misleading observations. They believe this problem led many researchers to interpret the mushroom shape in the jet and the vortex loops on each side of the jet as being joined circumferentially, wrongly pointing towards the vortex ring explanation. They find that the vortex loops commonly seen on the upstream and lee-side of the jet are actually
formed via the deformation of the jet column. These loops are independent of each other in contrast to the vortex ring theory where for every upstream loop there must be a lee-side loop, and either the upstream or lee-side loop arrangement can exist in isolation, a phenomenon which has been reported in low cross flow ratio experiments (Kelso et al, 1996) and those of buoyant jets. Illustrations of their findings are given in figures 1.4 and 1.5.

![Diagram](image)

**Fig. 1.5 Section View of the Formation of a Jet in Cross Flow (Lim et al, 2001)**

Haven and Kurosaka (1997) used a water tunnel with Laser Induced Fluorescence (LIF) and PIV to study the effect of hole exit geometry on the near-field characteristics of crossflow jets. The motivation of their study was related to film cooling, where it is critical that the jet remains attached to the wall. They suggest, in agreement with some other workers in the field (e.g. Sykes et al, 1986), that the origin of the CVP (or ‘kidney’ vortices, as Haven and Kurosaka refer to them) is in the jet pipe boundary layer. They also find that the outer portions of the leading edge boundary layer spill out sideways upon leaving the hole and combine with the side boundary layers. The central portion of the leading edge boundary layer, however, is shed periodically from the hole and rides on
top of the jet. This agrees with the observations of Lim et al (2001). Haven and Kurosaka (1997) also discuss kidney, and anti-kidney vortices, and show that they are developed from the boundary layer spillage. This could refer to the side arms seen in figures 1.4 and 1.5, and the development of those within the CVP. It is difficult to derive direct comparisons however, because most measurements are very close to the floor. They also make reference to the vortex ring, which Lim et al (2001) showed was not present.

Rows of opposing jets in cross flows are particularly relevant to the primary jet situation, and this topic was studied by Sivasegaram and Whitelaw (1986). They found that the flow field resulting was symmetric, but slight misalignment of the jets would result in large flow asymmetries. In agreement, McGuirk and Palma (1992) found that discrepancies in flow inclination caused strong asymmetry in the flow fields resulting. Hatch et al (1995) and Zhu et al (1995) found that momentum flux ratio was an important factor in opposing jets, whereby above a certain ratio (dependent also upon cross flow duct geometry) the jets impinge and bifurcate to create a portion of backflow which in turn results in a recirculation zone. Spencer (1998) found this to be the case for $R>4$. Liscinsky et al (1996) studied multiple jets issuing into a cylindrical duct and found that jet penetration remained similar with variations in orifice size, shape and spacing. Three-dimensional flow was observed as being a key part of efficient mixing, and the isothermal, non-reacting flow field seemed to represent the reacting case well. Besbes et al (2003) investigated the interaction of two opposed jets both experimentally (using PIV) and numerically. They found that, if one jet is heated, the stagnation point moves towards that jet.

Fernandes et al (1996) compared a single confined JICF with two opposed jets in a channel. They found that the single confined jet is similar to an unconfined jet in the near field, but is bent over more than an unconfined jet as it gets closer to the confining wall. They observe that the confined jet exhibits a slow decay of velocity excess in the far field, indicating poor transverse turbulent transfer of linear momentum. It is proposed that the CVP are the dominant mechanism of turbulent transport in the unconfined jet, and beyond the region of maximum deflection the confining walls limit the growth of the CVP, thus reducing the turbulent transfer. In the case of non-impinging opposed jets, the
jets are seen to periodically ‘pinch-off’ regions of the cross flow where they meet. These pinch-off regions are convected by the flow but remain unmixed for large distances. For impinging opposed jets, there is less bending and counter-rotating vortices occur quasi-periodically upstream of the jet impingement region.

Tsunoda and Saruta (2003) used PIV and Planar Laser Induced Fluorescence (PLIF) to study a jet in counter-flow. This setup is actually quite different to a jet in cross flow, but included in the discussion here because it is approximately representative of the swirler exit flow acting against the reverse flow column produced by the impinging primary jets. They found that the jet ‘wandered’ quite noticeably, causing a bimodal PDF near the stagnation point, a feature identified by Spencer (1998) at the head of the core recirculation. Bernero and Fiedler (2000) also studied a jet in counter flow using PIV, and via analysis of decomposed velocity fields (see chapter 3) identified periodic variation in jet flow penetration and vortex shedding.

1.4.5 Internal-External Coupling

Several years ago, the standard practice in both experimental and computational studies of combustors was to separate the internal and external flow field. As the internal flow field was considered of prime importance, assumed boundary conditions were enforced on the port flows by, for example predicting discharge coefficients and mass flow rates (Lowe, 1994). Such assumptions are clearly incapable of replicating any port or jet features resulting from coupling of the internal-external flows such as the through port vortices described in section 1.4.2. Instead of empirical predictions, Shyy et al (1988) used experimental five hole probe data to define jet entry characteristics at the flame tube in the reported numerical calculations. This approach was let down by the coarse grid upon which the experimental data was collected, illustrating the problem in the need for large amounts of experimental data to define boundary conditions of internal only calculations.

Experimental data of fully featured combustors is expensive to obtain, therefore early simulations of combustor flows featuring external flows and port flows (e.g. Coupland and Priddin, 1986; McGuirk and Palma, 1992; Lin and Lu, 1993) had little validation data in support. The solution to this problem seems to be the adoption of a ‘building blocks’
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approach, whereby CFD predictions are made for simplified geometries featuring combustor like flows for which experimental data is more easily obtained, before the validated computation scheme is applied to fully featured geometries (e.g. Spencer, 1998).

Merdjani (1989) mapped out an isothermal model of a circular section incorporating 8 ports supplied by a cross flowing annulus using five hole probes and HWA. In this investigation it was found that a toroidal recirculation zone existed upstream of the jet impingement for jet to cross flow ratios, R, greater than 4. LDA based experiments including coupled external-internal flow fields were carried out by Bicen et al (1989) for a generic can-type combustor using water as the working medium. The geometry included a swirl driven primary zone with two rows of radially inflowing jets. A similar configuration was investigated by McGuirk and Palma (1992) who found instabilities at the primary jet impingement in the 10-150Hz frequency range. The dilution jet was also found to have an unsteady penetration depth, oscillating at a frequency of 7Hz. In a parametric study, Spencer (1998) introduced annulus swirl at its inlet to emulate residual swirl exiting the compressor. This was found to have a significant effect on the internal core flow, skewing the inlet flow and thus deflecting the jets and therefore causing axial swirl within the core and misalignment of the normally impinging jets. A reduction in port C_d of 8% was also observed, together with a smaller primary recirculation zone due to the reduced impingement.

Hughes and Carrotte (2004) carried out experiments using HWA and LDA in the same fully featured combustor sector as Griffiths (2000). Cross correlations of simultaneous measurements indicated that velocity fluctuations in the feed annulus correlate with fluctuations inside the port and flame tube. They state that the results show that the time dependent external flow field can influence the flow field within the flame tube and hence, potentially, flame stability, mixing and emissions. This reinforces the notion that numerical modelling should include interactions between the feed annulus and flame tube.
1.4.6 Complete Combustor Studies

It is obvious that the swirler flow and primary admission port jets will interact in the combustor, although most experiments consider these factors in isolation for simplicity. Nevertheless a select number of complete combustor studies do exist. As the momentum of the primary jets is increased, their contribution (compared to the swirler flow) to the primary zone increases (Richards and Samuelsen, 1992). Koutmous and McGuirk (1989) state that at a certain point the jets will dominate the fluid entering the recirculation zone, with the swirler influence causing an increase in reversed flow rate (McGuirk and Palma, 1995). They found that instabilities exist in the primary jet with dominant frequencies corresponding to a Strouhal number of 0.27. Low frequency eddies and bimodal PDFs were observed for the azimuthal velocity component between impinging jets, which is a concern because such a flow characteristic may be a source of instability or noise. Bicen et al (1989) found that little swirl is imparted to the primary jet back flow, with most of the swirling fluid being taken down the combustor flame tube walls, although this would depend on the relative positioning of the swirler and admission ports. As the ports are moved closer to the swirler, the impinging flow directly feeds the toroidal recirculation within the swirl cone. McGuirk and Palma (1995) state that the aforementioned jet instabilities can be reduced if the swirler rate is increased. Liou et al (1989) studied the spectral characteristics of a model combustor where an axial flow represented the air and fuel entering through the swirler, and side wall jets represented the primary jet flows. They detected an axial oscillation at 37.5Hz associated with the axial jet, but a 75Hz oscillation in the transverse direction (associated with the side jets). There hence appears to be strong evidence for periodic type disturbances in gas turbine combustor flows. If these disturbances were to cause resonant interactions between driving processes and coupling modes, oscillations in the flow could induce many undesirable effects such as large amplitude structural vibrations, increased heat fluxes at the system walls, flashback, and flame blowoff (Ducruix et al, 2003). Rapid changes of the flame surface generate an intense radiation of sound, and mechanisms which are able produce or destroy the flame surface at a fast rate include collisions between the flame and neighbouring flow structures such as vortices.

Anacleto et al (1996) use an isothermal set-up for the detailed study of a ‘Rich-burn, Quick-quench, Lean-burn’ (RQL) can combustor model using an LDA measurement
The measurements of Griffiths (2000) in the fully featured combustor sector indicated that strong coupling existed between external and internal flows, with differences in geometries and flow splits in the inner and outer annuli causing opposed ports to draw air from different regions of the annuli, and with different velocities. This contributed to significant differences in discharge coefficient, initial jet pitch angles and flow topology between opposed jets. The primary jets downstream of the swirler were particularly sensitive to feed conditions, with the low pressure flow drawing the outer primary jet (with its ‘stronger’ jet characteristics) upstream. As a result, the primary jets become misaligned, the outer jet dominates the primary zone, and the inner jet is deflected downstream. The lack of impingement results in lower local turbulence levels, which would have a direct effect on mixing rates, efficiency, and pollutant emissions. Although a more conventional primary zone topology existed at the sector edges, the asymmetry seen by Griffiths (2000) at the sector centre was more severe than any annulus-core coupling effects of other workers, and showed that a conventional primary zone could not be assumed. The data proved that external aerodynamics must be included in predictions. In support of the findings of Merdjani (1989) and Spencer (1998), it also showed that a highly anisotropic turbulence field existed within the core, indicating that although the $k$-$\varepsilon$ turbulence model was suitable for calculating mean velocities, it was unsuitable for predicting the turbulence fields. Unsteadiness in the form of bimodal PDFs, particularly at jet impingement also suggested that time dependent predictions such as LES were required.
1.5 Particle Image Velocimetry

Particle Image Velocimetry (PIV) has been selected as the most suitable measurement technique to successfully complete the objectives of this investigation. Its suitability lies in the ability to provide spatially resolved velocity data across a domain of interest. PIV is a relatively young and hence rapidly developing technique which has undergone several interesting developments over the last 30 years.

"In this particular technique, the fluid is illuminated by a thin, pulsed sheet of laser light, and the components of the local velocity vector that lie in the plane of the sheet are inferred from the local displacements of the image field...The [PIV] technique provides measurements of the velocity that can, with sufficiently high particle concentrations, be nearly continuous in space at one instant in time. In this sense it compliments LDA; which provides measurements of the velocity that can be nearly continuous in time at one point in space" – Adrian and Yao (1985)

PIV essentially takes flow visualization and quantifies it. Hence the earliest roots of PIV must be attributed to the first flow visualization experiments. Leonardo da Vinci is known to have made detailed sketches of the structures within water flows, so perhaps this is the first use of 'flow visualization'. The first record of flow visualization being used in engineering type flows is that of Ludwig Prandtl's water channel experiments to visualize the flow behind an aerofoil (see Rotta, 1990). Prandtl changed the angle of the aerofoil in a channel seeded with mica particles on the surface, and made the first qualitative studies of unsteady aerodynamics. That same experiment has been reconstructed and quantitative information extracted via PIV analysis (Raffel et al, 1998).

As early as 1932, Fage and Townsend used manual particle tracking procedures to quantify 'instantaneous' fluid motion, and background papers (e.g. Grant, 1994) draw analogies to astronomers tracking individual stars as being involved in similar analysis procedures. Whereas Fage and Townsend tracked individual tracers within the flow (Particle Tracking Velocimetry, PTV), in Particle Image Velocimetry individual particles are not traced. In PIV the domain is discretised and the progress of groups of particles are followed and calculated: a key assertion that will be returned to in later chapters. Adrian (1991), one of the architects of the modern PIV technique, provides a useful
characterisation of the branches of what he termed ‘pulsed light velocimetry’ (see figure 1.6 below).

![Diagram of Pulsed Light Velocimetry]

**Fig. 1.6 Forms of ‘Pulsed Light Velocimetry’ (Adrian, 1991)**

Although this project utilises the branch of high image density PIV, it is important to recognise the other forms of imaging Velocimetry (described below). It should also be appreciated that the invention of the laser in the mid 1960’s played a crucial role in enabling the advancement of all branches of image velocimetry by providing a form of manageable coherent light source.

Molecular markers are important when looking at reacting substances, and can be used to indicate concentrations of chemicals within reacting fluids. Fluorescent markers are also frequently used today in spray analysis where a fluorescent dye is injected into a co-flowing fluid, thus enabling the separation of the two fluid motions. Speckle pattern velocimetry has its origins in solid mechanics, and came under the heading ‘Speckle Metrology’. The speckle interference pattern (Speckle Interferometry) produced when coherent laser light is reflected from an optically rough surface was utilised as a method of measuring surface distortion such as crack propagation (Archbold and Ennos, 1972), and is considered by many as providing the roots of modern PIV. Barker and Fournier (1977) used the method to measure displacements in transparent solids, thus indicating its potential to measure fluid motion.

In parallel with the direct analysis of speckle patterns via the displacement of speckle centres, the method of analysing high image density patterns using the Young’s fringe
method was developed as an efficient means of calculating high density image displacement, without reference to digital processing. Some attention was given to the potential of this method very early in the development of techniques (e.g. Burch and Tokarski, 1968), and more recently its application to PIV was proven by Adrian (1988). However digital processing, and the common recognition of its increased efficiency in terms of cost, performance and flexibility really overruled the advancement of Young's Fringe and Optical Correlation analysis in the 1990's (e.g. Huntley et al, 1993).

We thus arrive at modern Particle Image Velocimetry, which is defined in figure 1.6 as being categorised as either High Image Density PIV (with its inherent links to speckle velocimetry), and Low Image Density PIV (PTV). PTV, as already mentioned, is concerned with the tracking of individual particles, and therefore necessitates low particle image density in order for the processing algorithm to track each tracer with a sufficient confidence level. Cenedese and Querzoli (1997) applied PTV analysis in the experimental examination of simulated pollutant dispersion in the boundary layer of the atmosphere. They point out that PTV measures the velocity at random locations, as opposed to PIV's regular grid, and that PTV is more suitable than PIV in the case where the distance between particles is greater than the smallest spatial scale of the velocity field relevant to the investigation. Whereas PIV results in a succession of velocity values at fixed spatial locations (a Eularian description), PTV gives velocity samples along particle trajectories in a Lagrangian frame of reference, requiring a long series of single-exposed flow fields. Therefore PTV is more suited to low fluid velocities such as convection experiments, but it does hold some advantages over PIV: a Lagrangian description is relatively easily translated to a Eularian reference frame, whereas the converse is more difficult. And in PTV analysis particles can cross the paths of others with that motion being accurately tracked. This is not possible in PIV due to the way in which each velocity vector is obtained at a fixed location over a finite volume. It may not be obvious why this would be useful as it doesn't physically make sense that fluid would 'cross' itself in the same two-dimensional plane. However, if spray analysis is considered, where (liquid) particles are of different sizes, it is possible that those particles will cross each others path, a point raised by Zimmer (2001). Zimmer also shows that simultaneous drop sizing in such situations is possible. Despite these advantages of PTV, it is less widely adopted than PIV today because PIV allows the acquisition of higher data
Introduction

density in more demanding flow scenarios. Indeed, PTV is not used as an experimental tool in this project primarily for those reasons, but will be referred to in later chapters.

In High Image Density PIV the seeded flow field images are recorded on photographic film, or by a form of digital recording. The recorded field, or series of fields, are then divided into smaller regions called interrogation cells, and the fluid displacement within each interrogation cell calculated via cross-correlation algorithms. Adrian and Yao (1985) developed the two-dimensional spatial correlations that underlie the modern algorithms employed in today's PIV software analysis tools.

With the advent of modern digital recording, the use of photographic film in recording PIV images quickly became obsolete. Nevertheless, this practice was successfully used until the mid 1990's, when it was recognised that digital imaging techniques offered a more economical solution in terms of processing time, whilst at the same time retaining the accuracy of photographic film (Westerweel et al, 1996). In addition to the drawbacks of photographic film in terms of processing time, the recording had to utilise multiple exposed frames. This involved a mechanically shuttered or pulsed laser illuminating the seeded flow field twice (or more than twice), within a very short period of time. All particle images were captured on a single photographic frame, with the process being repeated at regular sampling intervals, according to the frame rate of the camera. Because all images occurred on the same frame (for a given 'instant'), directional ambiguity was a problem for this analysis technique, and so necessitated complicated techniques such as image shifting (imposing an artificial offset on the second of the two recorded fields; Adrian, 1986) or colour coding of the two pulses. The most widely adopted solution to directional resolution today is the use of separate singly exposed images – an approach that became viable with the advent of high speed electronic recording devices such as CCD (Charged Coupled Device) cameras. The use of such electronic devices has been widely referred to as Digital PIV (or DPIV) in the literature, but as it is the only form of PIV utilised in this project, it will simply be termed 'PIV'. Although even today's CCD arrays don’t match the spatial resolution of photographic film, the use of modern cross-correlation algorithms means that the results are as accurate, and calculated without the need for slide scanners or mechanically traversing scanning microscopes.

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Introduction

In addition to some of the examples described in Section 1.4, PIV has been successfully applied to a range of problems relevant to, or using actual components from, gas turbine combustors. Fujisawa and Satoh (2001) used PIV to study the flow in a model combustion chamber, although this investigation was restricted to an analysis of mean flows in a very simplified isothermal setup not specific to gas turbine engines. Seongkyu et al (1999) used PIV to investigate the effect of strain rate on NOx emissions in an opposed impinging jet flame combustor. Although the PIV measurements were at a relatively poor spatial resolution, they state that a reduction in NOx emission could be attributed to the enhancement of intermolecular mixing between cold and hot-spots in fully turbulent flows. Lazaro et al (1998) use both LDA and PIV to investigate the turbulent structure within a generic lean premixed prevaporized combustor. In their geometry which does not include admission port flows, they find that the swirling jet issuing from the main premixing tube dominates the flow within the main combustion chamber. Locke et al (1998) applied optical imaging techniques (PLIF and Mie scattering) to a rig under engine representative (combusting) temperatures and pressure, and Willert and Jarius (2002) report the applicability of PIV to combustor flows under 'realistic' (results being obtained at 3 bar pressure) operating conditions. However, these investigations mainly concentrate on the feasibility of carrying out optical based methods to obtain measurements in these challenging environments, rather than reporting detailed information about the fluid dynamics of the flows. Willert and Jarius (2002) report that the application of PIV in flames to date has been restricted to small-scale facilities or nearly laminar flow conditions (Han et al, 2000; Stella et al, 2000). They note the main challenges as restricted optical access, optical distortions (e.g changes in refractive index along the optical path), and flow seeding. Although it is pointed out that, despite the challenges, PIV is more robust than LDA in these environments due to LDA being reliant on the crossing of the two beams; a significant problem where beam steering effects due to refraction are strong.

A detailed description of the actual PIV system employed in the project is included in Chapter 2, followed by a mathematical description in Chapter 3.
1.6 Objectives

The objectives are inherently linked to the recommendations of the theses of Spencer (1998) and Griffiths (2000), with the aims having been alluded to in previous sections. Particle Image Velocimetry will provide the means to examine and quantify the underlying fluid dynamics of turbulent flows within gas turbine combustors. Therefore the overall objective of the project is to commission an appropriate PIV system, and apply it to the flows of interest, modifying the test facilities where necessary in order to acquire the data. Comparisons between the data collected from this new measurement technique and previous data must be carried out in order to assess its performance, and any resultant limitations identified. Appropriate methodologies to efficiently handle, analyse and present the PIV data should also be developed and validated. These methods will be used to demonstrate the findings from the two experimental facilities.

1.7 Thesis Structure

An introduction has been given in this chapter, including a comprehensive background to relevant studies and the development of PIV. The following chapter details the experimental facilities employed during the project, including the commissioning of those facilities. Chapter 3 details the PIV technique, including mathematical descriptions, optimisation procedures, and appropriate analysis and presentation techniques. Chapter 4 discusses correlation analysis using PIV data in both the spatial and temporal domains, and forms the basis for the ability to calculate the integral lengthscales and other fluid dynamic quantities discussed in chapter 5. Chapter 6 uses the aforementioned techniques in the analysis of sub grid filtering effects inherent to the PIV methodology in practical engineering flows such as the gas turbine combustor. Chapters 7 and 8 then present the results from the two experimental facilities, with chapter 9 concluding the Thesis.
Chapter 2

EXPERIMENTAL FACILITIES
The motivation behind this project was detailed in chapter 1, along with descriptions of findings so far in the research arena of combustor aerodynamics. The justification for using isothermal test facilities was also given, as were the reasons behind utilising PIV as the primary measurement instrument. In this chapter the two experimental facilities, and technical specification of the PIV system are comprehensively described, as is a brief description of the secondary measurement tool, the LDA system.

The two test facilities used in the project are located within the Department of Aeronautical and Automotive Engineering at Loughborough University. The first half of the test programme employed a water analogy rig to study simplified combustor representative geometries. This was followed by testing carried out on a fully featured sector of a Rolls-Royce combustor, the likes of which are in service on many aircraft today.

This project has concentrated on the development and the application of PIV to combustor aerodynamic measurements and has not required a new rig design. Instead, existing test facilities were utilised resulting in unavoidable overlap with the relevant chapters of Spencer (1998) and Griffiths (2000). Therefore, the test facilities are concisely described here, but should the reader require further detail they are referred to the aforementioned theses.

2.1 Water Analogy Facility

The previous chapter detailed several isothermal combustor experiments which utilised water as the working fluid. This practice is both common and feasible as the typical Mach number at combustor inlet is less than 0.30, hence compressibility effects can be ignored. Using water and matching typical Reynolds numbers of equivalent airflow experiments has the benefit that the velocities are significantly reduced, whilst maintaining the same fluid dynamic characteristics. The PIV system has a limited temporal resolution, and without these lower velocities, the study of space-time correlations (chapter 4) would be almost impossible. Additionally, PIV (and LDA) requires the flow to be seeded with suitable 'tracer' particles, a task more easily achieved in water than air.
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The water analogy rig is the same facility as that used by Spencer (1998) in his LDA experimental work. It is a vertically flowing rig which uses gravity as the driving force, thus maintaining a highly stable delivery. The test rig working section is pictured in figure 2.1 with the PIV camera and laser in place. Figure 2.2 shows a schematic of the rig layout and closed loop delivery system, illustrating how the water is delivered from a sump tank to the header tank. As the pump is designed to operate at a single delivery rate, a bypass valve is incorporated on the header tank feed line, returning excess mass flow to the sump tank whilst maintaining a stable delivery. The actual total mass flow through the rig is set by the outlet valves. Overflow pipes on the header tank mean that a constant water level (and hence driving pressure) can be maintained. The vertically flowing part of the rig comprises two concentric tubes, resulting in the core and annulus flows. Admission ports are located in the core/annulus wall in the test section allowing the annular flow to penetrate the core, as is the case in real combustors. The test section is designed such that the core pipe can be quickly and easily removed and replaced, allowing alternate sections to be incorporated with different admission port geometry. In addition to the valves controlling the core and annulus outlet mass flows, an inlet valve is located at entry to the core pipe, permitting the core/annulus inlet mass flow split to be
Experimental Facilities

varied. Flow entry to the working section from the header tank passes through straightener bundles to prevent any residual swirl passing downstream to the test section. The core flow develops over a distance of 18 core diameters prior to the test section, and the annulus flow over 53 annulus heights. The annulus flow can be considered as fully developed as it has passed through a distance of greater than 40 annulus heights (Schlichting, 1968), however the core flow may not. Nevertheless the core recirculation region is a highly turbulent flow so this is not thought to significantly affect the validity of the investigation.

Fig. 2.2 Water Analogy Facility Schematic Description (Spencer, 1998)

Figure 2.3 shows the test section, which is 600mm long. Each pipe is held concentric to the other by sets of NACA 0015 struts at the test section inlet and outlet. The test section can rotate, enabling different azimuthal planes to be studied without the need to move the laser and associated equipment. A square Perspex jacket containing water surrounds the test section. This reduces refraction effects and is standard practice in accordance with Bicen (1981). Outlet mass flow is calculated from the measured pressure differential across orifice plates (manufactured to BS1042) located in the return pipes. Inlet mass flow is calculated from the recorded inlet velocity profiles.
Experimental Facilities

Fig 2.3 Test Section Geometry (Spencer, 1998)
Experimental Facilities

Four different core test sections were available in the investigation: a blank (straight through) section, one with plain circular admission ports, the third with D shaped ports, and fourth with chuted ports. The core sections with admission ports present are shown in figure 2.3, each containing six equally spaced ports. Although data were collected from all four test sections only results from the blank and plain sections are presented in this document.

![Diagram of an annular test section with ports](image)

**Fig 2.4 Double Faced Step Geometry**

The blank test section separates the core and annular flows completely, and was used for two purposes: Firstly to check overall inlet and outlet mass flow balance (see section 2.1.4) and secondly to study the flow over a step during a study on the limitations of PIV in turbulent flows (see chapter 6). The latter was achieved by attaching a circumferential hoop of square cross section to the inner annulus wall (figure 2.4), thus emulating a double faced step flow. The nature of the annular gap meant that it is essentially simulating a two dimensional problem without the complications of end wall effects, and therefore ideal for this investigation. Despite the simplicity of the geometry, step flows incorporate a wide range of fluid dynamic phenomena including strong shear layers, flow reversal, and regions of recirculation. It is for these reasons that such flows are widely utilised as benchmark test cases, and the configuration was extensively used in studying the convergence of flow statistics with sample size (see chapter 3), and the effect of PIV
interrogation cell size in chapter 6. With respect to the plain port set-up, it was also used to study the flow in the wake of cylinders in the upstream annulus. Data from this setup was utilised in the assessment of sub grid filtering effects (chapter 6). Six 20mm diameter cylinders were positioned 60mm upstream of the port row and in line with the port centrelines (figure 2.5).

![Fig 2.5 Cylinders Upstream of Ports](image)

The test facility co-ordinate system is cylindrical polar, and is depicted in Figure 2.6. Also included on the diagram are the five mass flows denoted by core inlet (c), annulus inlet (an), jet (j), core outlet (o) and annulus bleed (b). The axial (streamwise direction) position is denoted by \( x \), the radial position (distance in perpendicular direction from the centreline) by \( r \), and the azimuthal angle (equivalent to core section rotation) by \( \theta \). Also indicated on the diagram are 'PIV co-ordinates', included to highlight the fact that obtaining true circumferential planes was not possible due to the planar nature of the PIV technique and, instead, 'pseudo-circumferential' (a term that will be utilised throughout this document) data were acquired in the \( x-z \) plane. In later chapters, it becomes necessary to use subscripts to define axial components in the form \( x_i \), \( R_{ij} \), and \( L_{ij} \). The subscripts \( i, j \) and \( k \) are not directions in space, but can assume numerical values of 1, 2 or 3, whereby 1 refers to the axial or streamwise direction (\( x \)), 2 is the normal direction (\( r \) or \( y \)), and 3 is usually the out of plane transverse direction (\( z \)).
Fig. 2.6 Test Section Co-ordinate System (Spencer, 1998; modified)

The mass flow rates through the rig are set by first calculating the desired mass flows, and hence pressure drops across the outlet orifice plates, using the calculation methodology in BS1042. This dictates the overall mass flow rate through the rig and means that the inlet flow can be adjusted (via the header inlet valves) to match the outlet and hence maintain a constant overflow from the header. Knowing the required core and annulus inlet (and obviously the jet) mass flow rates means that the core inlet valve can then be adjusted to achieve the required mass flow splits. Adjustment of the core inlet valve position is an iterative process, often requiring small adjustments to the outlet valves to maintain their correct mass flow rates. During the adjustment stage, inlet time-averaged velocity profiles are calculated from a relatively small number of PIV velocity vector fields. When the required mass flow rates are achieved, a greater number of PIV vector fields are acquired to accurately record the inlet flow conditions. To make certain the flow through the rig was well established and not varying with time, the rig was run...
for at least one hour prior to testing. This ensured that the water temperature was steady, that the trapped air could escape from the rig test section, and that the seeding was homogeneously distributed in the working fluid. The steady rig water temperature was always slightly higher than room temperature at 25±2°C due to the heating action of the pump.

2.1.1 Mass Flow Rate Calculation

The inlet mass flow rates are calculated from the integral of the velocity profiles:

\[ \dot{m} = \int_{\text{area}} \rho UdA = \rho \pi \int_{0}^{\theta} \frac{D}{2} \frac{\rho r}{2} dr \]

The equation indicates that velocity data at any radius is the average of two readings at 0°=0° and 0°=180°. Due to the 6 NACA0015 struts upstream of the test section in the annulus, there are wakes behind each aerofoil. This was considered in detail by Spencer (1998) who found that velocity measurements taken in between struts would be 1.02 times higher than the real average velocity, and that in the wake of a strut, the measurement would be 0.94 times the real velocity. This was taken into account in the calculated inlet mass flow rates.

The outlet mass flows were calculated from the pressure differential across the orifice plates in the outlet pipes. Both orifice plates are manufactured to BS1042 and have outer diameters of 51.0mm (equal to the outlet pipe diameters), and inner diameters of 35.2mm, and 27.5mm for the core and annulus respectively. BS1042 provides a series of equations to calculate the mass flow rate. This starts with an estimate for the approach flow pipe Reynolds number, a figure that can be reasonably estimated based on typical mass flow rates. This, together with the recorded pressure differential reading, allows an initial mass flow rate to be calculated, which in turn gives a more accurate Reynolds number, and allows further iteration. This iterative process was carried out within Microsoft Excel to give an accurate mass flow rate. Note that for manometer pressure drops of <1.5 inches of water, the accuracy of the readings is seriously reduced, although this was only of concern for low annulus bleed ratios of less than 20%, and hence was not an issue for the majority of data obtained.
2.1.2 Flow Parameters

The mass flow splits through the test facility are dependant only on the jet to cross-flow (core velocity) ratio, R, and the proportion of annulus bleed flow, B which are defined as follows:

\[ R = \frac{U_j}{U_c} \]
\[ B = \frac{m_b}{m_{an}} \]

The values used for R and B for the primary and dilution configurations were chosen to match those used by Spencer (1998) in order to be able to compare the data acquired from the LDA and PIV techniques. Spencer based his selection of R and B on typical (can type) combustor values, which although slightly dated – particularly for primary port flows – still result in flow topology that encompasses the important fluid dynamic phenomena of the combustor. Supplementary jet to cross flow and bleed ratios were also studied in the project, and are detailed in chapter 7. Jet Reynolds numbers quoted in the results chapter use the average jet velocity and port diameter, and flow conditions were chosen so that the minimum jet Reynolds number was \(2 \times 10^4\).

2.1.3 Refraction Effects

The effects of refraction were minimised (but not eliminated) by the use of the square water jacket. The effectiveness of the set-up was assessed using the image distortion and correction algorithm within the PIV software. The procedure involves acquiring an image of a calibration plate at the measurement plane location, which the software then uses to calculate a transformation matrix, thus allowing distorted images to be corrected prior to processing the velocity vector fields. The calibration plate consists of a regularly spaced arrangement of cross hair symbols, and is illuminated using domestic lighting for image capture. The user specifies the actual cross hair spacing in the measurement plane, and the software compares those to the (distorted) positions of the cross hairs actually recorded. The software reported that the level of distortion in this case was negligible, and manually examining the recorded image (figure 2.7) confirms the lack of visible distortion.
Fig 2.7: Recorded Image of The Calibration Plate Submerged in the Test Section

The calibration plate shown was produced in AutoCAD to give cross hairs at 3mm spacing, and then laminated to provide a water proof seal. The cross hair locations were estimated as being accurate to within ±0.1mm after the grid was laminated. The algorithm successfully located all cross hairs during the process, and only became confused in the core-annulus wall region, where refraction/reflection effects at the boundary makes it appear as though there is part of a calibration plate within the Perspex wall. This indicated that careful masking of the wall regions when processing the vectors would be required to avoid spurious data being reported there. Note that it appears as though the walls are thicker than 5mm in figure 2.7. This is because the cross-hair lines are 2mm in length, and there is approximately 2.5mm of clear (transparent) laminate between the edge of the plate itself and the wall to ensure the seal remains water-proof. The result is that the distance between the centre of the cross hairs closest to the core outer wall and inner annulus wall is 12mm, causing the wall to appear thicker. This also means that the centre of the closest cross hair from any wall is 3.5mm, and this is hence where the calibration matrix ceases. Therefore the software based procedure cannot
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detect whether distortion is important within this near wall region. According to the ray-traces of Spencer (1998), noticeable distortion only occurs within approximately 2mm of concave walls. Therefore as we cannot use the distortion correction algorithm to account for this, care must be exercised when scrutinising boundary layer profiles close to concave walls.

![Fig. 2.8 Illustration of LDA Beam Refraction for Streamwise Velocity](image)

In terms of the LDA measurements, the laser beams are refracted at the fluid-wall boundary, and it was therefore necessary to take account of this. The effect of refraction can change the position, orientation and fringe spacing of the control volume. Refraction of the beam path through the water jacket, annulus outer wall, and core-annulus wall is illustrated in figure 2.8 for the simplest case of the beam orientation for axial velocity measurement (where the refraction is exaggerated for illustrative purposes). The result is simply that the control volume is shifted a greater distance than the laser traverse indicates, due to the reduced effective beam half angle. In the case of axial velocity measurements no corrections are necessary to the fringe spacing or recorded velocity because the photomultiplier cancels out this effect by viewing through the same media interfaces. The streamwise orientation was that used for the vast majority of LDA data acquired, and simple application of Snell’s law enabled positional corrections to be made. Calculation gave the factors \( \frac{x_{cv}}{x_{traverse}} \) where \( cv = \) control volume as 1.333 for the core,
and 1.345 for the annulus, assuming that the datum is zero at the centreline, the slight difference being due to the presence of the annulus-core wall. For measurements in other orientations (radial and circumferential velocity measurements) where the refraction is complicated by the curved boundaries and fact that each beam can be refracted by a different angle, the procedures outlined by Spencer (1998) are followed, in which the equations presented by Bicen (1981) and Boadway and Karahan (1981) are adapted.

2.1.4 Mass Flow Rate Calibration
In order to assess the overall mass flow performance of the test facility, and ability of the orifice plates to give the correct mass flow rates, a set of calibration data were acquired for the test facility. This comprised both LDA traverse data, PIV velocity data, and orifice plate readings. To do this, the core pipe of the test section was replaced with the blank section, thus isolating the core and annulus flows. Figure 2.9 shows that the LDA and PIV data are in good agreement, and the only slight differences occur in the near wall regions, where the LDA control volume comes into contact with the wall. The agreement gives confidence in using either technique to derive the mass flow rates using the equation quoted earlier. The data also shows that the flow is symmetric about the centreline axis.

![Fig. 2.9 Comparison of LDA and PIV Velocity Profiles](image)
A comparison of the inlet mass flow rates calculated from LDA and PIV velocity profiles was made with the outlet mass flows from the orifice pressure differential readings. Figure 2.10 shows data exhibiting the expected linear relationship between inlet and outlet mass flow rates with a low spread of data, the standard deviation being less than 1%. The gradient of the slope is also almost exactly equal to 1 as would be expected, validating the mass flow measurement methodologies. The calibration data shows that the mass flow splits for the data obtained were determined to an accuracy of within ±2.0% of the desired values (to a confidence level of 95%).

2.2 VULCAN Phase 5 Combustor Airflow Rig

This test facility was defined and built during the LDA based investigation of Griffiths (2000), in which a comprehensive description beyond the scope of the one contained in this document is provided. The test facility was designed with two primary aims in mind; i) it must be representative of a modern combustion system, and ii) it must provide excellent optical access for laser based measurement techniques, necessitating the use of a sector design.
A 45° flame tube sector was obtained from an existing Rolls-Royce Phase 5 combuster, the likes of which are incorporated within the Rolls-Royce Trent series of large turbofan engines. Its design is typical of modern combustion systems and is shown in figure 2.11. Utilising an actual production flame tube meant that various detailed changes to the aerodynamic design, in terms of the Perspex casing which contained the metal flame tube, were required.

The co-ordinate system used for the PIV measurements is shown in figures 2.12 and 2.16. The x direction is along the flame tube axis, with its origin at the combustor heat shield. The radial axis (r) extends from the imaginary centre of the engine, but for convenience has its datum location on the flame tube centreline; hence negative r values indicate the inner portion of the flame tube. The azimuthal location is 0 with its datum on the central fuel injector centreline. However, due to the planar nature of the PIV measurement system, true (x-0) and (r-0) planes are not possible. Therefore, like the water rig co-ordinate system, a local pseudo-circumferential position is indicated by \( z \), thus allowing the use of Cartesian co-ordinates. The ‘local’ term refers to the z axis only, because for each measurement, its datum is the most appropriate geometric feature, e.g. port centreline. This is illustrated in figure 2.21.
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2.2.1 The Test Facility

In the engine geometry, metal annulus skirts link the flame tube to the casings. In the test facility the Perspex casing closely replicates the outer annular walls and skirt. This transparent casing was necessary in order to provide sufficient optical access. Other significant changes compared to the production combustor include the following resulting in the test facility geometry shown in figure 2.12:

- The casing follows the line created by the baffle of the outer dump cavity.
- Omission of the mounting pin.
- Turbine Nozzle Guide Vanes (NGVs) and Compressor Outlet Guide Vanes (OGVs) were omitted.

The first two changes were necessary in practical construction terms, and in order to provide adequate measurement probe access. The OGVs were omitted so that any subsequent CFD calculations did not have to attempt to resolve the associated wakes. Inclusion of NGVs and OGVs would have also added significant complexity to the sector geometry.
The vertically mounted test facility is pictured in figure 2.13. Due to the complex nature of the outer casing design and the fact that the flame tube is metal, PIV measurements were restricted to within the flame tube. Although this would mean annulus-core correlations would be impossible, valuable information could still be gathered within the flame tube, and inferences made with respect to the annulus feed flow. PIV measurements would also provide explanations as to the mechanisms driving bimodal behaviour noted by Griffiths (2000) within the core.

The facility operates by drawing air from atmosphere via a large plenum situated above the facility, through the inlet ducting and into the test section. Three inlet ducts feed the test section, one for the main flow and two sidewall cooling airflow streams (included for the provision of future combusting experiments). Having passed through the test section, the air is drawn through the outlet ducting of 425mm length (with a 25% area reduction to avoid instability problems) and into the exhaust plenum via a fan and exhaust system. The main inlet duct transitions from a 200mm circular bell-mouth intake to the test section inlet over a distance of 1.80m. A gauze was placed just downstream of the inlet.
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Flare to generate turbulence, and a 0.9mm boundary layer trip wire was placed around the perimeter inlet, 150mm upstream of the inlet section. The sidewall inlets are rectangular flared designs, and all three inlets sit approximately 300mm proud of the inlet plenum floor to prevent ingestion of poor quality flow from the floor of the plenum. The seeding necessary for the PIV measurements was introduced just above the bell-mouth intake of the main inlet section, and provided in the form of a mist of low viscosity oil particles provided by a TSI six-jet atomiser. Griffiths used the same seeding system for his LDA experiments and found it to give particles of approximately 1-2µm diameter, with a sufficiently high data rate. Further details of the seeding can be found in Chapter 3.

Trent 700 engine values at take off conditions were used to define the test facility inner and outer annulus mass flow bleed rates of 12.14% and 3.94% respectively (expressed as a percentage of the inlet mass flow rate), which is removed through offtake slots downstream of the secondary ports in the inner and outer annuli. The mass flow bleed rates were dictated by metering plates located prior to the flow returning to the flame tube exhaust stream. The bleed system is shown in figure 2.14.

---

**Fig. 2.14 Bleed System (Griffiths, 2000)**
The flame tube has opposed primary and secondary ports (but no dilution ports), where both incorporate chuted geometry. The design of the Phase 5 combustor is such that the primary ports are located in line with, and mid way between the fuel injectors. The secondary ports are circumferentially offset half way between the primary ports, and 44mm (approximately 3 primary port diameters) downstream of them. An actual Phase 5 combustor contains 48 pairs of primary ports and the same number of secondary ports, and 24 fuel injector swirlers. The swirler itself comprises inner, outer and dome sections, all imposing clockwise rotation to the flow. The port sizes and estimated mass flow rates are given in table 2.1.

<table>
<thead>
<tr>
<th>Port</th>
<th>Diameter (mm)</th>
<th>% of flame tube flow</th>
<th>% of total inlet flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Primary</td>
<td>14.5</td>
<td>8.0</td>
<td>6.6</td>
</tr>
<tr>
<td>Outer Secondary</td>
<td>20.1</td>
<td>15.2</td>
<td>12.6</td>
</tr>
<tr>
<td>Inner Primary</td>
<td>13.1</td>
<td>6.4</td>
<td>5.3</td>
</tr>
<tr>
<td>Inner Secondary</td>
<td>16.8</td>
<td>10.8</td>
<td>9</td>
</tr>
</tbody>
</table>

Table 2.1 Port sizes and mass flow rates

In addition to the air which enters the flame tube via the swirler and admission ports, 10.2% enters via the cooling holes in the heat shield (as shown in figure 2.15), with the remainder entering via the effusion liner cooling system, which provides a thin film of cooler air to protect the liners from the hot combusting gases. The cooling holes are 0.6mm in diameter and spaced approximately 5mm apart axially, and 2.5mm circumferentially. It should be noted that the actual fuel injector galleries were sealed off during the testing, thereby preventing any mass flow entering via this route.
Experimental Facilities

Fig. 2.16 Circumferential Location of Primary ports (Griffiths, 2000)

Fig. 2.17 Circumferential Location of Secondary ports (Griffiths, 2000)

As the rig is a sector design, it was necessary to carry out measurements in the central sector of the three contained in the 45 degree test facility, as this was sufficiently isolated from significant sidewall effects. In addition to relatively small windows in the inner and outer casings, the sidewall windows provide excellent optical access to the entire flame.
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tube, and with air as the working fluid there are negligible refraction effects. The sidewalls are fed with cooling air in order to allow for future combusting experiments to be carried out, although none were completed during this programme. Significant efforts were made by Griffiths (2000) to ensure the sidewall cooling had minimal effect on the flow field. This resulted in the cooling air being provided via a slotted arrangement, and each sidewall receives the equivalent of 4.35% of the inlet mass flow via separate intake sections.

It was noted by Griffiths (2000) that the flame tube deviated slightly from the actual production design, probably due to a lack of hoop stresses that would be present on the full flame tube, causing the sector to have expanded slightly. Also, due to manufacturing tolerances, the circumferential location of the ports varied slightly from the actual design as shown in figures 2.15 and 2.16 and were therefore not exactly symmetric about the injector-swirler locations. Although these differences exist between the actual combustor design and test facility, the variations were not considered significant. It should also be recognised that some excursion from the design values may be present in production combustors due to manufacturing tolerances and thermal expansion during operation.

2.2.2 Rig Modification

During this project, it was necessary to make some adjustments to the test facility, in order to provide a means of introducing the PIV light sheet across the flame tube. Previous LDA measurements only required optical access for the beam itself (and collection via the photomultiplier), whereas in using the PIV system a means of delivering a light sheet would be necessary, in addition to suitable viewing access for the camera.

The objective was to use the PIV system to make measurements in all three physical planes; x-r, x-z, r-z. For the x-r and x-z planes, it was not feasible to mount the laser in the inlet or exhaust to introduce the light sheet directly, therefore the options available were to use a mirror placed somewhere in the exhaust ducting, or a fibre optic delivery system. The mirror option, despite providing a blockage to the exhaust was deemed the only possible cost effective solution. The mirror had to illuminate the x-r, and x-z planes within the flame tube at selected azimuthal and radial locations. This meant that it had to
be possible to translate it in the circumferential direction in order to provide a light sheet at the required azimuthal locations. It also had to be able to tilt it on its z axis in order to position the sheet at the necessary radial locations. The mirror therefore required multiple degrees of freedom.

Fig. 2.18 Mirror Mount Location (horizontal orientation)

Fig 2.19 Picture of Mirror in Test Facility (in physical vertical orientation)
The most suitable location was chosen as being within the exhaust of the flame tube, in line with the position where the annulus bleed flow is returned to the main exhaust flow as shown in figures 2.18 and 2.19. This position was chosen because, had the mirror been placed further downstream, the inner flame tube wall profile would have obscured the light sheet’s line of sight of the inner ports. The best means of fixing the mirror was deemed to be on a block mounted within the inner annulus bleed return section. The block had a profile to match the radius of the casing meaning that it could translate circumferentially along a constant radius. Spring loaded screws were used to hold the block at the desired azimuthal location. The mirror itself was attached to the block on a gimballed mount, which allowed the multitude of freedom necessary. The mirror selected was an elliptical polished Aluminium component providing 98% reflectance for the 532nm Nd:YAG pulse.
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There were a number of physical and geometrical constraints that posed further challenges: the minimum distance from the laser light sheet optics to the mirror was 90mm due to the Perspex walls of the casing, and a height restriction within the void through which the light sheet was introduced meant that the incident light sheet had a maximum effective divergence of 11.04°. This in turn resulted in a maximum illuminated FoV in the r-x measurement plane at the primary port using a flat mirror of only 40mm. A 40mm FoV would be ideal for studying individual port flow behaviour, but it would not allow a study of the instantaneous flows occurring between the opposing jets, which would be a major focus of this part of the project. The solution was to have a convex mirror manufactured that would increase the divergence of the light sheet as it was reflected, thus increasing the illuminated area. Ray trace calculations indicated that a curvature of 104mm equivalent radius would provide enough divergence to illuminate the entire flame tube height. The actual set up in operation is shown in figure 2.20. Although diverging the sheet in this manner would obviously degrade the quality of the

Fig. 2.21 Laser and Mirror Positioning for various Measurement Orientations
Experimental Facilities

light sheet, the data gathered using the convex mirror was only for the full height FoV, and despite the difficult set-up, the PIV data quality remained satisfactory. Where smaller FoVs were investigated the flat, elliptical mirror was used (as shown in figure 2.19), providing a higher quality light sheet. When collecting data for the r-z plane, the camera actually viewed the area of interest through the flat elliptical mirror; i.e. the position of the laser and camera were exchanged, and the object plane actually viewed through mirror. This worked well and only required adjustment of the relevant axis of the data to conform with the correct assigned directions, although the FoV was obviously limited by the mirror size. The various experimental setups for the three measurement planes are illustrated in figure 2.21.

In placing a mirror in the exhaust ducting, a blockage to the flow was evident. However, as already discussed, the location of the mirror provided the only feasible means of getting any PIV data from this test facility during the project, and therefore the fact that it created a blockage had to be accepted. The frontal area of the mirror and mount was calculated as being 4% of the total exhaust duct at that plane. Filippone (1999) states that for aeronautical testing blockages must be less than 5% to avoid affecting the inlet velocity condition, and therefore the overall effect of the presence of the mirror was predicted as being small. Its presence would certainly have little bearing on the inner annulus return flow as the metering plates provided a much greater upstream restriction to the mass flow. It should also be appreciated that the flow exiting the flame tube is highly turbulent, and therefore the upstream region of interest is unlikely to be affected at all. The affect of the presence of the mirror (and air screen) was assessed by comparing pressure distributions, where it was confirmed that its presence would have an insignificant effect on the region of interest (see section 2.2.3).

A secondary problem existed with the mirror with respect to the oil seeding used: the seeding would quickly contaminate the mirror and therefore considerably degrade the light sheet (or camera view) quality. This meant that some means of protecting the mirror from a rapid build up of seeding was necessary. This was provided by constructing a small bore copper air pipe feed to the upstream edge of the mirror. This pipe was fed from a separate compressed air line, and issued a screen of air across the face of the mirror through a slit in the copper pipe. This acted to deflect the oil seeding
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particles away from the mirror, slowing the build up of seeding to a satisfactory degree in order to be able to acquire good quality data. The air screen was driven at 20psi and the system was pictured in figure 2.19.

2.2.3 Calibration Data

In order to provide PIV data that could be compared with the LDA data collected by Griffiths (2000), it had to be ensured that the rig was operating as expected, and hence at the same conditions as those previous experiments. This was achieved by acquiring button hook traverse data at the centreline of the pre-diffuser inlet, and static pressure tapping data in the annuli, flame tube exhaust section, and across the bleed flow metering plates. The measurement locations were indicated in figure 2.12, and the button hook probe is pictured in figure 2.22. Further information on the use of button hook probes can be found in Carrotte (1993). The data-set, comprising flow information at the most sensitive locations, would provide sufficient confidence in the entire rig operation. Selected data was also acquired with the mirror in place within the flame tube exhaust section to ascertain the effect of the presence of the mirror (and air screen) upon aerodynamic performance. In all of the data acquired from this test facility, the rig was set on condition according to the inlet Mach number (at pre-diffuser centreline) of 0.1205±2%.

![Fig 2.22 - Button Hook Probe and Manual Traverse](image)

The pre-diffuser inlet traverse profile data (figure 2.23), normalised by inlet Mach Number, shows excellent agreement with the previous work. This is encouraging since the inlet ducting is slightly modified due to a revised test facility location. The boundary
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layer data also shows good agreement, although the data suggests that it was marginally thicker in previous measurements. However the shape parameter (table 2.2) is almost identical and indicates a fully turbulent boundary layer. Overall, the values are very close, and within experimental error.

1.2

\[ \delta \delta' \delta^* \delta \delta \]

Fig. 2.23 - Pre-Diffuser Inlet Profiles (XP1)

<table>
<thead>
<tr>
<th>Boundary Layer</th>
<th>( \delta ) [mm]</th>
<th>( \delta^* ) [mm]</th>
<th>( \theta ) [mm]</th>
<th>( H )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner</td>
<td>4.860 (4.642)</td>
<td>0.569 (0.662)</td>
<td>0.430 (0.494)</td>
<td>1.324 (1.341)</td>
</tr>
<tr>
<td>Outer</td>
<td>3.980 (5.181)</td>
<td>0.501 (0.573)</td>
<td>0.381 (0.437)</td>
<td>1.317 (1.312)</td>
</tr>
</tbody>
</table>

Table 2.2 – Boundary Layer Data (data from Griffiths in brackets)

The sidewall mass flow rates were assessed by measuring the pressure drop across baffle plates installed just below the sidewall inlet plane. Griffiths (2000) specified that the mass flow rate through each sidewall should be 4.35% of the main inlet mass flow rate, and measurements gave mass flow rates of 4.83% and 4.84% for the left and right hand inlets respectively. This mass flow rate was deemed adequately close to the previous setup.

In terms of the static pressure measurements in the annuli, good agreement between the data can be seen in figure 2.24 within the measurement region, but discrepancies are evident in the skewed inner annulus profile of the current data. The reason for this was elusive, despite a variety of remedies having been explored:
E. vperinuciital Facilities

- Complete examination of rig for upstream ‘trips’ or asymmetric flow supply.
- Altering metering plate mass flow by systematically opening/blocking metering holes.
- Removing extended inlet section
- Altering sidewall inlet baffle plates

The problem could not be solved, and pursuing a remedy had to be abandoned due to the timescales of the project. It can however be said that the discrepancy only occurs outside the measurement region, and hence will not invalidate the results to follow. Also, flow visualisation with cotton tufts indicated that there was no evidence of flow separation within the annuli to cause this discrepancy in pressure distribution, which had been a source of major concern in the work of Griffiths (2000).

**Fig. 2.24 XP2 static pressure measurements**

The pressure drop across the metering plates was recorded, and by using a $C_d$ of 0.94 (Griffiths, 2000) for the holes present, the mass flow rates across the plates were calculated. This is the same method as used by Griffiths and the mass flow across each portion is expressed as a percentage of the inlet mass flow. The quoted percentages in table 2.3 for each portion/section are calculated such that the value is that which would exist across the entire metering plate based on the pressure drop for that portion only. Again, previous data is shown in brackets. Differences in the outer annulus mass flow rates may be explained in part due to the hole size used in previous calculations. The
actual holes were 0.3mm larger (between 5% and 7%) than described in the work of Griffiths. Nevertheless, the agreement is acceptable.

<table>
<thead>
<tr>
<th>Exit</th>
<th>Mass Flow (% of inlet)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Outer (left)</td>
</tr>
<tr>
<td></td>
<td>Centre Sector</td>
</tr>
<tr>
<td></td>
<td>Outer (right)</td>
</tr>
<tr>
<td>Inner Annulus</td>
<td>13.29% (13.07%)</td>
</tr>
<tr>
<td></td>
<td>13.03% (12.96%)</td>
</tr>
<tr>
<td></td>
<td>13.66% (12.97%)</td>
</tr>
<tr>
<td>Outer Annulus</td>
<td>3.10% (3.91%)</td>
</tr>
<tr>
<td></td>
<td>3.40% (3.98%)</td>
</tr>
<tr>
<td></td>
<td>3.29% (3.89%)</td>
</tr>
</tbody>
</table>

Table 2.3 – Metering plate mass flow rates

Selected measurements were re-acquired with the mirror in place and air screen active. The inlet profile would be unaffected by the presence of the mirror, and therefore the traverse was not repeated, although with the mirror in place the fan had to be driven at a slightly higher speed to achieve the required inlet velocity. However the presence of the mirror would have zero impact on the actual inlet traverse. The sidewall mass flow rates were also unaffected by the addition of the mirror.

Figure 2.25 shows that the static pressure dropped by a consistent 5% across the measurement region in both the inner and outer annuli at X12 and XO2. This is not a large difference and should not affect the results to follow, especially as the effect of the mirror’s presence has actually been to make the static pressure distribution closer to that
measured by Griffiths (2000). The mass flow across the metering plates were only subtly affected by the mirror and are given in table 2.4.

<table>
<thead>
<tr>
<th>Exit</th>
<th>Mass Flow (% of inlet)</th>
<th>Centre Sector</th>
<th>Outer (right)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner Annulus</td>
<td>13.47% 13.29%</td>
<td>13.39% 13.03%</td>
<td>13.54% 13.66%</td>
</tr>
<tr>
<td>Outer Annulus</td>
<td>3.26% 3.10%</td>
<td>3.49% 3.40%</td>
<td>3.22% 3.29%</td>
</tr>
</tbody>
</table>

Table 2.4 Metering Plate Mass Flow Rates with (top row) and without mirror (bottom row)

In addition to the above, the static pressure distribution 50mm upstream of the mirror position in the core exhaust was compared to that without the mirror in place (figure 2.26). The difference with the mirror in place shows a less than 5% drop in static pressure across the measurement section.

![Fig. 2.26 Exhaust Outlet Static Pressure Distribution](image)

It had therefore been established that the overall rig operation was as expected with the exception of certain minor differences which will not have implications on the comparability of the data collected in this programme of testing. The rig has been successfully modified to be able to acquire PIV data using the Nd:YAG laser and CCD
camera via the addition of a mirror in the exhaust duct. The addition of the mirror (and air screen to keep the mirror clean during seeding) has resulted in a less than 5% change in all of the rig flow distribution characteristics. Therefore the overall impact on the comparability of the PIV data to be acquired, and that taken using LDA previously will be negligible.

2.3 PIV System

One of the first tasks of the project was to source the PIV equipment and software. A conscious decision was made to purchase a complete proprietary system rather than attempt an in-house build, as this would be a more time-effective approach and would enable greater focus on the task of applying a PIV system to collect combustor aerodynamics data.

2.3.1 System Selection

In choosing a PIV system a number of factors had to be considered in terms of cost, required hardware specifications, likely experimental conditions, and software features. Actual specification in terms of laser power and camera CCD size were based on supplier recommendations regarding the latest available equipment, cost limitations, and extrapolation of trends in hardware utilised in similar experiments (e.g. examples in Raffel et al, 1998). The following detailed criteria were derived (note that the terminology and technical operation of the PIV system is explained further in Chapter 3):

- **Laser Power**: Ability to adequately illuminate 1µm particles in an air flow, when viewing a Field of View (FoV) of 0.14m x 0.14m in ideal conditions. This necessitated a 50mJ power output. A selection of diverging optical lenses would also be required to enable the illumination of a variety of Field of Views.

- **Camera Resolution**: Sufficient resolution to be able to analyse PIV data for a field of view equal to the full diameter of the water analogy rig or full height of the VULCAN airflow rig (140mm and 120mm respectively); the largest FoVs encountered in the project. Therefore a 1000 x 1000 pixel CCD Camera would be required.
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- **Temporal Resolution**: Maximum possible temporal resolution within cost limits resulted in a requirement for a 15Hz camera rate in double frame (cross-correlation) mode, and an associated laser repetition rate. Camera sensitivity has to be compromised (i.e. 8 bit rather than 12 bit) for temporal resolution – the latter being considered more important to be able to investigate space-time cross-correlations.

- **Dynamic Resolution**: Ability to resolve flows up to approximately 40m/s which would be experienced in the airflow sector rig, necessitating a theoretical minimum inter-frame time of 10µs. However this would also be dependent on the magnitude of the out of plane motion. A dual head pulsed laser and camera with the associated abilities (i.e. inter-line transfer) would thus be required.

- **Software**: High level of traceability within the software, allowing the user to define all of the acquisition and processing parameters. Software should also be user friendly and include the ability to modify and include in-house algorithms, thus enhancing flexibility. The ability of the software to export and import data from other packages such as TecPlot would also be important. The software (and hardware) should be easily expanded to incorporate simultaneous Laser Induced Fluorescence (LIF) and PIV measurements, high speed PIV applications, and stereoscopic PIV, all envisaged as future projects.

- **Computer**: To be included in cost, and to comprise state-of-the-art processing capabilities and large memory (RAM) capacity, to enable the capture of many images in single test runs. A large hard drive (>30GB) would also be required for storage of data.

- **Hardware**: Compact, durable and easy to move to different locations.

- **Support**: High level of support from supplying company in terms of both repairs and software support. The company should be well established with sufficient references of past sales.

Five suppliers were considered in the search for a system, and based on it meeting the criteria most closely, the system offered by LaVision GmbH was chosen.
2.3.2 The LaVision PIV System

A basic illustration of the PIV set-up is shown in Figure 2.27, and Table 2.5 gives the technical specification of the equipment. Particle Image Velocimetry theory is explained in detail in the following chapter.

![PIV System Arrangement](image)

**Fig. 2.27 PIV System Arrangement**

(note external trigger not used in this investigation – internal rate generation employed)

<table>
<thead>
<tr>
<th>Camera</th>
<th>Kodak Megaplus E.S. 1.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resolution</td>
<td>1000 x 1000 Pixels @ 8bit dynamic range</td>
</tr>
<tr>
<td>Frame Rate</td>
<td>15Hz (double frame)</td>
</tr>
<tr>
<td>CCD Size</td>
<td>9mm x 9mm</td>
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</table>

<table>
<thead>
<tr>
<th>Lens</th>
<th>Nikon</th>
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</thead>
<tbody>
<tr>
<td>Focal length, z</td>
<td>50mm (24mm &amp; 60mm subsequently acquired)</td>
</tr>
<tr>
<td>F-number, F#</td>
<td>2.0 to 22</td>
</tr>
<tr>
<td>Magnification, M</td>
<td>1&lt;(M&lt;0.05)</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Laser</th>
<th>New Wave Solo Dual Head Pulsed Nd:YAG</th>
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</thead>
<tbody>
<tr>
<td>Power</td>
<td>50mJ (per laser)</td>
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<tr>
<td>Repetition Rate</td>
<td>(\leq18)Hz</td>
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<tr>
<td>Pulse Duration</td>
<td>6ns</td>
</tr>
<tr>
<td>Beam divergence</td>
<td>Selectable 6°, 12°, 30°, 60°</td>
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</table>

<table>
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<td>Memory</td>
<td>3GB</td>
</tr>
<tr>
<td>Storage</td>
<td>1 x 30GB SCSI HD</td>
</tr>
<tr>
<td></td>
<td>1 x 40GB IDE HD</td>
</tr>
<tr>
<td></td>
<td>1 x CD-RW</td>
</tr>
<tr>
<td>Software</td>
<td>DaVis v.6.03</td>
</tr>
</tbody>
</table>

**Table 2.5 PIV System Specification**
Experimental Facilities

2.4 LDA System

The LDA system was used as a comparison/validation technique for the PIV data during development of analysis techniques (Chapters 4, 5, 6) on the water analogy rig. The particular system employed is well proven and has been used for a number of previous studies (e.g. Spencer, 1998). It comprises a He-Ne 20mW laser and Dantec photomultiplier, and was operated in forward scatter mode to achieve data rates of up to 1.5kHz. The same 20µm diameter Polyamid seeding used during PIV waterflow testing was utilised during the acquisition of LDA data.

The laser and photomultiplier were rigidly connected to a traversing mechanism, meaning that the LDA control volume could be moved without the need to realign and refocus the photomultiplier. Digital position indicators allowed the control volume to be positioned within ±0.01mm in the x and z directions, and a mechanical indicator on the y traverse allowed positional accuracy of ±0.25mm. However, due to the nature of the rotating core section, movement in the yL direction was seldom required.

The LDA system is described in Table 2.6. For further detail on the LDA system, the reader is again referred to Spencer (1998).

<table>
<thead>
<tr>
<th>LDA System</th>
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</tr>
</thead>
<tbody>
<tr>
<td>Photo-Multiplier</td>
<td>TSI IFA550</td>
</tr>
<tr>
<td>Processor</td>
<td>ZECH Electronic 1400A</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>LDA Laser</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Helium-Neon</td>
</tr>
<tr>
<td>Power output</td>
<td>20mW</td>
</tr>
<tr>
<td>Wavelength, λ</td>
<td>632.8nm</td>
</tr>
<tr>
<td>Diameter of Beam at 1/e^2</td>
<td>0.68mm</td>
</tr>
<tr>
<td>Beam half angle, θ</td>
<td>5.53°</td>
</tr>
<tr>
<td>Fringe spacing, d_f</td>
<td>3.28μm</td>
</tr>
<tr>
<td>Number of fringes</td>
<td>46</td>
</tr>
<tr>
<td>Frequency shift, f_shift</td>
<td>0.40MHz (Typical)</td>
</tr>
<tr>
<td>Minor axis of control volume, x_{cv}</td>
<td>0.30mm</td>
</tr>
<tr>
<td>Major axis of control volume, x_{cv}</td>
<td>3.12mm</td>
</tr>
</tbody>
</table>

Table 2.6 LDA System Specification
2.5 Closure

The test facilities have been described in detail, together with calibration data which proves their operation matches those of previous works, ensuring that comparison of results will be fair and accurate. The novel modifications to the airflow facility in order to obtain PIV measurements have been detailed, and will allow collection of data previously impossible with LDA. The ethos behind the selection of the LaVision PIV system was detailed, and the system specification presented. The following chapter gives a more detailed description of PIV, and the optimisation procedures for successful measurements in practical engineering flow regimes.
Chapter 3

PARTICLE IMAGE VELOCIMETRY AND ITS OPTIMISATION
Particle Image Velocimetry and its Optimisation

The background and basic concept of the Particle Image Velocimetry technique was given in chapter 1, followed by a description of the physical arrangement and particular system to be used in this investigation in chapter 2. In this chapter the theory and mathematical description of PIV is discussed. As mooted in the introductory chapter, it is important to appreciate that the following descriptions given all refer to digital cross-correlation PIV using pulsed lasers, although brief comparisons will be made with photographic methods, auto-correlation and mechanically shuttered continuous wave lasers.

PIV is a complex measurement technique, with many parameters available to change and optimise, a fact that will become startling apparent in the following sub-sections. The first section outlines the PIV ‘process’ at its most basic level. This is followed by more detailed discussion of each step of this process; acquiring the raw image data, the calculation (or ‘processing’) of the quantitative velocity vector information, validation of the velocity information, and presentation and data analysis techniques. Optimisation of the whole PIV setup is then discussed followed by a description of the limitations of this measurement technique. The chapter concludes following a discussion of advanced PIV techniques which may be applied in future projects of this nature. Note that a discussion of the lack of optical distortion present with both test facilities was given in chapter 2, and therefore no further reference is made to this topic.

Although PIV techniques continue advancing at a significant rate, the basic concepts of cross-correlation PIV are well established having been developed over the past two decades (see review articles; Grant, 1997; Prasad, 2000; Samimy and Wernet, 2000). Therefore, the reader will find that the mathematical descriptions in this chapter are provided at a level required to comprehend the way in which PIV works. Complete mathematical derivations of every intricate algorithm are not included as they are beyond the scope of this project, and unnecessary in being able to understand the concepts presented. It should be remembered that the PIV system employed is a proprietary system, and substantial detail as to the technical background of it can be found in the relevant manual (LaVision, 2001). For further detailed reading on the subject of PIV, the reader is referred to Raffel et al (1998).
3.1 The PIV Process

Prior to the detailed descriptions, it is useful to describe the basic steps of collecting data via the PIV process:

Acquisition

This is the process of actually collecting the raw images and is arguably the most important step in the collection of data. It is widely accepted that a more productive use of one’s time is to spend two weeks setting up the experiment, rather than two weeks trying to make poor quality data look better than it really is by post-processing (Westerweel, 2001). In addition to accurately setting up the actual flow conditions of interest, which includes addition of the most suitable seeding to the flow, the PIV hardware must also be accurately set up. The light sheet illuminating the flow must be correctly and accurately positioned and the thickness of the sheet should be optimised for the experiment. The camera should be properly orientated and the focal parameters optimised to ensure the best quality image. Another very important parameter that must be optimised is the timing synchronisation and inter-frame time settings.

Processing

This is the method by which the velocity vector field is generated from the raw image data. Prior to the actual calculation, there is the option to pre-process the image by subtracting mean intensities or smoothing the image, however it is the general experience of the author that this should not be necessary in the majority of cases providing the physical set-up is optimised, a notion supported by Chan (2001) and Westerweel (2001). Nevertheless, where unavoidable background reflections are present, subtracting the background image (i.e. the illuminated FoV with no seeding present) can be a genuinely useful pre-processing tool that results in an increase in signal to noise ratio and hence data quality.

Once the experimentalist is content that the best possible physical set-up has been achieved, the way in which the vector field is calculated must be chosen and optimised. This is, in many ways related to the acquisition step, in terms of timing parameters and the size of the field of view. Choices to be made include vector grid density, and interrogation cell size and overlap.
Validation
This is the method by which the quality of the vector map is quantified. This step is often referred to as ‘Post-Processing’ – a term which is off putting for some experimentalists who think of this as the invention of new data, or ‘smoothing’ to make the vector field look better. A good validation procedure will only remove spurious vectors, and will use a suitable scheme to replace data voids where appropriate. This is an acceptable procedure providing the raw data is of the highest possible quality from the outset.

Analysis
This is the step where the vector field information is used to derive other quantities such as vorticity and turbulence statistics. One must at this stage appreciate how the data has been generated, and the limitations of the technique, before implementing calculation schemes which may be suitable for other data forms, but not for PIV.

Presentation
A vast amount of data is generated in a PIV experiment, with typically hundreds of instantaneous vector fields providing potentially important data. The challenge is therefore to extract the important points in a succinct manner, and the ability to do this is aided by the use of conditional averaging and decomposition of vector fields. Workers in the field are continually striving to extract more useful information from the data, an example of this being found in Adrian et al (2000).

3.2 Acquisition
Four fundamental aspects define the success of the image acquisition: the tracer particles, flow illumination, image recording parameters, and to a certain extent the timing parameters (although the overlap between the acquisition and processing stages is present for these issues). Note that it is assumed the physical test rig set-up and flow field of interest is accurately defined prior to the PIV acquisition considerations. This includes providing sufficient optical access.
3.2.1 Tracer Particles

The tracer particles within the flow field must achieve two objectives; they must accurately follow the flow, and they must scatter sufficient light to be accurately recorded. There have been many PIV experiments in conditions similar to those investigated here (see those in Raffel et al, 1998). Based upon those experiments, probable suitable seeding was selected as Polyamid (polystyrene) particles of 20µm diameter in the water based experiments, and atomised Shell Ondina oil (a low viscosity oil) particles of a maximum 2µm diameter for the airflow measurements, produced by the TSI six-jet atomiser described in chapter 2.

In any PIV experiment, the flow following ability of the tracer is critical to the success of the measurements attained, because the tracer displacement is measured in order to infer the fluid displacement. This is assessed in terms of two parameters: the particle response time (Elghobashi, 1994);

$$\tau_p = \frac{\rho_p d_p^2}{\rho 18\nu}$$

and the velocity lag (Raffel et al, 1998)

$$U_s = d_p^2 \frac{(\rho_p - \rho)}{18\mu} a$$

In the case of the water flow experiments, the Polyamid particles used are approximately neutrally buoyant and so the ratio of the particle density to fluid density is nearly unity, causing the velocity lag to be virtually zero. The oil particles, however, have a density somewhat greater than the surrounding air. The various fluid parameters and typical flow properties are given in table 3.1, and the suitability of the tracer particle in terms of their flow following characteristics are compared to the smallest of the turbulent scales, the Kolmogrov scale (obtained using the procedure outlined in chapter 5). In the table an estimate of the maximum acceleration has been made. This was done by assuming the maximum acceleration occurs at the point of jet impingement, where (for the purpose of this estimate) it is assumed that the fluid is brought to rest from its maximum velocity in 1µs. The minimum Kolmogrov scale also coincides with the impingement point, and therefore this point represents the worst case scenario. As can be seen, in both cases the
particle response time is slightly greater than the Kolmogorov time scale. However, in the vast majority of the field the response time is much lower than the Kolmogorov scale, as is ideally required. At the impingement point, the velocity of the particle will lag that of the fluid very slightly in the airflow investigation, but the acceleration is much lower in the remainder of the field and therefore the velocity lag is generally negligible.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Water Analogy Rig</th>
<th>VULCAN Airflow Rig</th>
</tr>
</thead>
<tbody>
<tr>
<td>Particle diameter, $d_p$ [μm]</td>
<td>20</td>
<td>2</td>
</tr>
<tr>
<td>Particle density, $\rho_p$ [kg/m³]</td>
<td>997.2</td>
<td>800</td>
</tr>
<tr>
<td>Fluid density, $\rho$ [kg/m³]</td>
<td>997.2</td>
<td>1.207</td>
</tr>
<tr>
<td>Fluid dynamic viscosity, $\mu$ [Ns/m²]</td>
<td>8.904x10⁻⁴</td>
<td>1.812x10⁻⁵</td>
</tr>
<tr>
<td>Fluid kinematic visc, $\nu$ [m²/s]</td>
<td>8.920x10⁻⁷</td>
<td>1.501x10⁻⁵</td>
</tr>
<tr>
<td>Maximum acceleration [m/s]</td>
<td>2000</td>
<td>30000</td>
</tr>
<tr>
<td>Particle velocity lag, $U_0$ [m/s]</td>
<td>0</td>
<td>0.294</td>
</tr>
<tr>
<td>Particle response time, $\tau_p$ [μs]</td>
<td>24.9</td>
<td>9.8</td>
</tr>
<tr>
<td>Kolmogorov Time Scale, $\tau_s$ [μs]</td>
<td>15</td>
<td>5</td>
</tr>
</tbody>
</table>

Table 3.1 Flow Properties and Particle Response and Velocity Lag

The second important characteristic of the tracer particles, their light scattering properties, is a massive area of physics. In basic terms, the scattering of light by small particles depends upon the ratio of the refractive index of the particle to that of the surrounding medium, the particle’s size and shape, and the observation angle relative to the incident light. For spherical particles with diameters larger than the wavelength of the incident light, Mie’s scattering theory can be applied (see van de Hulst, 1957). The theory gives formulae for the geometric and diffraction limited particle sizes which are discussed in section 3.2.3. Although the theory is accepted as accurate, the author found that the theoretical values were actually a relatively poor indication of the actual particle sizes observed in the image plane. This was most likely due to a number of reasons, including the use of a polarizer in the water rig experiments to reduce reflections and the fact that the reflected light from a particle is dependent upon the incident light energy; therefore the laser power settings would have a direct impact on the particle image sizes. Hence the suitability of the particles with respect to light scattering was mainly evaluated in a practical manner by simply observing the recorded images, and the quality of the generated vector data (in terms of peak locking).

It was found that the Polyamid particles in water produced images of 2-3 pixels diameter for typical fields of view (80mm) at optimum camera settings, and using 50% of the
available laser power. The oil particles in air also produced particle images of around 2 pixels at typical fields of view (40mm) due to their greater scattering and hence diffraction limited image, and the lack of polarizing lens present for the airflow experiments. As will be seen later, a particle diameter of 2 pixels is the optimum for digital PIV analysis (Westerweel, 1997).

In addition to the above considerations, a sufficient number of particles should be present in the fluid for recording (the term ‘sufficient’ being defined later), but at the same time not influencing the flow. Ideally the particles should also be homogeneously distributed. Approximately homogeneous distribution was easily achieved in the water analogy facility by admitting the seeding to the (re-circulating) flow approximately one hour prior to data acquisition, thus ensuring an approximately uniform tracer distribution. Homogeneous distribution was more difficult to achieve in the airflow facility, due to the high levels of turbulence, and complex geometry negotiated by the air carrying the seeding. Although some areas of higher or lower seeding concentrations were evident in the large FoV recordings in particular, the data quality remained sufficiently high. It should also be appreciated that due to the short time between the two frames of each double frame cross-correlation recording, although areas of higher and lower density seeding exist in the same FoV, the same seeding distribution exists in both frames. Therefore, providing the lowest density of seeding is sufficient, the data quality should remain acceptable.

3.2.2 Flow Illumination

Lasers are widely used in PIV, because of their ability to emit monochromatic light with high energy density, which can easily be bundled into thin light sheets for illuminating and recording the tracer particles (Raffel et al, 1998). Detailed mathematical description of lasers emission theory is not given here, but many items are available on the subject in the literature should the reader be interested (e.g. Svelto, 1998). A pulsed laser is used in this PIV work, and is the type of common choice today. Unlike Continuous Wave (CW) laser which emit a constant beam (as would be required in LDA), pulsed lasers allow the energy to build within the pump cavity before a device called the Q-switch ‘opens’, allowing a very short (≈6ns) burst of high intensity laser light to emit. In this way, the energy output from the laser in a single ‘flash’ is much higher than achievable by the
equivalent CW units. As a consequence the pulsed laser units can be made smaller and are far more efficient for the PIV role.

The type of pulsed laser employed is a Neodym Yttrium Aluminium Garnet (Nd:YAG), and is a solid state laser in which the beam is generated by Nd\textsuperscript{3+} ions, which are incorporated in the yttrium-aluminium-garnet (YAG) crystals. The YAG crystal is pumped with energy from white light flash lamps within the laser head unit. Nd:YAG lasers are currently the most widely used in PIV applications because of their robustness, high amplification, and good mechanical and thermal properties. In most PIV applications the fundamental wavelength output of the Nd:YAG laser is frequency doubled to 532\textmu m (green light) because the standard output is in the infra-red spectrum. In all PIV specific pulsed lasers, there are actually two lasers within the head unit. This allows the separation time between pulses to be infinitely small in theory (although practically \(\Delta t>0.02\mu s\) to avoid overlap of the pulses), despite each laser operating at typically less than 20Hz. A schematic of the laser head is shown in figure 3.1.

![Fig. 3.1 Laser Head Schematic (Raffle et al, 1998)](image)

The beam delivered from the laser is shaped into a light sheet via spherical (diverging) and cylindrical lenses. The spherical lens first focuses the beam, and the cylindrical lens then forms the light sheet. Adjusting the distance between the spherical and cylindrical lenses changes the position of the cylindrical lens relative to the focal point of the beam,
thus changing the thickness of the sheet at a given distance from the lens. The thickness of the sheet is normally chosen to be as small as possible. This was practically evaluated by observing the sheet at very low laser powers. A relatively thin sheet (~1mm) helps to eliminate out of plane motion being recorded, although some instances where substantial out of plane motion is present in the area of interest require a thicker sheet (and camera depth of field) in order to retain any useful tracer motion within the inter-frame time. The cylindrical lens can be changed to alter the divergence, and hence illuminated field of view. A collimator was not used to create a constant height light sheet, due to the increased complexity of such an optical set-up, and the fact that the equipment available provided a sufficiently high quality light sheet.

3.2.3 Recording

The illuminated flow field is recorded onto an electronic Charged Coupled Device (CCD) array within the camera housing. The CCD array consists of a regular arrangement of light sensitive ‘pixels’. These pixels act in the reverse manner to those found on a computer screen monitor – instead of emitting light, the CCD pixels register light intensity. This is achieved by the image being focussed onto the CCD array, and the individual pixels collecting the light (photons) in the form of charged electrons. The charge represents the digital intensity at each pixel, and translates into the raw data image plane, which is essentially a digitised (and rescaled) version of the real object plane. Diagrams in figure 3.2 illustrate how the CCD works. CCD arrays are relatively robust but can suffer from elevated intensity exposure. This can lead to blooming, which is the overspill of electrons from one pixel to the adjacent one, artificially making it appear as though the adjacent pixel has detected some light. High exposure can also ‘burn-out’ the CCD sensor, resulting in void pixels in the array and reducing data quality. Therefore care has to be taken to avoid the use of unnecessarily high laser power settings.

The Kodak E.S. 1.0 camera used has a 1k x 1k inter-line transfer CCD, which provides a fast form of moving the image data (see figure 3.2) and allows the short inter-frame times necessary. Upon the illumination of the first ‘frame’ (frame A) of illumination, the active light-sensitive columns of the CCD are used to capture the image. Between illuminations that data is transferred to the adjacent columns, making the active column ready for the capture of Frame B. Data in the ‘passive’ column is transferred to a PCI frame-grabber.
card in the PC whilst the active column collects the current illuminated image. In this way the camera can operate in ‘double frame, single exposure mode’, as is required for the cross correlation algorithm to perform. Because the shift of the active image data to the adjacent passive column is very efficient, the CCD camera can work quickly, allowing very short inter-frame times between each of the frame pair, thus working in unison with the dual head pulsed laser.

![CCD Sensor Operation](image)

**Fig. 3.2** CCD Sensor Operation (Raffel et al, 1998)

It should be noted that in modern PIV systems, it is the (image) data transfer rate which usually limits the temporal resolution. Each double frame (1k x 2k) 8 bit image occupies approximately 4Mbytes of RAM (although this is significantly compressed when stored on the hard drive), and although cameras and lasers are available which operate at higher frequencies than 15Hz, it is the act of getting that data into the PC memory where the bottleneck occurs. Cameras and frame grabber boards have been developed with their own dedicated memory in order to attempt to over-come this problem in high speed (kHz) PIV systems, which are now being successfully applied (e.g. van Doorne *et al*, 2003)
Accurate recording of the flow field requires careful selection of the camera lens, and consideration of the camera parameters. Nikon 24mm (wide angle), 50mm, and 60mm focal length lenses were available with f-numbers ranging from 1.8 to 22. These lenses are high quality and ensure negligible aberrations are present. Choosing the lens object distance and f-number are some of the most important factors in the physical set-up.

In carrying out the experiment, one of the first questions must be ‘what is the Field of View (FoV) to be studied within the flowfield?’ The physical size of the required FoV must be assessed, and then the question posed as to whether that field of view can be accurately imaged onto the CCD array. This hinges on the particle image diameter, \(d_t\), discussed earlier and given by (Raffel et al, 1998):

\[
d_t = \sqrt{M^2d_p^2 + d_{diff}^2}
\]

Where \(M\) is the magnification of the camera and \(d_p\) is the physical particle diameter. The magnification, \(M\), is given simply by the image size (CCD array size) divided by the actual FoV, and the diffraction limited image diameter, \(d_{diff}\), is given by:

\[
d_{diff} = 2.44(1 + M)f^n\lambda
\]

Where the f-number (\(f^n\)) is set on the camera and is defined as the ratio of the focal length to the diameter of the aperture, and \(\lambda\) is the wavelength of the incident light (532nm). Given that there remains only two unknowns, a graph can be produced which acts as a quick reference to determine particle image diameter. Figure 3.3 shows the relevant data for the Polyamid particles used in the water analogy facility. Note again that the optimum particle image diameter is 2 pixels (explained later) and that the physical size of each CCD pixel is 9\(\mu\)m. As discussed earlier, it is the authors experience that the actual particle size recorded is somewhat larger than the theoretical sizes given by the above formulae, and normally larger by a factor approximately equal to 3. This was assumed to be due to the effect of the polarizer where utilised, and the effect of the incident laser light energy. It should also be appreciated that the geometric size dominates the diffraction limited image in the water based experiments due to the relatively large particles used. It is worth noting at this stage that practical f-number settings for PIV recordings are between 2.8 and 5.6.
The general message here is that the particle image size will be too small if the FoV is too large (and hence the magnification too small). This may necessitate splitting the FoV into smaller regions for capture and analysis, or using larger particles, however the latter solution must be used with caution and the particle response parameters considered. The particle image size can be artificially increased by slightly defocusing the image, but this results in a blurred particle image and is only advised if absolutely necessary and all other avenues have been explored. It was therefore avoided in all of the experiments carried out in this project.

Another inter-dependency on the parameters mentioned above is the depth of field. This is the line-of-sight range over which the image remains in focus, i.e. the object focal depth. All particles remain in focus when the light sheet thickness is smaller than focal depth, and so standard practice is to choose a depth of field slightly larger than the light sheet thickness (Adrian, 1991). The depth of field, $\delta z$ is given by the following equation (Raffel et al, 1998):

$$\delta z = 4(1 + M^{-1})^2 f^{\#2}\lambda$$

Therefore, another reference graph can be constructed as in figure 3.4:
As the light sheet is typically of 1mm thickness, and magnification often around 0.2, it is straightforward to see why the typical f-number is 4.0.

The recording set-up is clearly determined by a number of inter-dependant parameters, and although the equations given can be utilised to predict the optimum set-up, often hands-on trial and error was necessary to completely optimise the configuration.

### 3.2.4 Timing

Within PIV there are two main timing parameters; the data acquisition rate (temporal resolution) and the inter-frame time. As discussed earlier, the maximum data acquisition rate is determined by the camera and laser repetition rates, and is 15Hz for this set-up; where 15Hz is equal to the reciprocal of the sampling period (=1/ΔT). This should be borne in mind when considering the fluid motion between successive measurements. For example, in the water flow measurements for the primary setup (chapter 7), the typical convection velocities of 0.5m/s mean that a ‘packet’ of fluid will translate approx 33mm between instantaneous measurements.

The inter-frame time, Δt, is set by the user and is extremely important. The velocity of the fluid is calculated from the recorded tracer displacement within the known time
separation. Hence, some fluid dynamic properties within the FoV must be known in order to select a suitable $\Delta t$. A timing sequence illustrating the way in which the camera and laser work in relation to the acquisition rate and inter-frame time is given in figure 3.5.

The so-called 'one quarter rule' of Keane and Adrian (1990) states that the displacement magnitude of each particle group, $d_s$, within each interrogation cell should be less than one quarter of the cell dimension ($\Delta X=\Delta Y$); i.e. if the interrogation cell size is 32x32 pixels, then $d_s$ should be less than 8 pixels. Therefore, based on some knowledge of the expected maximum velocities within the flow field, $\Delta t$ should be chosen to give the appropriate optimum pixel displacement. Thus, another inter-dependency exists between the magnification and this inter-frame time parameter, $\Delta t$. Although the pixel displacement should be chosen to be 'less than' one quarter of $\Delta X$, the smaller the displacement, the greater the relative error on the calculated displacement becomes. This is because the particle position can only be determined to a finite accuracy. Hence, if the positional accuracy is calculated with a typical tolerance of less than ±0.1 pixels (Surmann, 2003), the relative accuracy of the calculated displacement for a particle shift of 2 pixels is ±5%, whereas for a shift of 8 pixels it is ±1.25%. Therefore there are advantages to maximising $\Delta t$ in order to maximise $d_s$, and hence the calculated displacement accuracy.

However, the effect of dynamic averaging must also be considered. This relates to the degree to which the real particle motion is averaged as a result of the time between

---

Fig. 3.5 PIV Timing Sequence
recorded displacements. This phenomenon is illustrated below in figure 3.6, and shows that dynamic averaging would result in reduced recorded turbulence levels, as the effect is to smooth the recorded displacement. Therefore, in addition to optimising the particle shift (and hence inter-frame time) relative to the interrogation cell size, it must also be optimised whilst considering the fluid dynamic properties of the flow - i.e. by making the inter-frame time equal to or less than the Kolmogorov time scales present. It also has to be borne in mind that the flow field will probably contain a range of velocities, and therefore the inter-frame time must be suitable across the entire field of view. Clearly there are a number of trade-offs and compromises to be made, and often the theoretical ideal limits have to be stretched in order to provide the best possible data quality.

![Illustration of Dynamic Averaging](image)

**Fig. 3.6 Illustration of Dynamic Averaging (for an arbitrary flow velocity)**

The above has discussed in-plane (two-dimensional) motion, without regard to out-of-plane motion. If there is significant out-of-plane motion such as those in swirling flows (e.g. Midgley *et al.*, 2004), the following equation should be used to determine an acceptable limit of out-of-plane motion (*Raffel et al.*, 1998):

\[
\frac{W\Delta t}{\delta z} \leq 0.3
\]

Therefore the out-of-plane motion should be less than 30% of the light sheet thickness. This limitation on the inter-frame time could obviously reduce the amount of in-plane
motion and thus increase the relative error on the measured particle shift. This may mean in such cases that investment in a three-dimensional stereoscopic PIV system is warranted.

3.2.5 Statistical Accuracy
Like all other measurement techniques, one must consider the effect of the number of samples upon the calculated flow statistics information. It is well known that average quantities converge with an increasing number of statistically independent sample points, but it is important to recognise and quantify the errors on the calculated statistical data. It was with this in mind that the convergence was studied by utilising the flow over the step configuration detailed in chapters 2 and 4 (see also Hollis et al, 2001).

Figure 3.7 shows data for point B in the shear layer of the step flow (see figure 4.11), and typifies the convergence of the mean velocity and second order moments. The convergence can be seen to compare extremely well with standard error estimate curves given by standard texts (e.g. Montgomery and Runger, 1994):

\[ \varepsilon_U = z\sigma \sqrt{\frac{1}{N}} \quad \varepsilon_{uu} = z\sigma^2 \sqrt{\frac{2}{N}} \quad \varepsilon_{vv} = z\sigma \sqrt{\frac{1}{2N}} \]

The value of \( z \) relates to the confidence band, and figure 3.7 shows a 99% confidence band for \( z=2.576 \). The value \( \sigma \) is the true standard deviation (r.m.s.) of the sample, and in the absence of the true value, the population value can be used. \( N \) is the number of statistically independent samples. The convergence agrees well with the formulae, and it was therefore possible to use them to decide the number of samples required. The system was restricted to collecting 600 samples in a single test run giving 99% confidence that the mean velocities were within ±1.1% (for a typical 10% local turbulence intensity). However \( N=600 \) gave r.m.s. accuracy of ±7.4%, and therefore 4 blocks of 600 samples were usually obtained to ensure 99% confidence that the r.m.s. results were within 3.7% of their true value.

It should be noted that each measurement shown in figure 3.7 will be influenced by spatial averaging (described in chapter 6). This is why extreme values are absent in the convergence data, and the reason that the error band magnitudes could be reduced and still capture 99% of the data points.
The comments above regarding statistical accuracy do not incorporate the relative errors on the individual vectors used to create the ensemble (time average) result because of the arising inter-dependencies on particle shift and local turbulence intensity. It is possible to use Pythagorian theory and a standard formula to calculate a ‘total’ relative error:

\[
\frac{\epsilon_U}{U} = \sqrt{\left( \frac{\epsilon_{ds}}{ds} \right)^2 + \left( \frac{\epsilon_U}{U} \right)^2}
\]
where $\varepsilon_{ds}/ds$ is the relative particle shift ($\varepsilon_{ds}$ being the ability to measure displacement to an accuracy of $\pm0.1$ pixels), and $\varepsilon_{U}/U$ is the statistical error (for $N$ samples) described earlier. Therefore, if it is assumed the local turbulence intensity is 10%, and the number of samples is $N=2400$,

$$\varepsilon_{U}/U = \pm 2.576\times 0.1 \times (1/\sqrt{2400}) = \pm 0.53\%.$$ 

And if the average particle displacement of the measurements to calculate the average is 6 pixels,

$$\varepsilon_{ds}/ds = 0.1/6 = \pm 1.67\%$$

and therefore the total error is

$$\varepsilon_{U}/U = \pm 1.75\% \text{ (using 99% confidence bands)}.$$ 

The mean velocity error estimate could be replaced with the r.m.s. error equation for further estimates of the statistical accuracy. Using the above figures gives:

$$\varepsilon_{u}/u^* = \pm 4.08\% \text{ (using 99% confidence bands)}$$

Caution should be applied if using the formula where near zero velocities are evident because percentage errors are not necessarily always the most appropriate form. In these cases the $\varepsilon_{ds}/ds$ approaches infinity, despite the absolute error on individual instantaneous vectors remaining $\pm0.1$ pixels. In such scenarios, it is recommended that an absolute total error (e.g. in m/s) be calculated, the value of which is highly flow dependent:

$$\varepsilon_{\bar{U}}^2 = \varepsilon_{ds}^2 + \varepsilon_{U}^2$$

### 3.3 Processing

The previous section referred to the practical aspects of PIV and their optimisation. This section looks at the next stage – the processing and calculation of the vector field. As mentioned earlier, there is the option to 'pre-process' the image prior to vector calculation. This involves a mathematical manipulation of the raw image and was normally avoided because the basic raw images provided excellent quality data. The only
useful pre-processing option has been the subtraction of a background image to reduce the effect of reflections and glare upon the quality of the data. This was especially useful for the air flow facility with its metal flame tube.

3.3.1 Vector Calculation

The basic ethos of particle image velocimetry vector calculations has been discussed and hinges on two fundamental quantities, time and displacement (Adrian, 1991) -

\[ U(x,t) = \frac{\Delta X(x,t)}{\Delta t} \]

It has also been discussed that the inter-frame time, \( \Delta t \), is defined by the user, and that \( \Delta X \) is calculated by determining the displacement of particle images. In order to calculate the velocity vector field, the raw image is discretised into finite interrogation cells, and a velocity vector calculated in each cell, as illustrated in figure 3.8. In these calculations the assumption is made that all particles move homogeneously within each interrogation cell, but, as can be seen from the illustration in figure 3.8, this may not be true. This topic of sub-grid filtering is extensively covered in chapter 6, but for the purposes of the discussion in this chapter, it will be presumed that the assumption of homogeneous movement of particles within cells is (at least approximately) true.

![Fig. 3.8 Image Discretisation and Individual Cell Illustration](image)
The number of particles within each interrogation cell has an impact on the accuracy of the calculated velocity vector because the greater the number of particles contributing to the correlation peak, the stronger the signal to noise ratio. LaVision (2001) state that adequate data quality is obtained with 3 or more particles per cell, whereas Raffel et al (1998) state that 5 particle images are required within each cell to achieve 95% valid detection in cross-correlation mode. The quoted number of particle images required has generally decreased with the development of better algorithms over time, but the approach used in this project has been to ensure an average of 8 particles per cell across the domain, with the result that at least 5 should exist in any one interrogation cell (note that an arbitrary nine particles have been shown in figure 3.8 for clarity). The reader should not become confused with older publications where the number has been quoted as being as high as 12 or 20 as these generally refer to poorer particle location fitting schemes (e.g. centroid schemes) and the use of photographic film (e.g. Keane and Adrian (1990).

Within each cell, the spatial correlation between the first and second frame of the image pair is calculated via Fast Fourier Transforms (FFTs). This results in a correlation map with a peak intensity positioned where the average displacement within the cell lies. If the displacement is homogeneous and there are a sufficient number of particle images within the cell, and the noise levels are low within the cell, then the peak will be very sharp. It has been shown in the relevant literature (e.g. Westerweel, 1997) that where Gaussian fitting schemes are employed, the optimum particle image diameter is 2 pixels. For particle images of greater than 2 pixels diameter, no further accuracy is gained (although no accuracy is lost either). At diameters of less than 2 pixels, peak locking can occur, which is discussed later. The logic behind the appropriateness of the Gaussian fitting technique is based on the fact that the light distribution of a spherical particle will be an Airy distribution, which is very similar to that of a Gaussian one. Achieving the optimum particle image size of 2 pixels ensures the most accurate determination of particle locations, and thus the most accurate calculated velocity. From the calculated displacement and known inter-frame time, the velocity is obtained. An example of a 3D visualisation of a correlation intensity map is shown in figure 3.9.

Considering the issue of particle density within cells, this is clearly determined by the cell size (pixels), the magnification, and the volumetric seeding density. The magnification
Particle Image Velocimetry and its Optimisation

has already been discussed, and the optimum is determined by the recorded particle image size. Choosing the cell size must be considered carefully, and the choice should theoretically adhere to two criteria:

1. The cell should be small enough to result in homogeneous movement of the particle groups within individual cells. This (in theory) necessitates cell sizes of order of the Kolmogorov length scale (or smaller).

2. There should be a sufficient number of particle images within individual cells to return a valid correlation signal.

If we consider that a typical Kolmogorov length scale in the water flow experiments is 40µm, and that the particle diameter used is 20µm, there are obvious conflicts in the two criteria requiring compromises, a topic returned to in chapter 6. It must also be remembered that the seeding should not influence the flow, and the seeding density should not be so high as to create a fog, thus degrading the image quality. Typically 32x32 pixel cells are utilised in PIV experiments, as for most practical magnification levels such cells provide the optimum set-up. However, as will be seen later it is also possible in some cases to utilise 16x16 cells. When doing so one must consider the effect of reduced particle image density and hence increase in relative noise levels which reduces the probable detection of valid vectors. It is theoretically possible to utilise even smaller cells by employing ‘super resolution PIV’ (e.g. Hart, 2000)

A separate issue is that of the cell overlap, defined as a percentage of the cell size. Cell overlap is the amount by which each interrogation cell is overlapped with subsequent ones, and can simply be seen as a way of increasing vector grid density and hence perceived flow information. For example, using the typical 50% overlap increases the data yield by a factor of 4 (as \[\frac{1}{(1-0.50)}^2=4\]). Examples in the literature exist where up to 92.5% overlap has been used, resulting in a data yield factor increase of 178! (Scanaro and Reithmuller, 2000). However 50% is the most commonly adopted value. By implementing an overlap, the raw image information is used more than once at each location, resulting in the higher data yield. There is however, a genuine reason for using overlap, and that is in terms of vector validation. As will be seen later, it can be argued that if 50% overlap is implemented in the vector calculation, in the event of a single spurious vector being detected (which is surrounded by valid vectors), replacement by
linear interpolation is a valid approach because the raw image information has already been utilised in producing the adjacent vectors (Westerweel, 2001). However, conflicting arguments would say that if the information is already there, why was a spurious vector detected at all? And implementing any overlap implies that the velocity field must be completely homogeneous over the measurement domain. Each cell should contain particles moving totally homogeneously, and if cells overlap, the implication is that there should be no variation across the entire field. Hart (1998) extended the use of cell overlap to effectively reduce background noise and increase peak ratios.

In terms of quantifying the quality of the data calculated across a vector field, this can be assessed in terms of the peak ratio – that is the ratio of the strongest correlation peak (the assumed real displacement) to the next highest peak, where the smaller peak(s) are deemed to be noise. This measure is therefore a quantification of the signal to noise ratio and it is defined as (LaVision, 2001):

\[
Q = \frac{(P_1 - \text{min})}{(P_2 - \text{min})}
\]

Where \(P_1\) is the highest correlation peak, and \(P_2\) the next highest peak. A \(Q\)-ratio of greater than 2 indicates a reasonably strong confidence that the vector is valid, whereas values close to 1 indicate that the vector is probably false (see figure 3.9).

![3D visualisation of Correlation Map Intensities](image)
Particle Image Velocimetry and its Optimisation

A second quality measure is the degree of 'peak locking' – a phenomenon which usually only occurs with small particle image diameters. The degree of detected peak locking is a good indicator of the accuracy of the particle displacement. A visual assessment of the degree of peak locking is provided by a histogram of displacements across the whole vector field; if there is a noticeable bias towards integer displacements, peak locking is occurring; i.e. the particle images are too small for the Gaussian fit to work properly. Examples where peak locking is, and is not occurring is shown in figure 3.10.

![Particle Diameter, d_p=2-3 pixels](image1)

A particle image size of 2-3 pixels guarantees only a slight bias to integral values in PIV evaluation.

![d_p=1 pixel](image2)

A particle image size smaller than 1 pixel causes strong bias to integral values in the evaluation.

Fig. 3.10 Examples of no Peak Locking (top) and Strong Peak Locking (bottom)

The DaVis software also outputs a numerical quantity which quantifies the amount of peak locking, by considering the bias of the displacements towards integer values. Across the vector field a range of velocities exist which are calculated from the pixel displacement between frames A and B, ranging from \( ds_{\text{min}} \) to \( ds_{\text{max}} \). The quantification of the peak locking subtracts from each displacement its greatest integer value, hence \( ds = 4.72 \) pixels becomes \( ds_{\text{dec}} = 0.72 \). The modulus of this number, \( |ds_{\text{dec}}| \) is then calculated (-0.72 becoming 0.72), and then any value greater than 0.5 is truncated to form a value representative of its centre of mass, \( ds_{\text{mass}} = 1 - |ds_{\text{dec}}| \). Therefore the range of \( ds_{\text{mass}} \) exists only from 0 to 0.5. The average \( ds_{\text{mass}} \) of the field indicates the average centre of mass of
the velocity histogram relative to integer displacements and hence the degree of 'Peak Lock' (LaVision, 2001). The actual calculation is given by the following equation:

\[
\text{Peak Lock} = 4 \times (0.25 - d_{\text{mass}})
\]

A value of 0 indicates no peak locking, and a value of 1 indicates all velocities are peak locked at integer values. LaVision (2001) state that Peak Lock < 0.1 is acceptable, resulting in the pixel displacement accuracies of ±0.1 stated earlier.

3.3.2 Advanced Techniques

Techniques exist to enhance the quality of the calculated vector data, and number of calculated valid vectors.

Second Order Correlation

In this technique (also referred to as the 'Hart Correlation' after its author, Hart, 1998), the correlation between frame A and frame B in each cell is calculated twice on slightly shifted interrogation windows. The two calculated correlation functions are then multiplied resulting in an increased correlation peak magnitude, and suppressed noise peaks. However, in situations where the data is generally lower quality, spurious peaks may be enhanced by the multiplication process, hence it is best to first check the quality using the first order correlation; this was the procedure adopted in this work.

Adaptive Multi-Pass

In this method larger cell sizes are used initially, and the velocity vector calculated for the larger cell is used to shift the subsequent smaller cells, thus minimising loss of in plane particle pairs. This method can be used to process at smaller than normal cell sizes, thus enhancing spatial resolution, but one still has to consider the problems associated with low particle density. The adaptive multi-pass method was used extensively in the project to increase quality, and in the overwhelming majority of cases produced data of higher quality than without its use. Typically the initial cell size was chosen as 64x64 pixels down to 32x32, where two iterations were performed.

Deformed Grids

Velocity gradients across cells mean that the particles don't all move exactly the same distance, resulting in a broadening of the correlation peak. The deformed grid method
Particle Image Velocimetry and its Optimisation

originates from the Particle Image Distortion (PID) technique first proposed by Huang et al (1993), and uses a similar idea to that of adaptive gridding. However the initial pass(es) are used to estimate the velocity gradient across cells. Then, based on this information, the image frames are deformed (skewed) in an attempt to eliminate the gradient in the raw image data. This enhances the correlation peak, with bilinear interpolation being used to reconstruct the real velocity vector grid. This method has proven very successful (e.g. Scarano and Reithmuller, 2000), a finding echoed by the author where it was found that higher data quality was always provided when implementing this technique.

Using a combination of the three aforementioned techniques typically increases the Q ratio by 20%, and increases the number of first choice vectors by 10% (with a similar decrease in the number of interpolated vectors).

3.4 Validation

PIV validation methods should not be used to re-invent vast quantities of idealised, smoothed data. Validation is the process of evaluating the quality and reliability of the vector field generated. In most modern PIV software, the process of identifying spurious vectors is automated (as manually checking 1000 vector maps, each containing 4000 vectors could be somewhat time consuming). Inclusion of spurious data in calculations can have dramatic consequences, especially on R.M.S. levels. There are a number of ways in which validation can take place, and all are discussed below.

3.4.1 Pre-defined limits

This really covers two options: geometry masking, and allowable velocity limits. Masking can be applied to any region where the user knows vectors should not appear, such as walls. If the walls are transparent, vectors may be generated because of refracted images in the wall itself, and these may confuse the appearance of the vector field. Masking was extensively utilised in the project to define such areas, which are then eliminated in the calculation. This has the added benefit that masking unused areas prior to vector computation reduces calculation time.
Defining global allowable vector limits for the velocity field requires some previous knowledge of the flow itself, as implementing it excludes any velocity vectors outside the defined range. This should be used with caution because, unless the flow is very well behaved (e.g. unidirectional), it is possible that real vector data will be excluded, perhaps due to a burst of energetic flow or unexpected separation. This validation technique is not so useful where there is a large range of velocities within the flow, because it is likely that spurious vectors will fall within the range of the velocities present. Therefore this validation method was not employed.

3.4.2 Vector Quality
By considering the Q ratio of each vector, it is possible to define allowable limits below which the vector is eliminated on the basis that it is probably noise; i.e. if the ratio of the highest peak to the next highest peak is low (close to 1), then it is likely that the vector is invalid. However, if noise is high, and the peak ratio is generally low, this validation technique may remove good vectors. It is a useful validation technique if the quality of the raw data is high to start with, although choosing a threshold can be difficult, and the risk is always that good data may be removed. A threshold value of 1.3 was utilised during this project.

3.4.3 Consideration of Local Flow Conditions
The most trusted method of flow validation is that which considers the neighbouring vectors. This method was proposed first by Westerweel (1993), who stated that if a vector deviates substantially in direction or magnitude compared to its neighbouring vectors, flow continuity is not satisfied and the vector must be spurious. He proposed a technique whereby at each vector location the average magnitude and standard deviation of the surrounding velocities (usually 8 vectors) be calculated. Then the vector in question was compared to the average of its neighbours, plus or minus a factor (defined by the user) multiplied by the standard deviation of the neighbouring vectors. If it fell outside the defined range the vector was deemed spurious and removed. The factor chosen which is applied to the standard deviation depends on the turbulence of the flow, but is typically 1.3.
Subsequent to Westerweel's ideas, the method was improved. This included taking the mode (most 'popular') value of the surrounding vectors instead of the average, as this avoided any large deviations in the magnitude due to a single surrounding vector contaminating the average value.

### 3.4.4 Replacing Removed Data

It is important that if data is removed from a vector field because it is deemed to be spurious, that it is then replaced. If the data is not replaced, it can affect any calculated statistics (and especially higher order moments) in the event that zeros are included. It is also important that vectors are replaced when calculating data dependent on the vector information being continuous across the whole field, such as correlation data (see chapter 4) and integral length scales (see chapter 5).

The most popular form of data replacement is linear interpolation. This method simply calculates the average magnitude of the surrounding vectors and replaces the missing vector with such value. This ensures flow continuity but if there are too many spurious vectors removed, the effect of replacing via linear interpolation can be to 'smooth' the flow field. However, it must be remembered that, if the data quality is poor, then the initial experimental set-up should be revised and re-optimised, as there is little point trying to 'create' good data.

Another method of replacement is that employed by the LaVision DaVis software, which considers the validity of vectors calculated using the smaller correlation peaks. If the highest peak is deemed invalid, what about the next highest peak? Does that fit the validation criteria? This is a logical method because it may be that a strong spurious correlation peak is recorded (for whatever reason), whereas a smaller correlation peak which appears initially to be noise, contains the real flow information. Hence the method first considers the highest correlation peak, but if it fails to fit the correlation criteria, the next highest peak is checked against the validation process, and this is repeated with the 3rd and 4th highest peaks (any further peaks are deemed to be real noise, and in fact it is rare that a vector is replaced using the third or fourth correlation peak). If none of the peaks fit the validation criteria, only then is linear interpolation employed.
Note that 'smoothing' has been completely ignored, as this type of post-processing makes no consideration of the actual quality or validity of the data. It only serves to make the flow continuity look 'better' than has actually been recorded, and as such is never implemented in this project.

3.5 Analysis and Presentation

Each instantaneous vector field, \( U(x,t) \), contains a wealth of quantitative information regarding the arrangement of structures within the flow, but a distinct challenge exists in presenting the information in a compact and descriptive manner. Streamtraces are widely utilised during the presentation of data in this document, because when combined with simple velocity contours (e.g. \( U_i < 0 \)), they provide an excellent visualisation of the flow topology. However one must always bear in mind when examining such data that, despite the Lagrangian description implied by the traces, they do not show the path taken by single fluid packets because they are constructed using the instantaneous planar vector data.

Statistical quantities such as time average velocities, \( \overline{U}(x) \), and root mean square (r.m.s) velocity data, \( u'(x) \), are obviously calculable but offer no extra information compared to point measurement techniques. Additional quantities are available due to the planar nature of the data, and chapters 4 and 5 show that other important statistical fluid dynamic quantities can be directly calculated including integral length scales. This, together with certain assumptions, allows the calculation of dissipation rates and Kolmogorov scales of the flow.

Conditionally averaged velocity fields were a valuable means of identifying dominant structures linked with a certain feature occurring in the field in this project. These were generated by selecting a threshold condition, A or B, for the instantaneous velocity at a point (or across a region) of interest, \( U(o) \). The instantaneous vector fields fitting the criteria are summed and divided by the number of samples fitting the criteria, \( N_c\alpha \), creating an ensemble conditional average, \( \overline{U}_{c\alpha}(x) \). This is described mathematically as follows (note that N represents the total number of samples in the data series):

\[ \overline{U}_{c\alpha}(x) = \frac{1}{N_c\alpha} \sum_{i=1}^{N_c\alpha} U_i(x) \]
The following two sections describe two other important categories of data available from PIV.

### 3.5.1 Decomposed Vector Fields

The use of decomposed fields in their various guises is becoming widely used in the presentation of PIV data, and is supported by workers in the field (e.g. Adrian et al, 2000). By subtracting the time average vector field, the well known Reynolds decomposition results which highlights instantaneous fluctuating components relative to the time average flow. The traditional conception is that motions such as vortices are more easily identified by this form of the vector data. It is described by the following:

\[ u(x, t) = U(x, t) - \overline{U}(x, t) \]

A method was devised during the investigation whereby the conditional average and Reynolds decomposition were combined to provide a powerful means of identifying dominant flow characteristics. This has been called 'PDF conditioning.' As defined earlier, a criteria is specified for the instantaneous velocity at a point of interest, but prior to the averaging process the time average of the entire data set is subtracted from the vector fields satisfying the relevant criteria:

\[ \overline{U}_{ca}(x) = \frac{1}{N_{ca}} \sum_{i=1}^{N} a_i (U(x)_i - \overline{U}(x)) \]

In PDF conditioning the conditions A and B are related to the velocity distribution (PDF) at the point of interest, \( U(o) \), and chosen so that only the vector fields with \( U(o) \) in the tails of the PDF satisfied the criteria. The operation is illustrated in figure 3.11. Separate criteria for each tail of the PDF was necessary:

\[ \overline{U}_{ca-}(x) = \frac{1}{N_{ca-}} \sum_{i=1}^{N} a_{i-} (U(x)_i - \overline{U}(x)) \quad a_{i-} = \begin{cases} 0 & \text{if } U(o) > \overline{U}(o) - Ku'(o) \\ 1 & \text{if } U(o) \leq \overline{U}(o) - Ku'(o) \end{cases} \]

\[ N_{ca-} = \sum_{i=1}^{N} a_{i-} \]
K is a coefficient chosen to maximise the effect of the averaging. If the value of K is too small, then events (vector fields) that do not satisfy the motion of interest will be included in the averaging, whereas if K is too large too few events will satisfy the criteria and hence the ensemble average will have higher uncertainty due to a lack of samples. The author has found a value of K=1.5 to perform well. The result of the operation are two PDF conditioned (and Reynolds decomposed) ensemble vector fields. One of the main uses of this data form was to verify and physically explain the space-time cross-correlation maps (see chapters 4 and 7).

\[
\overline{U_{ca}}(x) = \frac{1}{N_{ca}} \sum_{i=1}^{N} a_i \left( U(x) - \overline{U(x)} \right)
\]

\[
a_i = \begin{cases} 
1 & \text{if } U(x) < \overline{U(x)} + K u'(x) \\
0 & \text{if } U(x) \geq \overline{U(x)} + K u'(x)
\end{cases}
\]

\[
N_{ca} = \sum_{i=1}^{N} a_i
\]

Fig. 3.11 Conditional Average : PDF Conditioning Illustration

Adrian et al (2000) introduce two other decomposition processes for PIV: Galilean and Large Eddy Simulation (LES) decomposition. Galilean decomposition is the simplest, whereby a fixed (assumed) convection velocity is subtracted from all instantaneous vectors. This simple technique allows convection velocities associated with vortex motions to be identified, but does require a range of convection velocities to be investigated in order to reveal all vortical motions. LES decomposition, however, is a
more efficient means of identifying the vortex motions. In this case the velocity field is firstly low pass filtered to obtain the large scale motions in the field:

\[ \tilde{U}(x, t) = \int_{D} G(r, x) U(x-r, t) \, dr \]

The function \( G \) is the filtering kernel operating on the local instantaneous velocity at \( x \). The filter operates locally on each vector \( U(x,t) \) over a range according to \( r \), and over the entire local domain, \( D \). The type of LES decomposition discussed here utilises a simple box filter which results in a filtered velocity that is a spatial average over a square matrix of vectors centred at \( x \). The vector \( r \) operates according to the filter width \( \Delta \), which is essentially the vector grid dimension of the averaging matrix. Hence, \( \Delta^2 \) represents the number of vector grid positions, \( n \), used in creating each local spatially averaged velocity. However, it is the small scale (high pass filtered) velocities, \( \tilde{u}(x,t) \), that are usually of most interest, and are obtained by subtracting the local instantaneous velocity from the low pass filtered velocity. We therefore have a velocity field decomposed according to the relative size of the motions:

\[ \tilde{u}(x, t) = \tilde{U}(x, t) - U(x, t) \]

The benefit of the LES decomposition (a term that will be used to describe the high pass - small scale - filtered field from now on) is that the calculated small scale velocities are always relative to the local convection velocity, which means that all vortical motion should be identifiable. Although this discussion has referred to the box filter, a standard homogeneous filter, significant work has been directed at investigating the power of utilising inhomogeneous filters such as using Proper Orthogonal Decomposition (POD) to define the filter (e.g. Adrian et al, 2000; Bernero and Fiedler, 2000). It is believed that POD analysis of PIV data will become a popular technique in the near future.

### 3.5.2 Derivative Quantities

With PIV being a planar measurement technique it is possible to easily calculate derivative quantities, \( \partial U_i / \partial x_j \) (where \( i=j \) or \( i \neq j \)), which are required in the calculation of
Particle Image Velocimetry and its Optimisation

several scalar quantities. The vorticity perpendicular to the measurement plane is an important indication of fluid dynamic quantity and is defined as follows:

$$\omega_k = \frac{\partial U_j}{\partial x_i} - \frac{\partial U_i}{\partial x_j}$$

Where the measurement plane exists in the i and j directions and k is the axis perpendicular to the plane, i.e. $i \neq j \neq k$. All derivative data presented herein are calculated via a central differencing scheme utilising the nearest neighbours to the subject vector. For example, in (I, J) space the calculation of the vorticity becomes:

$$\omega_k = \left[ \frac{U_j (I+1, J) - U_j (I-1, J)}{(I+1, J) - (I-1, J)} \right] - \left[ \frac{U_i (I, J+1) - U_i (I, J-1)}{(I+1, J) - (I, J-1)} \right]$$

A limitation exists in utilising the vorticity quantity to identify regions of swirling fluid as it identifies shearing motion as well as vortex cores (Adrian et al, 2000). An alternative quantity to vorticity is the swirl strength ($\lambda_{ci}$) — a value presented by Zhou et al (1999) who show that the local velocity gradient tensor $D$ can be decomposed in Cartesian coordinates as shown in figure 3.12.

The swirl strength is the imaginary part of the complex eigenvalue pair of the local velocity gradient tensor, and quantifies the strength of the local swirling motion. Unlike vorticity, swirl strength does not identify areas that contain significant vorticity that are
absent of swirling motion (e.g. shear layers). It will be seen in chapter 7 that the swirl strength is a highly effective means of identifying vortical motion. However, due to the complex nature of the value of swirl strength, it assumes only positive values and therefore vorticity remains a useful quantity in that sense. Like the conditional averaging of the vector data described earlier, the same idea was applied to the swirl strength and vorticity data in identifying the loci of strongest swirling motion.

3.6 PIV Optimisation

The theory of PIV optimisation has been included in the previous sections and the entire optimisation process has been summarised in the flowchart of figure 3.12. It assumes that the PIV hardware is already in place and cannot be changed (e.g. the camera cannot be changed for a higher resolution CCD in order to get 2 pixel diameter particle image). It also assumes that the out-of-plane motion is within the criteria specified, or that the laser sheet thickness and depth of focus can be increased to accommodate that without degrading the results beyond the acceptable level. If not, a stereoscopic PIV system would be necessary.

Figure 3.12 covers all of the main points with respect to optimisation, without reference to the equations given earlier. It provides a model path to follow, but in reality the practicalities often overcome the theoretical ideals, and the whole optimisation process is a highly iterative one. For example it is unlikely that the final cell size can be chosen equal to the Kolmogorov length scale, whilst providing an adequate number of particle images within individual cells. It is also unlikely (especially in low Reynolds number flows) that the inter-frame time can be chosen as small as the Kolmogorov time scale and still result in adequate particle shift to give a reasonably accurate recorded displacement. In reality time and length scales equal to 1/5 (Hoest-Madsen and Neilsen, 1995) or a 1/6 (Pope, 2000) of the integral scales will probably replace the ‘Kolmogorov’ terms in figure 3.13, and a reduction in the ability to measure the smallest scale flows must therefore be accepted. This whole issue is discussed in considerable depth in chapter 6. Subsequent to the flow chart, table 3.2 identifies the typical parameters employed during the acquisition of data in this project.
Particle Image Velocimetry and its Optimisation

START HERE

1. Position Camera and Set Magnification to Capture Ideal FOV
   - Can laser power be adjusted to give individual particle image diameters of 2 pixels?
     - YES: Is it vital that all 'zones' within the FOV be imaged simultaneously?
     - NO: Choose optimum seeding that should (theoretically) follow flow accurately and scatter sufficient light

2. Magnification to Capture Ideal FOV
   - Is seeding of increased size be used that has sufficient flow following capabilities?
     - YES: Acquire raw data in different zones on separate test runs
     - NO: Choose optimum seeding that should (theoretically) follow flow accurately and scatter sufficient light

3. Re-design experiment using:
   - Different flow scenario
   - Different geometry
   - Different PIV set-up

4. Choose optimum seeding that should (theoretically) follow flow accurately and scatter sufficient light
   - Could seeding of increased size be used that has sufficient flow following capabilities?
     - YES: Select (final) interrogation cell size equal to Kolmogorov length scale and select 0% overlap
     - NO: Increase seeding density as far as possible whilst maintaining image quality. Are there now at least 5 particles within each cell?

5. Can sub-grid filtering inaccuracies/correction be accepted? (see chapter 6)
   - YES: Increase cell size to achieve ≥5 particles per cell (see chapter 6 for resultant sub-grid filtering effect)
   - NO: Can final cell overlap be increased (within acceptable limits) to achieve data yield

6. If reduced accuracy cannot be accepted, can the magnification of 'zone' (or AOI) be increased?
   - YES: Can At be decreased whilst maintaining large enough particle shift across flow field to give adequate velocity measurement accuracy?
   - NO: Can reduced accuracy in terms of dynamic averaging be accepted?

7. Can At be decreased whilst maintaining large enough particle shift across flow field to give adequate velocity measurement accuracy?
   - YES: Choose At for maximum particle shift of 1/4 (initial) cell size
   - NO: Can final cell overlap be increased (within acceptable limits) to achieve data yield

8. Does final cell size provide required grid resolution (data density)?
   - YES: Select Adaptive Gridding with initial cell size large enough to contain at least 5 particles
   - NO: Can final cell overlap be increased (within acceptable limits) to achieve data yield

9. Can At be decreased whilst maintaining large enough particle shift across flow field to give adequate velocity measurement accuracy?
   - YES: Is At < Kolmogorov time scale?
   - NO: Can reduced accuracy in terms of dynamic averaging be accepted?

10. If reduced accuracy cannot be accepted, can the magnification of 'zone' (or AOI) be increased?
    - YES: Choose At for maximum particle shift of 1/4 (initial) cell size
    - NO: Can final cell overlap be increased (within acceptable limits) to achieve data yield

11. Can reduced accuracy in terms of dynamic averaging be accepted?
    - YES: Successful PIV
    - NO: Resolve issues and repeat process

Fig. 3.13 PIV Optimisation Procedure (* see notes on next page)
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* Cell size can be chosen smaller than Kolmogorov length scale if a higher spatial resolution is required, however it is unlikely to be possible in anything but micro-PIV investigations (e.g. Santiago et al, 1998; Lindken et al, 2002)

** If cell size is increased beyond approximately 1/5 of the integral length scale, this will have serious effects in terms of sub-grid filtering and on the calculated RMS values (see chapter 6).

*** Even when using Adaptive Grids, although only 5 particles are, in theory, required in the initial cell size, it must be ensured that approximately 3 particles remain in the final cell (Lavision, 2001) in order to give high enough confidence in the calculated vector. This means practically that 12 particles are required in the initial cell dimension if it is twice that of the final cell size. Even if super-resolution PIV (where effectively the final pass results in the tracking of cells containing only a single particle) is possible, 16 particles would be necessary in the initial cell if the final cell has dimensions of a quarter of the initial one (e.g. 128x128 to 32x32).

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</tr>
<tr>
<td>Remove vectors outside range</td>
<td>&gt; 1.3x standard deviation of neighbours</td>
</tr>
<tr>
<td>Replace removed vectors</td>
<td>With 2(^{\text{nd}}), 3(^{\text{rd}}), or 4(^{\text{th}}) choice peaks</td>
</tr>
<tr>
<td></td>
<td>Or interpolated data where no peak fits surrounding fluid dynamic behaviour</td>
</tr>
</tbody>
</table>

Table 3.2 : Typically Employed PIV Parameters

3.7 Limitations

There are two types of limitations to consider when acquiring PIV data: those related to the fundamental specification of the hardware and those related to the processing. The main limitations have all been mentioned earlier (e.g. temporal resolution) but it was thought useful to compare the limitations of LDA and PIV in table 3.3.
<table>
<thead>
<tr>
<th>LIMITATION</th>
<th>LDA</th>
<th>PIV</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simultaneous instantaneous</td>
<td>Only one measurement at a given instant in time.</td>
<td>Number of instantaneous (2D) measurements dependent upon the number of interrogation cells. Typically 63x61 cells giving 3843 instantaneous velocity vectors.</td>
</tr>
<tr>
<td>measurements</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temporal resolution</td>
<td>Temporal resolution usually around 1kHz enabling temporal autocorrelation to be performed, and time scales derived. Temporal resolution dependent on particle arrival rate.</td>
<td>Temporal resolution typically only 15Hz, making it extremely difficult to generate time scales using autocorrelation to any degree of accuracy. Temporal resolution dependent upon hardware limitations.</td>
</tr>
<tr>
<td>Spatial Averaging Issues</td>
<td>Individual particles are effectively tracked and the software requires F (typically 8) Doppler fringes are passed consecutively in order to return a valid velocity value. If the system recorded any fringes passed by any particle within the CV, then there would be some group averaging effects.</td>
<td>Group of N (e.g. 8) particles ‘tracked’. The result is that the average (modal) velocity of the N particles is recorded and any fluctuations within the cell are filtered.</td>
</tr>
<tr>
<td>(see chapter 6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Particle displacement accuracy</td>
<td>Dependent upon fringe spacing (typically 3µm)</td>
<td>Dependent upon particle size (ideally d_P = 2 pixels) and sub pixel fitting accuracy (typically 0.1 pix, equating to around 5µm)</td>
</tr>
<tr>
<td>Measurement location accuracy</td>
<td>Dependent upon CV size. The particle can pass through any part of the CV and the measurement be recorded (e.g. 3.0mm in major axis, and 0.30mm on minor axis)</td>
<td>Calculated vector could be based on particle movement anywhere in cell, hence dependent upon interrogation cell size which is typically ΔX=1mm. Hence vector location accurate to +/-0.5mm. In practice the ‘central’ group of particles is tracked.</td>
</tr>
<tr>
<td>Dynamic averaging</td>
<td>Δt equivalent is dependent on fringes passed (F), fringe spacing (d_f), and flow velocity, U, where Δt = (F.d_f)/U. Fringe spacing typically 3µm, and flow velocity (in water) approx 1m/s, giving Δt=24µs. The degree of dynamic averaging determines ability of the method to get an ‘instantaneous’ measurement. A measurement is instantaneous if Δt ≤ Kolmogorov time scale.</td>
<td>Δt determined by the user. Δt typically 200µs in water, resulting in ds (which is equivalent to d_P) of 400µm for a 1m/s flow. As PIV records a 2D plane, the range of velocities can be large, and one must ensure Δt is small enough to capture ‘instantaneous’ velocities, but sufficient to result in large enough displacement of particles to be detected.</td>
</tr>
<tr>
<td>Spatial resolution</td>
<td>Single point only for inst values. For average values, only depends on accuracy to which CV can be located.</td>
<td>For instantaneous values, dependent upon interrogation cell size.</td>
</tr>
</tbody>
</table>

Table 3.3 Comparison of LDA and PIV Limitations
An additional challenge is the aspect of data management. With each image occupying 2MBytes of hard disk space, and each vector field occupying 100KBytes, a complete set of 600 instantaneous velocity fields is over 1GByte. And this neglects to mention further calculations (correlations, decomposed fields, etc.). The strategy adopted during the project was to delete all image files after processing the vector fields, apart from the image files of the most critical data sets. This significantly reduced the hard disk space occupied.

3.8 The Future of PIV

Being technology based and with an enthusiastic and growing research community, PIV naturally advances at some pace. Thirteen years ago the first PIV images were being acquired using video (Willert and Gharib, 1991), a technique which spawned the use of digital cameras in the arena. Nowadays the thought of using photographic film as a medium for capture is unheard of, apart from in very high speed PIV studies (e.g. Williams et al, 2003) where digital cameras still strive to achieve the multi-kilohertz frame rates. Indeed, as an example of the rate of progression, during the course of this project the ability to carry out PIV measurements at kilohertz rates has became a realistic proposition. Van Doorne et al (2003) combined this high speed PIV with a stereoscopic set-up to achieve high quality spatially and temporally resolved measurements in a pipe flow. As the ability to acquire three component data at rates rivalling point based LDA techniques, and with digital camera resolution and sensitivity rapidly progressing, PIV is becoming a real alternative to those point based techniques. Stereoscopic measurements are relatively commonplace now, and we are seeing other multi-camera, multi-laser configurations allowing calculation of acceleration fields (Christensen and Adrian, 2002) and dual plane measurements (Adrian, 2003). Holographic PIV remains an elusive challenge, but examples of single instantaneous whole volume PIV data have been acquired (Hinsch et al, 1999), and it surely will not be many years before this technique comes to the forefront. In the world of CFD Reynolds-Averaged Navier-Stokes (RANS) calculations have to an extent been superseded by LES, which will in turn become ‘outdated’ (eventually) due to the ever increasing computing power that fuels Direct Numerical Simulations (DNS). And one day, kilohertz frame rate holographic PIV will be there to validate it. We will then have to ask what other advancements PIV offers the
research community, if any? Or maybe PIV, like LDA, and like the researchers driving these techniques, simply matures whilst a new more exciting, more efficient alternative arrives. However, it will be many years before PIV has no more to offer us in the way of the quantification of newly identified fluid dynamic phenomena, as this document will demonstrate in the subsequent chapters.

3.9 Closure

The PIV process has been discussed in detail, together with the optimisation of the various parameters. The most important group of steps in the process is the physical setup: light sheet, seeding, camera, and timing. By ensuring a good quality basic experimental set up and interactively checking the quality of the data (signal to noise ratio, peak locking, number of first choice vectors) during the tuning of the timing parameters, the PIV results should be of the highest possible standard. The procedures in this chapter have been adopted as 'best practice' throughout the project.

PIV is not without its limitations though, and the next chapters develop a basis around which the effects of spatial averaging on PIV measurements in turbulent flows can be corrected for. This is extremely important in practical engineering flows, such as those within gas turbine combustors.
Chapter 4

CORRELATION FUNCTIONS
A major objective of the project was to ascertain the inter-dependencies in the fluid dynamic behaviour within the combustor flow fields, and provide quantitative information on flow coupling effects. A powerful method of determining fluid dynamic interactions separated in time and space is via the calculation of correlation coefficients. The PIV system is the ideal platform for doing this because it provides instantaneous planar velocity data, from which spatial correlations can be calculated. The ability to calculate space-time correlations was not an in-built feature of the commercial system, and therefore a method had to be developed, and validated.

The chapter firstly explains the concept and background to the use of correlation functions as a fluid dynamic analysis tool. Subsequent to this the mathematical descriptions of correlation coefficients are given, followed by a description of the software written to calculate the data from the instantaneous PIV vector fields. Finally the software validation process is explained which involved the generation and use of synthetic data exhibiting known correlations. It is important to realise that the 'correlations' discussed in this chapter and the results chapters refer to calculations using the velocity data. It is not the same as the 'cross-correlations' discussed in chapter 3 with reference to the processing of raw image data to obtain the velocity vectors.

4.1 Background

A correlation is defined as a mutual relation, especially of phenomena regularly occurring together. In fluid dynamic engineering terms it provides a quantitative measure of the relationship of a property at a point in time and/or space, relative to another point in time and/or space. In calculating correlations, information is generated which identifies how a perturbation at point A (at time t) influences point B (at a different time).

Traditionally, the calculation of correlation coefficients was restricted to Eularian time correlations – i.e. single point time varying quantities or 'Autocorrelations’. This was due to the limitations of the single point anemometer. Alternatively correlations performed for zero time offsets are termed 'Spatial Velocity Correlations’. Such correlations were difficult and extremely time consuming to perform with the single point anemometers, necessitating multiple probes or delicate traversing mechanisms, an example of which can be found in Kroeff and Carrotte (2002). With multi probe
configurations, the spatial velocity correlation is easily adapted to perform space-time correlations, or ‘Cross Correlation Functions’. These allow the experimentalist to assess whether a perturbation at one point can be found to survive in space and time, and subsequently influence another point in the flow. Hughes and Carrotte (2004) used two HWA probes to identify a correlation between the port feed flow and the primary jet characteristics of the VULCAN sector rig described in chapter 2. However, performing such correlations is an extremely time intensive procedure, and can be analogous to searching for a needle in a haystack.

With a temporal resolution of only 15Hz, utilising the PIV system to acquire useful autocorrelations is limited to low velocity flows. However, its planar nature is ideal for calculating spatial velocity correlations. Also, given a relatively low convection velocity (such as those within the water analogy facility) the calculation of valuable cross correlation functions becomes feasible, with the advantage that multi point correlation maps can be generated making it easier to identify correlated phenomena. Adrian (2003) calculated multi point correlation data from PIV data in assessing the convection velocity of disturbances in boundary layer flows, and enthuses that multi point correlations are a “big bonus” of two dimensional data, and that “as fluid dynamicists, we should take advantage of this.”

4.2 Mathematical Description

The correlation is the normalised version of the covariance, covij, where the covariance between points xA and xB separated by time τ is given by:

\[
\text{cov}_{ij}(x_A, x_B, \tau) = \frac{\overline{u_i(x_A, t)u_j(x_B, t+\tau)}}
\]

In the above equation, the velocity direction is assigned via the use of subscripts i and j where, as described in chapter 2, each letter can represent any of the three co-ordinate directions, i.e. in Cartesian space streamwise (x1), normal (x2), or transverse (x3). The covariance can be calculated between velocities in the same (i=j) or differing directions (i≠j). The separation of points A and B is described by a vector relationship:

\[
x_B = x_A + \mathbf{r}
\]
Correlation Functions

It is important to realise that the velocity component in the covariance equation is the fluctuating component, obtained from the Reynolds decomposed data series:

\[ u_i(x, t) = U_i(x, t) - U_i(x) \]

Normalisation of the covariance by the product of the root mean square value of the fluctuating velocities at point A and B is standard practice, yielding the generalised correlation function (where \( x_A \) has been replaced simply with \( x \)):

\[ \text{CCF} = R_{ij}(x, r, \tau) = \frac{u_i(x, t)u_j(x + r, t + \tau)}{\sqrt{u_i(x, t)^2} \sqrt{u_j(x + r, t)^2}} \]

The above, normalised correlation function will be referred to as the Cross Correlation Function (CCF). The CCF can also be considered in two simplified forms discussed earlier:

- The Spatial Velocity Correlation (which will be referred to as the 'spatial correlation' or SVC), where correlations are performed at the same point in time but the location of point B (and hence the vector, \( r \)) varies:

\[ \text{SVC} = R_{ij}(x, r) = \frac{u_i(x, t)u_j(x + r, t)}{\sqrt{u_i(x, t)^2} \sqrt{u_j(x + r, t)^2}} \]

This equation is refined further when considering spatial correlations for a particular velocity component, i.e. for the spatial correlation of the streamwise velocity component (where the streamwise velocity direction is \( x_1 \)):

\[ \text{SVC}_{11} = R_{11}(x, r) = \frac{u_{11}(x, t)u_{11}(x + r, t)}{\sqrt{u_{11}(x, t)^2} \sqrt{u_{11}(x + r, t)^2}} \]

And likewise for the normal velocity component \( x_2 \); substitute '2' for '1'.

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Correlation Functions

- The Eularian Time Correlation (which will be referred to as the 'autocorrelation' or ACF), where the location of point B is identical to that of point A \((r=0)\), but the time shift, \(\tau\) is varied.

\[
ACF = R_{ii}(x,\tau) = \frac{u_i(x,t)u_i(x,t+\tau)}{\sqrt{u_i(x,t)^2} \sqrt{u_i(x,t)^2}}
\]

In a similar manner to the earlier equations, one can refine the above for a particular velocity component.

The SVC gives an indication of how a quantity at a spatial location is correlated with other quantities in the domain. The SVC allows the direct calculation of integral lengthscales (see chapter 5). Likewise, the ACF gives an indication of how a quantity is correlated in time. It allows the direct calculation of integral timescales.

4.3 Algorithm Structure and Operation

The code implemented to calculate the correlations was written using the 'C' like DaVis CL macro language within the PIV system software. Throughout this section only velocities are referred to as the correlated quantity, and only the generalised CCF is described. Functionality was built into the software to allow the correlation of other fluid dynamic quantities, but the investigation of alternate correlation quantities (e.g. vorticity) proved unrewarding. When considering the generalised CCF in the following paragraphs, it is simple to replace the relevant functions with the ACF or SVC equations given earlier.

Chapter 3 highlighted that two dimensional digital PIV essentially records a series of planar data, where each planar domain is a regularly distributed grid of instantaneous velocity data. This is unlike point measurement techniques which record a series of data at a single point. All of the PIV data collected in this project was sampled at 1511z \((\Delta T=1/15)\) – the maximum repetition rate of the camera and laser. Hence the time variation, \(\tau\), is replaced with \(n\Delta T\) in the correlation equation, where \(n\) represents the buffer 'offset'. A graphical representation of a data series is given in figure 4.1.
Points A and B will be referred to throughout this section, and because we are explicitly considering PIV data, it is possible to vary point B across the domain range; i.e. the correlation can be calculated between the ‘origin’ of interest (point A), and every other point in the domain (multiple point B’s). One must appreciate that, although ‘points’ A and B are referred to for simplicity, A and B are in fact individual interrogation cells of finite volume, and cannot strictly be thought of as true point measurements. This issue is elaborated on in chapter 6 regarding sub grid filtering.

In keeping with the way in which DaVis stores the PIV data in a buffer array, the location of the first sample in the data series is called Buffer\textsubscript{first} and the last sample Buffer\textsubscript{last}. The ensemble average correlation is calculated for the defined temporal offset $\tau=n\Delta T$ over the data set available. Figure 4.2 illustrate how correlations between A (at time t) and B (at time t+n\Delta T) are calculated for the example of n=4. Each vertical line represents a two dimensional vector buffer (as illustrated in figure 4.1). The average correlation is thus calculated over $[((\text{Buffer}_{\text{last}}-\text{Buffer}_{\text{first}})+1)-n]$ samples. In figure 4.2 Buffer\textsubscript{start}=1 and Buffer\textsubscript{last}=499 therefore the correlation is the average over 496 samples.
Correlation Functions

Define temporal offset \( n_{\text{first}} \) and \( n_{\text{last}} \)

Define point A: \( x_A = (i_A, j_A) \)

Point B range, \( x_B \):
(\( (i_{B\text{ first}}, j_{B\text{ first}}) \) to \( (i_{B\text{ last}}, j_{B\text{ last}}) \))

Define buffer input range: Buff\(_{\text{first}}\) to Buff\(_{\text{last}}\)

Note the use of \( i, j \) space co-ordinates (instead of \( x, y \)) to simplify the application to a 2D computational domain.

Fig. 4.3 Correlation Algorithm Flowchart
Correlation Functions

Varying point B allows a correlation contour map for various values of n, and hence nΔT, to be generated. This satisfies one of the objectives in terms of being able to identify the peak correlation within the measurement domain for each value of nΔT (for a chosen origin A), and therefore allows the user to effectively track the motion and evolution of the correlated fluid flow for each temporal offset. Relative to the origin, the way in which the surrounding ‘upstream’ (for -n) and the ‘downstream’ (+n) fluid motion affect or are affected by the motion at point A can be scrutinised.

The flowchart in figure 4.3 summarises how the algorithm obtains the CCF from the PIV data. Notice that the calculations use the Reynolds decomposed velocities.

4.4 Synthetic Velocity Data

The ideal way to test the validity of the algorithm is via the use of velocity data with known correlation properties. Examples in the literature show that synthetically generated data has proved useful in testing the performance of PIV algorithms (e.g. Lecordier et al, 2001) due to the high level of control and a priori knowledge of the synthetic fluid dynamic behaviour. It is possible to generate synthetic data suitable for testing the algorithm using software by Klein et al (2003). The software was designed to provide inflow boundary conditions for spatially developing Direct Numerical or Large Eddy Simulations. The synthetic data is able to reproduce first and second order one point statistics as well as user-specified time and lengthscales. It would therefore provide a direct means of testing the correlation algorithm, with only a few minor modifications to the original code in terms of the output format.

The software by Klein et al (2003) is based upon a method proposed by Lund et al (1998) which takes into account cross correlations between velocity components. A velocity signal \( U_i \) is defined for each velocity component which possesses a prescribed two point statistic (length scale, energy spectra). This is solved using an idea based on that of Lee et al (1992) which utilises inverse Fourier transforms. Initially, the velocity component is defined such that;

\[
\bar{U}_i = 0 \quad \bar{U}_i \bar{U}_j = \delta_{ij}
\]
where $\delta_{ij}$ is the Kronecker delta. The following transformation is then performed where $u_i$ is the required velocity signal:

$$u_i = u_i + a_{ij}U_j$$

where

$$a_{ij} = \begin{bmatrix}
(R_{11})^{1/2} & 0 & 0 \\
\frac{R_{21}}{a_{11}} & (R_{22} - a_{21}^2)^{1/2} & 0 \\
\frac{R_{31}}{a_{11}} & \frac{(R_{32} - a_{21}a_{31})}{a_{22}} & (R_{33} - a_{31}^2 - a_{32}^2)^{1/2}
\end{bmatrix}$$

\[\text{Fig. 4.4 Synthetic Data Co-Ordinate System (Unmodified Format)}\]

The co-ordinate frame of the original outputted data was of the form shown in figure 4.4. The three velocity components ($u_1$, $u_2$, $u_3$) were initially contained in a two dimensional ($x_2$, $x_3$) plane, and the velocity vectors varied with time, $t$. Within the algorithm, the user specifies the integral timescale in the primary (streamwise) direction, $T_{L11}$, and the longitudinal integral lengthscales in the normal and transverse directions, $2L_{22}$ and $3L_{33}$. Therefore the velocity vectors are correlated in time in the primary direction according to $R_{11} = f(T_{L11}, \tau)$, and in space in the normal and transverse directions according to $R_{22} = f(x, 2L_{22})$ and $R_{33} = f(x, 3L_{33})$. For a full description of the integral time and lengthscales, the reader is referred to chapter 5, but for simplicity these are often referred to simply as time and length scales in this section.
Correlation Functions

The model correlations are of the following 'special' shape, which is representative of homogeneous turbulence in a late stage:

\[ R_{22}(x, \Delta x_2) = \exp\left( -\frac{\pi(\Delta x_2)^2}{4(L_{22}^2)^2} \right) \]

The \( R_{33} \) correlation takes a similar form. The lateral lengthscales are not specified, but are the same as the longitudinal lengthscales in the relevant directions, i.e. \( 2L_{33} = 2L_{22} \) and \( 3L_{22} = 3L_{33} \), and hence;

\[ R_{22}(x, \Delta x_2) = \exp\left( -\frac{\pi(\Delta x_2)^2}{4(L_{33}^2)^2} \right) \quad \text{and} \quad R_{33}(x, \Delta x_3) = \exp\left( -\frac{\pi(\Delta x_3)^2}{4(L_{22}^2)^2} \right) \]

The correlation function is mathematically related to the energy spectrum (in wavenumber space), the latter coming from an integral function of the velocity spectrum tensor, which itself forms a Fourier transform pair with the two point correlation function. The model correlation function type therefore has important implications upon the spectrum and hence the type (or stage) of the turbulence that is simulated. The squared dependencies in the above formulae form a model which most closely represents homogeneous turbulence, where the production and dissipation of energy occurs at relatively small flow scales. This detail is extraneous in terms of testing the correlation algorithm performance, but is important when using the algorithm in the consideration of sub grid filtering in PIV measurements (chapter 6).

All velocity components are correlated in time according to the single defined integral timescale; \( T_{L11} = T_{L22} = T_{L33} \) which satisfies the following equation in the simulation:

\[ R_{11}(x, \tau) = \exp\left( -\frac{\pi(\tau)^2}{4(T_{L11})^2} \right) \]

Notice in the above equation that the temporal separation, \( \tau \), has replaced the spatial separation distance, \( \Delta x \), and likewise the integral timescale has simply replaced the integral lengthscale, compared to the earlier formulae. Therefore the four spatial velocity correlation functions are satisfied at any point in time, and at any point in space the three autocorrelation criteria are satisfied.
4.4.1 Synthetic Velocity Data: Code Modification

Although it was possible to test the correlation algorithm using the data generated in the original three component two dimensional form described above, it would be more advantageous to generate the data akin to the three planar orientations that would be acquired using conventional two component, two dimensional PIV in Cartesian space. The data was therefore modified and output in the form shown in figure 4.5.

![Figure 4.5: Synthetic Data Co-Ordinate System and Data Planes (Modified Format)](image)

Now only two velocity components are contained in any one plane; e.g. $u_1$ and $u_2$ are contained in the $(x_1, x_2)$ plane. It must be realised that although the above figure implies it, the three velocity components are not actually generated in three dimensional computational space. The data is still generated as shown in the original figure with the three velocity components being generated in two dimensional space on the $(x_2, x_3)$ inlet plane. The data contained at different positions in the $x_1$ direction comes from different time steps. Therefore the relationship of the primary velocity at two points separated by a distance $\Delta x_1$ in space, and $\Delta T$ in time is given by:

$$U_1(x + \Delta x_1, t) = U_1(x, t - \Delta T)$$

This means that the flow does not evolve or decay as it passes through $x_1$ space – a perturbation will be seen to be completely unchanged as it passes through the $(x_1, x_2)$ or $(x_1, x_3)$ planes. Therefore the longitudinal integral lengthscale of the streamwise velocity
Correlation Functions

component is identical to the integral timescale. Although this simplification is not strictly representative of a real flow, it was necessary to enable the generation of the synthetic data in a suitable form, and it does not hamper the usefulness of the data in assessing the validity of the correlation algorithm. In fact it enhances assessment of the CCF because flow in the streamwise direction is not complicated by the decay of the flow with time.

4.4.2 Prescribed Synthetic Velocity Data Sets

Two data sets were created, where Set 1 contains 1024 consecutive synthetic data planes, and Set 2 contains 1024 temporally ‘uncorrelated’ data planes. Each data plane is a 62x62 grid of velocity vectors – the dimensions chosen to mimic a PIV vector grid. The data planes in Set 1 are output from the code at every consecutive timestep, and therefore each successive plane is strongly correlated with those immediately upstream and downstream of it. It is used to validate the correlation algorithm by comparing the calculated correlations of the synthetic data set with those of the prescribed model functions. Set 1 is also used to compare the calculated timescale with that prescribed in the code (see table 4.1); temporally correlated data is necessary to construct autocorrelation functions with good resolution. Set 2 contains data outputted from the synthetic code every 60\textsuperscript{th} timestep, and therefore given the relatively small prescribed timescales, this data set contains planes which are essentially uncorrelated in time. This second data set is used to compare the calculated lengthscales with those prescribed in the code, each data plane containing spatially correlated velocities that are necessary to construct spatial velocity correlation functions with good resolution. Set 2 is also used to assess the convergence of the SVC towards the model function with increasing number of independent samples.

The generated synthetic data were chosen to represent a typical turbulent flow with a dominant (primary) flow direction, U\textsubscript{1}. The time and length scales are integral values, which are discussed in detail in chapter 5. In the following sub section, the calculated integral time and length scales are ensemble averages of 64 individual scales from random locations across the synthetic data domain.
It should be noted that the time average velocities and Reynolds Stresses were also calculated for the outputted synthetic data, and in all cases agreed with the prescribed values to within the expected tolerances as defined in section 3.2.5.

4.4.3 Validation Results: Autocorrelation

Figure 4.6 shows a typical comparison of the prescribed autocorrelation model and that of the calculated data for both the streamwise and normal velocity components. Note that the x-axis shows the temporal separation normalised by the integral timescale. It is clear from the plot that the calculated correlation closely matches that of the model input, confirming the validity of the algorithm. The slight difference between the model and the outputted data is due to the finite sample size of the synthetic data set, which also accounts for the random oscillation about the x-axis with large time separations (see figure inset). Table 4.2 shows the excellent agreement between the specified integral timescales and those calculated from the data. Note that all three integral timescales are quoted despite $T_{L_{11}}=T_{L_{22}}=T_{L_{33}}$ in the definition of the synthetic data characteristics.

<table>
<thead>
<tr>
<th>Percentage difference compared to prescribed values</th>
<th>$T_{L_{11}}$</th>
<th>$T_{L_{12}}$</th>
<th>$T_{L_{33}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$(x_1,x_2)$ plane</td>
<td>+2.01%</td>
<td>+0.53%</td>
<td></td>
</tr>
<tr>
<td>$(x_2,x_3)$ plane</td>
<td></td>
<td>-0.99%</td>
<td>-1.09%</td>
</tr>
<tr>
<td>$(x_1,x_3)$ plane</td>
<td>-1.12%</td>
<td></td>
<td>+0.54%</td>
</tr>
</tbody>
</table>

Table 4.2 Comparison of Integral Timescale Data
4.4.4 Validation Results : Spatial Velocity Correlation

Figure 4.7 shows a comparison of the model and calculated SVC map for the transverse velocity component \( R_{33} \) at the inlet plane \((x_2, x_3)\). Excellent agreement can be seen between the maps, hence validating the SVC calculation algorithm. Taking a one dimensional profile through the origin of the SVC map in the \( x_3 \) direction gives the SVC curve shown in figure 4.8. Also shown on this curve are equivalent SVC curves calculated using smaller sample sizes. The convergence of the curve towards the model with increasing sample size is clearly evident, and highlights the need for a large number of measurements when calculating correlation coefficients. A trend is evident in the convergence of the calculated correlation curve with respect to the prescribed model. The absolute differences with respect to the model for all points constituting each curve were recorded and the maximum, average, and standard deviation values recorded, as shown in figure 4.9. Standard error estimates have been discussed in chapter 3 for the mean velocity, r.m.s., and normal stresses, which have the following form:

\[
\varepsilon = \pm z f(u_{\text{true}}) f(N)
\]

The trends shown in figure 4.9 also correspond to the above form, and the error band curve of the standard deviation is almost exactly as follows:

\[
\varepsilon = \pm z \left( \frac{1}{\sqrt{N}} \right)
\]
Fig. 4.7 SVC map of the Transverse Velocity Component ($R_{33}$)

Fig. 4.8 SVC Profile and Convergence Towards the Model Curve with Increasing Sample Size ($L_{33\,\text{true}} = 4.30$)
The lack of a dependence upon \( u' \) in the formula is logical since the correlation function is a dimensionless quantity. The trend is also seen to hold for the convergence of the ACF and CCF calculations, and is extremely important because it provides a means of estimating the background correlation levels in real experimental correlations. It therefore allows the user to confidently say that a calculated correlation is associated with the fluid dynamic behaviour when the values exceed the expected background levels for the utilised sample size.

Like the integral timescales, the lengthscales compare very well with the prescribed values, and the percentage differences are given in table 4.3 below. It should be borne in mind that the accuracies quoted used ensemble averages of 64 calculated lengthscales, and they are not subject to experimental errors and the finite accuracy with which the PIV measurements can be made. Therefore one should expect the lengthscales calculated from experimental data to have somewhat larger error bands, and section 5.3 shows that 95% confidence bands of around ±14% can be expected.

<table>
<thead>
<tr>
<th>Percentage difference compared to prescribed values</th>
<th>( ^1L_{11} )</th>
<th>( ^2L_{22} )</th>
<th>( ^3L_{33} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>((x_1, x_2)) plane</td>
<td>+3.42%</td>
<td>+0.74%</td>
<td></td>
</tr>
<tr>
<td>((x_2, x_3)) plane</td>
<td></td>
<td>-1.77%</td>
<td>+0.13%</td>
</tr>
<tr>
<td>((x_1, x_3)) plane</td>
<td>+0.13%</td>
<td></td>
<td>+0.75%</td>
</tr>
</tbody>
</table>

Table 4.3 Comparison of Integral Lengthscale Data
4.4.5 Validation Results: Cross Correlation Function

The success of the CCF version of the algorithm is assessed by examining correlation maps in the \((x_1, x_2)\) plane at successive time steps. Figure 4.10 illustrates the translation of the \(R_{11}\) correlation map across the domain with time. The map is seen to translate 8 units in 5 time steps – due to the convection velocity of 1.6 and unit time step. The turbulence does not evolve or decay with time as it passes through the plane, due to the manner in which the data is outputted (see section 4.4.1). In a real flow, the value of \(R_{11}\) would decay with time due to the turbulent energy in the flow.

![Translation of CCF map with time shift, \(\tau\).](image)

4.5 Experimental Examples

A 'benchmark' experimental configuration is used to demonstrate the usefulness and applicability of the correlation calculations to real flow situations. The benchmark flow scenario utilised is that of the flow over a wall mounted obstacle of square cross section. This experiment was chosen because it was relatively simple to setup in the water analogy facility (see chapter 2) and yet comprises many features including separated flow regions, recirculation, and strong velocity gradients. The region downstream of the obstacle is concentrated upon and resembles the flow topology of a backward facing step, which is considered in many fundamental studies and test cases in the literature (e.g. Kostas et al., 2002). The geometric setup was detailed in section 2.1, and consists of a continuous square cross section (7mm x 7mm) hoop fixed to the inner annulus wall of the blank test section. In doing so, essentially a closed channel two-dimensional passage was
Correlation Functions

created with the added benefit that the axisymmetric nature of the section results in zero end wall effects. The geometry is described in figure 4.11 along with the time average velocity field and an example of the instantaneous flow. In the time average data, in addition to the large scale recirculation bubble in the wake of the step, two smaller recirculation zones are evident close to the step forward and rear edges. The upstream flow is accelerated around the sharp leading edge of the step which causes it to separate upstream of the x=0mm position. Following the large recirculation bubble, the flow reattaches to the inner wall at x=36.5mm. Note that the bulk average upstream velocity was $U_{ref}=0.340\text{m/s}$ giving a Reynolds number (based on channel height, $h=20\text{mm}$) of 7620.

Fig. 4.11 Step Flow Geometry (left), Time Average Flow (upper right) and Example of Instantaneous Flow (lower right)

The three points marked on the velocity maps form a focus of the ‘benchmark’ investigations in this and subsequent chapters, and were chosen because of their very different characteristics:

A : A point close to the centre of the main recirculation bubble where large scale motions are expected to dominate the flow giving high turbulence levels but near zero mean velocity.
Correlation Functions

B: A point within the shear layer separating the recirculation zone and accelerated free-stream flow, where the velocity gradient is high and small scale turbulence will dominate.

C: A point in the ‘free-stream’ region. This provides comparatively low turbulence contrasting points A and B.

PIV data was collected at four different magnification levels, and processed using both 32x32 and 16x16 pixel final interrogation cells; both schemes using adaptive multi-pass gridding starting at 64x64 pixel cells on the first pass. All PIV processing utilised deformed grids and second order schemes, as detailed in chapter 3. The varying levels of magnification and choice of two different final cell sizes was primarily for the purpose of investigating sub grid filtering effects discussed in chapter 6, where the reader will also find extensive details of the flow statistical properties. The correlation data shown here were calculated from the data at M=0.164 with 32x32 pixel interrogation cells.

4.5.1 Experimental Results: Autocorrelation

Although the 15Hz sample rate of the PIV is unsuitable for calculating ACFs in the majority of realistic engineering flow scenarios, point C has a relatively long timescale for the streamwise ($U_1$) velocity component, and figure 4.12 shows that the ACF ($R_{11}$) from the PIV data compares well with that calculated using a 1kHz LDA time history series taken at an identical location. However it must still be realised that the lack of temporal resolution would result in a poor estimate of the integral timescale from the PIV, and is therefore no replacement for the point measurement technique in this case.

![Fig. 4.12 ACF ($R_{11}$) Comparison for Point C](image)

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4.5.2 Experimental Results: Spatial Velocity Correlation

Figure 4.13 shows a SVC map ($R_{22}$) calculated within the main recirculation bubble downstream of the step. The positively correlated region extends across the shear layer separating the recirculation bubble and free stream, and is somewhat asymmetric in the normal direction. The extent of the correlated area suggests a perturbation of the normal velocity in the recirculation zone influences almost the entire height of the channel. The expected background correlation levels for this map (created using 520 samples) is ±0.113 using the formula derived in section 4.4.4, and accordingly all values of $|R| < 0.1$ are omitted from figure 4.13 via the use of appropriate contour colours. This was the approach for the presentation of all correlation maps shown in later chapters. The figure indicates that the negatively correlated regions are real, and important. It also shows the correlated flow to extend across the shear layer, indicating the flow in the wake of the step influences that in the free stream.

![Fig. 4.13 SVC ($R_{22}$) near Point A](image)

Section 3.5.1 introduced the PDF conditioning technique, suggesting that it could provide an efficient means of understanding the flow topology behind correlation maps. The technique effectively isolates the motions associated with the tails of the velocity PDF at a point of interest (see figure 3.11). Using the radial velocity at origin of the correlations indicated in figure 4.13 as the point of interest (o) to drive the ‘conditioning’ allows a demonstration of the power of the technique. Initially the raw vector fields are used to produce PDF conditioned ensemble averages. For convenience the formulae are
Correlation Functions

reproduced below. A value of K=1.50 was used to isolate the PDF tails giving around 30 (from a total of 520) vector maps to create the ensembles;

\[ \bar{U}_{ca^-}(x) = \frac{1}{N_{ca^-}} \sum_{i=1}^{N} a_{i^-} U(x), \]
where
\[ a_{i^-} = \begin{cases} 0 & U(o) > U(o) - Ku'(o) \\ 1 & U(o) \leq U(o) - Ku'(o) \end{cases} \]
and
\[ N_{ca^-} = \sum_{i=1}^{N} a_{i^-} \]

(similar formula for ca+; notice that the '2' subscript is missing from the U which would indicate the normal velocity component is the 'subject')

The resulting vector fields are shown in figure 4.14, and have effectively isolated the motions associated with the shedding of the recirculation bubble in the wake of the step. The vector plots are therefore a form of pseudo phase locked information.

Fig. 4.14 PDF Conditioned Ensemble Averages; ca- (top), ca+ (bottom)
(Scales as figure 4.11)
Correlation Functions

By creating ensemble averages of Reynolds decomposed versions of PDF conditioned vector fields using the formulae below (and the same value of K), a clear understanding of the correlation maps is gained, as shown in figure 4.15 where the resulting vector fields are superimposed on the correlation maps. The vectors reflect the opposite sign of the correlated fluid adjacent to that surrounding the point of interest and show that the recirculation in the wake of the step clearly influences the flow in the main stream, which assumes a vortical motion of the opposite sense.

\[
\bar{U}_{ca-}(x) = \frac{1}{N_{ca-}} \sum_{i=1}^{N_{ca-}} a_{ci} \left( U(x)_i - \bar{U}(x) \right)
\]

![Fig. 4.15 PDF Conditioned (Reynolds Decomposed) Ensemble Averages Superimposed on Spatial Velocity Correlation Maps; ca- (top), ca+ (bottom)](image)

The analysis has provided information regarding the underlying fluid dynamics of the flapping shear layer seen in figure 4.11, and a wealth of quantitative information above and beyond what point measurement techniques can offer. This type of analysis proved invaluable in the combustor investigations shown in chapters 7 and 8.
4.5.3 Experimental Results: Cross Correlation Function

The convection of correlated fluid within the shear layer is evident in figure 4.16. The topology of the $R_{11}$ correlation map changes as it is convected downstream, unlike the synthetic data shown in figure 4.10. This would be expected since the turbulence levels in the shear layer are relatively high, and the fluid dynamic characteristics change quite significantly as the shear layer spreads. Between $\tau=0\Delta T$ and $\tau=1\Delta T$ the correlated fluid is seen to shift approximately 20mm, giving a convection velocity of 0.3m/s which agrees well with the actual time average velocity. By $\tau=2\Delta T$ the majority of the correlated fluid has left the measurement domain in the streamwise direction, and only the tail of the correlated flow is still identifiable.

![Fig. 4.16 CCF ($R_{11}$) within the Shear Layer ($\tau=0\Delta T$, $1\Delta T$, $2\Delta T$)](image)

4.6 Closure

The background, usefulness, and mathematical formulae describing correlation functions have been presented. A method for calculating the correlation functions of velocity data obtained using the PIV system has been developed and validated through the use of synthetic data with known correlation characteristics. A trend has been identified for the statistical convergence of the correlation functions towards that of the prescribed models, and it has been shown that for a suitably large data set size the calculated correlation function matches that of the inputted model function very closely. This convergence trend will enable accurate estimates of the expected background correlations in experimental results, allowing truly correlated flow to be identified with confidence.
The algorithm described here has been used extensively in the examination of the flows studied experimentally and examples were given for the 'benchmark' flow over a step test case. The ability of the PDF conditioned averaged velocity fields (introduced in chapter 3) to efficiently explain the topology associated with the correlation maps was also established as an extremely valuable analysis tool. Many interesting features have been identified using correlations and conditional averaging which has enabled a greater understanding of the nature of turbulent and unsteady flows in gas turbine combustor type geometries. These examples can be found within chapters 7 and 8. It should also be appreciated that the methodologies presented in this chapter could equally be applied to other planar velocity data, such as that from computational simulations such as LES and DNS.

Prior to the results sections, chapter 5 extends the use of correlation functions to obtain integral time and length scales, which have been discussed briefly in this chapter. Chapter 5 describes how obtaining integral scales can allow the calculation of other important fluid dynamic characteristics and turbulent scales, and further shows the wealth of quantitative information that can be obtained from planar velocity measurements.
Chapter 5

SCALES OF TURBULENT FLUID FLOW
Chapter 4 discussed utilising PIV to calculate correlation functions in the temporal and spatial domains. The spatial velocity correlation was discussed in depth, and it was shown that contour maps can be produced to show the correlation between the velocity at a point with every other point in the domain. In doing so a characteristic size and shape of the turbulent eddies present at the point of interest are identified. Data can thus be obtained from PIV measurements which quantify the scales of the turbulent fluid motion. Quantitative information regarding turbulent length scales is extremely important in the analysis of fluid dynamic phenomenon, and can be used to identify the extent of the energy containing motions and dissipation range, as well as allowing the calculation of other important fluid dynamic properties, such as dissipation rate and Reynolds numbers based on the scales of turbulence. This type of information is extremely useful when attempting to validate computational simulations. Also, as will be seen in the following chapter, it is essential to be able to calculate the integral length scales when correcting for sub grid filtering effects.

5.1 Turbulence Scales

A range of scales of turbulent motion exist in any fluid flow, and in any domain the energy in the various motions is transferred and eventually dissipated. Central to the concept of scales within turbulent motion are the Kolmogorov hypotheses, which stem from the idea of the energy cascade; a concept introduced by Richardson (1922): Kinetic energy enters the ‘turbulence’ of the flow at the largest scales via the production mechanism. The energy is then transported through progressively smaller scales (by means of inviscid motion) until at some point it is dissipated by viscous action. This transfer from large to small scales is also known as ‘forward scatter’ of turbulence. There are situations where ‘backscatter’ occurs – where energy is produced at small scales and transferred to larger scales, although this is the rarer event, examples including the backscatter of energy from eddies in boundary layers. The assumption here is that the transfer is exclusively via forward scatter, as is the case in isotropic turbulence.

An ‘eddy’ eludes precise definition, but is conceived to be a turbulent motion, localised within a region of size $l$, that is at least moderately coherent over this region. (Pope, 2000)
The largest eddies contain any number of progressively smaller eddies. The largest eddies are unstable and undergo a breakdown, progressively 'cascading' their energy through to smaller motions. And so the process repeats this energy cascade, until the eddy motion becomes stable, and small enough so that viscosity acts to dissipate the kinetic energy at motions of order the Kolmogorov microscales ($\eta$) of the flow. Kolmogorov (1941) hypothesised that both the associated velocity and timescales decreased with eddy size.

Kolmogorov states that the large eddies are anisotropic, being affected by the boundary conditions of the flow, i.e. they have directional properties or 'shape' – a fact that is confirmed in several of the SVC maps in chapter 7. However, through the cascade process the directional properties of the eddies are lost, and so 'at sufficiently high Reynolds number the small scale turbulent motions are statistically isotropic' (Kolmogorov's hypothesis of local isotropy). The notion of local isotropy with respect to turbulent motion is a key idea that will be returned to later. Kolmogorov also said (in his first similarity hypothesis) that all geometrical information of the largest eddies is lost, and so the smallest scale motions are essentially universal in every high Reynolds number flow, and determined by the kinematic viscosity, $\nu$, and the dissipation rate, $\varepsilon$. This is known as the universal equilibrium range and it is the inertial sub-range within this where the classic -5/3 gradient exists in the energy spectrum.

An important definition is that of the characteristic 'turbulence' lengthscale $L_c = k^{3/2}/\varepsilon$ (k being the turbulent kinetic energy – see chapter 2), which Pope (2000) describes as 'characterizing the large eddies.' $L_c$ is of the same order as the physical boundaries of the flow, $L$, and the integral lengthscale, $L$ ('L' being used to refer to the [longitudinal] integral lengthscale in general discussion – clarification of this scale and its co-ordinate frame being given later in this section). $L_c$ is mathematically related to the longitudinal integral lengthscale, and for typical Reynolds number flows is approximately twice the size – a relationship discussed later in this chapter. Considering an arbitrary turbulent flow, the ratio $\eta/L_c$ decreases with increasing Reynolds number and so for very high Reynolds number flows the largest range of scales relative to $\eta$ occurs. Therefore, there exists a range of scales much smaller than $L_c$ but much larger than $\eta$, and Kolmogorov
states (second similarity hypothesis) that in this range 'the motions have a universal form that is uniquely determined by $\epsilon$, independent of $v$.'

Pope (2000) provides lucid evidence that the bulk of the energy containing motions exist over the range of lengthscales $6L > l > 1/6L$. Therefore a separation is introduced for high Reynolds number flows between the large scale anisotropic eddies, and the existence of the isotropic smaller eddies; $l \approx 1/6L$ (note that the $1/6$ coefficient is an approximate value applicable to very large Reynolds number flows). In the range $l$ approximately less than $1/6L$ the scales are small enough to allow the turbulent motions to maintain a dynamic equilibrium with the energy transfer rate established by the larger eddies. Pope also describes a separation between these and the smallest motions, thus splitting the universal equilibrium range into the 'inertial sub range' where $l$ is approximately greater than $60\eta$ (but less than $1/6L$) and the 'dissipation range' for which $l$ is approximately less than $60\eta$. A Graphical interpretation of Pope's delineation of the scales of turbulent motion is given in figure 5.1 below:

![Diagram of Turbulent Fluid Flow Scales](modified from that of Pope, 2000)

Note that the numeric coefficients of $60$ and $1/6$ are approximately appropriate for very high Reynolds number flows. In lower Reynolds Number flows to be seen later, greater overlap between the dissipation and production ranges exists.

**Fig. 5.1 Delineation of the Scales of Turbulent Fluid Flow**

*(modified from that of Pope, 2000)*
It is logical for flows exhibiting forward scatter of energy that the rate of transfer from the large scales determines the constant rate of energy transfer through the inertial subrange, thus the rate at which the energy leaves the inertial subrange and enters the dissipation range, and hence the overall dissipation rate, $\epsilon$. In cases where backscatter of energy occurs, it is possible for the energy leaving the inertial subrange to be greater than the energy being transferred from the 'energy containing' range, causing the integral lengthscale to grow.

### 5.2 Integral Length Scales

The integral scale is 'characteristic of the larger eddies' (Pope, 2000). This statement probably should read as 'a characteristic... ' because it has already been defined that the turbulence lengthscale $L_\epsilon=k^{3/2}/\epsilon$, is 'the lengthscale characterizing the large eddies.' In reality both lengthscales characterise the geometry of the larger turbulent motions, because the largest eddies represent the bulk of energy containing motion, and this energy containing motion exists over the range $1/6L$ to $6L$. However in this section, we are concerned only with the integral scale, as it is directly calculable from the spatial velocity correlations (see chapter 4 for a description of correlation terminology and formulae).

The integral lengthscale is mathematically defined as the integral with respect to distance, of the SVC; $R_{ij}(x,r)$. In calculations of the integral lengthscale $i=j$. The longitudinal integral lengthscales are defined as the integral of the spatial velocity correlation along a co-ordinate direction, $k$, parallel to that of the velocity component. Therefore $i=j=k$ in the following equation:

$$kL_{ij}(x,r) = \int_{0}^{\infty} R_{ij}(x,r)dr$$

Hence the longitudinal integral lengthscale of the streamwise velocity component, $U_1$ is;

$$L_{11}(x,r) = \int_{0}^{\infty} R_{11}(x,\Delta x_1)dx_1$$
The lateral integral scale is similarly defined but now the integral is calculated along a co-ordinate direction perpendicular to that of the velocity component. I.e. \( k \neq i = j \) and so two lateral integral length scales exist for each velocity component when considering three dimensional space. The equation below defines the lateral integral lengthscales for the primary velocity component:

\[
L_{11}(x,r) = \int_0^\infty R_{11}(x,\Delta x_2)\,dx_2 \quad \text{and} \quad L_{11}(x,r) = \int_0^\infty R_{11}(x,\Delta x_3)\,dx_3
\]

Therefore three longitudinal integral lengthscales, and six lateral integral lengthscales exist in three dimensional space. However, as the PIV system employed is a planar measurement technique only two lengthscales are obtainable for each velocity component in that plane. Nevertheless, re-orientating the measurement domain relative to the experiment axis allows all nine integral scales to be obtained in theory.

It should be realised from the formulae that, in its true mathematical definition given above, the integral lengthscale is calculated from the origin to an infinite displacement, whereas as a matter of necessity in practical calculations, the standard practice is to integrate over the distance from the origin to the first crossing point – the value of which is actually very close to the ‘origin to infinity’ calculation for real flows. An important feature of the integral length scale is that it describes the geometry of the eddies which are themselves anisotropic and retain the directionality described by Kolmogorov. An indication of the isotropy of a given flow is that for isotropic turbulence, the ratio of the longitudinal to lateral scale is 2.0 (Hinze, 1959).

The early discussion in this chapter referred to the integral lengthscale, \( L \) – this simple symbology being adopted because when a ‘lengthscale’ is referred to, one commonly means the (longitudinal) integral lengthscale. It has however been defined that nine integral scale exist in three dimensional space. A variety of nomenclature is used in the literature to describe the integral lengthscale(s). Pope (2000) uses ‘\( L_{11} \)’ as a general description of the longitudinal integral lengthscale because of assumptions of isotropic turbulence, whereby \( L_{11} = 2L_{22} = 3L_{33} \). In the same vain, ‘\( L_{22} \)’ describes the lateral integral lengthscale because if isotropy is assumed \( L_{22} = 2L_{11} \). Hinze (1959) and Hoest-Madsen
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and Neilsen (1995) use $\Lambda_f$ and $\Lambda_g$ to describe the integral length scales, whereas Libby (1996) uses $L_{lp}$ and $L_{ll}$ for the parallel (longitudinal) and lateral scales. However, all of these nomenclatures are only suited to isotropic homogeneous flows and can confuse the descriptions of real three dimensional non-isotropic flows. To avoid such confusion in this thesis, the full description ($kL_{ij}$) will be used to describe the integral length scales in most formulae, and is consistent with ESDU (2001). However, a shorthand notation is also utilised when longitudinal and lateral integral scales need to be referenced which are not particular to a certain velocity component or axis, in the form of $L_f$ and $L_g$ respectively. In general discussions ‘$L$’ will continue to be used to describe the (longitudinal) integral lengthscale.

5.3 Calculation of the Integral Scales Using PIV Data

It has been discussed that taking one dimensional profiles of the spatial velocity correlation function through the origin in the primary and secondary flow directions allows the integral to be calculated under the curve, thus giving the integral lengthscales. PIV data allows the integral lengthscale to be calculated across the entire planar field of view for a given experiment. However, there are a number of critical issues that have to be addressed in actually utilising the SVC maps generated from the PIV data to calculate the integral length scales:

1. The calculation of each single SVC map is time intensive (with $10^8$ floating point operations required per map – taking approximately 15 minutes on a dual 800MHz processor Pentium III computer). A map centred on every point (63x61) in the domain would be required for an entire description of the lengthscales across the measurement domain.

2. The integral lengthscale calculation requires SVC data on both the positive and negative distance offset ($\Delta x_k$) axis, and requires data to the $x$ axis crossing point for the integral.

3. Unlike the (temporal) ACF which is always symmetric about the y-axis, this is not always the case for the SVC. For example, obstacles downstream in the flow can make the positive (downstream) part cross the axis earlier than the upstream part.
Point 1 is relatively easily addressed: only SVC data is required through the origin in the two in-plane axis directions to calculate the integral values. The entire two dimensional correlation field map is not required and therefore instead of 63x61 (3843) points for each origin location, only 63+61-1=123 points are calculated, reducing the number by 97% and hence drastically reducing computational time.

The second point is more difficult to manage. For large fields of view, where the data plane is much larger than the integral lengthscale, curtailment of the SVC function would only be a problem towards the measurement boundaries where fluid passed in and out of the domain. But for small relative fields of view curtailment was inevitable. This issue would be most pronounced in the streamwise direction where the longitudinal integral lengthscale is naturally larger. In the transverse direction, the issue is actually aided by the fact that the flow is often physically bounded by walls where, by definition, the correlation function must be zero. To manage the issue of incomplete SVC data, two options existed:

i. A curve fitting routine to the available data

ii. An estimate of the ‘missing’ data contribution to the integral

A curve fitting routine was employed by Bridges and Wernet (2002) with some success, but its implementation has the natural consequence of smoothing the SVC function. In Bridges and Wernet the curve fitting algorithm is employed everywhere in the domain, and not exclusively in regions of curtailed data. For the current project it was decided that, where available, the actual SVC data would be employed in the calculation with some estimate for the missing SVC curve where curtailed. It should be realised that in many cases, the entire SVC curve is present and so no estimate would be required. The missing integral value was calculated based on data given in Wygnanski and Fiedler (1969) for a typical SVC function. This model SVC function is shown in figure 5.2 alongside a typical experimental curve. The model function is a simple exponential of the form $R_{ij} = e^{\alpha}$, and gives a simple but reliable estimation of the missing integral contribution. Figure 5.2 illustrates that the estimation is made by shifting the model function horizontally to match the experimental data at the curtailment point. The actual integral is then a summation of that using the experimental data available and the model
function 'downstream' of the curtailment point. In figure 5.2 the experimental data is artificially curtailed to illustrate the procedure. The actual integral lengthscale is 5.1% greater than that calculated using the estimated portion, therefore confirming that the methodology gives a reasonable estimate of the lengthscale in the event of the correlation function being curtailed.

Fig. 5.2 Illustration of Estimate of Curtailed SVC Curve

The shape of the model SVC function in figure 5.2 is slightly different to the form of that prescribed by the synthetic model in chapter 4. Figure 5.2 shows a function most applicable to higher Reynolds number flows, which is therefore more representative of real engineering scenarios. Indeed, the form of the curve is typical of those seen in the real flows discussed in later chapters.

With respect to the third issue (asymmetric functions) of integral length scale calculation, one could argue that two different lengthscales should be quoted; the 'upstream' (negative part of the Ax axis) and 'downstream' values. This would be a legitimate process, and indeed the level of symmetry is actually a form of testing the data against Taylor's Frozen Turbulence hypotheses. However the integral lengthscale refers to a characteristic of the eddy size present (centred) at that point, and therefore the quoted...
lengthscale should be the average of the upstream and downstream parts of the integral of the SVC function. The fact that the upstream and downstream parts may be different does however indicate that one will gain valuable information by examining the spatial SVC maps. Where the upstream and downstream SVC functions are very different, it is an indication that the eddy undergoes a significant change in passing through the measurement region. An interesting example of such a case is shown in figure 5.3 for the flow over the step, introduced in the previous chapter. The figure shows that, due to the upstream obstacle facing the flow, the integral of the streamwise velocity downstream of the origin is significantly smaller than the upstream side. In fact, the integral value of the negative part of the axis is over three times that calculated using the positive part. Clearly the correlation value must be zero at the face of the step and the close proximity of the origin results in a small integral value, whereas there is no such problem with the upstream portion. An alternate viewpoint may be that the integral should be performed along the (time average) stream line of the flow, which curves upwards and around the step. Such an integral may prove to be symmetric, but is prohibitively computationally expensive to calculate via an automated process. One would also be faced with the challenge of developing a notation system.

![Figure 5.3 Asymmetric SVC example – (R11) upstream of the Step](image)

The average of the upstream and downstream axis integrals is hence used in calculating the integral lengthscales. However, in the regions close to the domain (inflow and outflow) edges, one part of the axis will lack a significant portion of the real SVC function, it being curtailed close to the origin. Tests showed spurious results (both over and underestimations) were possible close to domain edges due to the calculated...
lengthscale relying on estimated SVC functions for a large proportion of half of the axis. Therefore, based on the examination of several sets of lengthscales calculated from experimental data, a method was developed that utilised a 'confidence level' related to the amount of real (calculated) SVC function data available for the respective part of the axis, and took the form:

\[
\begin{align*}
C &= 0 & \text{SVC}_{\text{curtailed}} > 0.9 \\
C &= 1.125 - (1.25 * ACF_{\text{curtailed}}) & 0.1 \leq \text{SVC}_{\text{curtailed}} \leq 0.9 \\
C &= 1 & \text{SVC}_{\text{curtailed}} < 0.1 \\
\end{align*}
\]

(where C=confidence coefficient)

The above means that if the function is curtailed at anything greater than a value of SVC=0.9, no confidence can be attributed to the integral for that part of the axis because it relies too heavily on an estimated curve. If the function is curtailed at less than 0.1, entire confidence can be placed on the integral, because the estimated portion is so small. Between the two values, a linear relationship between confidence and the curtailed value exists. The actual lengthscale is then calculated from the weighted average, dependant upon confidence levels.

\[
^kL_{ij} = (C_{-ve} \, ^kL_{ij\, -ve} + C_{+ve} \, ^kL_{ij\, +ve})/(C_{-ve} + C_{+ve})
\]

(Where +ve and -ve subscripts refer to parts of the separation distance axis)

The weighted average calculation does mean that near domain edges, the lengthscale is essentially calculated using only the upstream or downstream part of the axis, which contravenes the earlier statement with respect to it having to be an average. However, this method provides more reliable and fluid dynamically consistent values at the domain edges than simple averages. The user must simply be aware of this in interpreting the data near domain inflow and outflows.

Figure 5.4 shows the calculated integral lengthscales \(^3L_{33}\) compared to the true value along a line of constant \(x_2\) for the synthetic data introduced in chapter 4. The corresponding SVC data integrated to arrive at the value of \(^3L_{33}\) at \(x_3=31\) was shown in figure 4.7. Although the correlation functions used to calculate the integral lengthscales from the synthetic data are unlikely to be highly asymmetric (as they may be in real experimental flows), the variation is thought to be fairly representative of experimental data.
Fig. 5.4 Profile of Variation of Lengthscale ($l_{33}$) Across Domain (at $x_2=31$)

The variation is typical of that across the measurement domain for all lengthscales calculated from the synthetic data. The variation exists because the SVC function map at every ‘origin’ across the domain is not identical to the model. Therefore the integral of the SVC curve at varying origin locations will result in some differences in the calculated lengthscales. From the synthetic data sets, the standard deviation of the error of the calculated lengthscale, as a percentage of the actual lengthscale was 7%. Therefore one has 95% confidence that the calculated lengthscale lies within ±14% (i.e. 2 standard deviations) of the true value, given an N=1024 sample size. However, chapter 6 highlights a problem regarding the calculation of integral length scales from PIV data which is related to the cell size relative to the scales of the flow. This must also be consulted when using experimental data.

5.4 Calculation of Other Scales and Fluid Dynamic Properties

It has so far been proven that PIV data allows the calculation of the following statistical fluid dynamic data (considering only 2 velocity components):

- Mean Velocities, $\bar{U}_1$, $\bar{U}_2$
- RMS Velocities, $u_1'$, $u_2'$
- Integral Length scales, $L_{11}$, $L_{22}$, $L_{12}$
Scales of Turbulent Fluid Flow

This data puts the PIV experimentalist in a powerful position, in that theoretically from these data, other important fluid dynamic scales and properties are calculable. In order to do this though, it is necessary to apply certain assumptions and generate some of the data via iterative calculations. The main assumption is that of the existence of isotropic turbulence. Although this assumption is not strictly correct for practical engineering flows, it will be demonstrated that it provides the most appropriate means (within the scope of this project) of progressing insight into the flows of interest. And before completely dismissing the assumptions of isotropic turbulence, one should consider Kolmogorov’s hypotheses which include the statements that only the largest turbulent scales of the flow are anisotropic, whereas for wavenumbers beyond the energy containing range, local isotropy prevails and flow history and geometry is lost. In fact, some of the flows considered in this project are relatively low Reynolds number scenarios – especially the experiments utilising water as the fluid medium. The calculation procedure using the relevant isotropic assumptions is detailed below.

Piirto et al (2001) gives the following equation for the estimation of dissipation rate:

$$\varepsilon = A (u'^3/L)$$

Where $A$ is a constant (equal to unity in isotropic flows) and $L$ is the physical geometric characteristic length in the flow (e.g. port diameter). The value of $u'$ is not related to a single velocity component direction but instead defined as $u' = (\frac{1}{2}k)^{\frac{1}{2}}$. The Reynolds Number based on $L_c$, can then be calculated using the estimated value of $\varepsilon$:

$$Re_{L_c} = k^2/(\varepsilon v)$$

Therefore an estimation for $L_c$ can be calculated (using the true value of $k$):

$$L_c = (Re_{L_c} v)/k^{\frac{3}{2}}$$

The lateral Taylor Length scale can also be calculated using the above values:

$$\lambda_g = (\sqrt{10})Re_{L_c}^{-1/2}L_c$$

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For isotropic turbulence the longitudinal Taylor Length Scale is related to the lateral one by $\lambda_f = (\sqrt{2})\lambda_g$. At this point it should be highlighted that, in theory, the PIV data could be used to directly calculate the Taylor Length scales using the following relationship:

$$\lambda_f = \lambda_{11} = \sqrt{\frac{\lambda_{u_1u_1}}{\frac{\partial u_1}{\partial x_1}}}$$

$$\lambda_g = \lambda_{22} = \sqrt{\frac{\lambda_{u_1u_1}}{\frac{\partial u_1}{\partial x_2}}}$$

However, it is the author's experience that the calculated Taylor scale using the above method is directly related to the separation distance, $\partial x$, of the velocity vectors (see figure 5.5). Saarenrinne et al (2001) experienced similar trends in their laboratory mixer experiments. This occurs because the Taylor lengthscale is based on the derivative of the spatial correlation function at the origin, and the formulae above are based on a Taylor series expansion. Obtaining the derivative at the origin is therefore very sensitive to vector separation and would require unfeasibly high spatial resolution in most practical PIV experiments. Using this approach is therefore unreliable, and hence the alternative iterative procedure is adopted as described.

![Fig. 5.5 Trend of Directly Calculated Taylor Length Scale for Point B of Step Flow](image)

The final step of the estimation stage is to calculate the Taylor Reynolds number based on the calculated estimates thus far:

$$Re_\lambda \equiv u' \lambda_g / v$$
This quantity \( \text{Re}_\lambda \) is important because there exists a relationship between the longitudinal integral length scale, \( L \) and the lengthscale, \( L_c \), which varies with \( \text{Re}_\lambda \) (for the model spectrum) according to figure 5.6.

\[
\begin{array}{|c|c|}
\hline
\text{Re}_\lambda & L/L_c \\
\hline
0 \text{ to } 80 & 2.60 \text{Re}_\lambda^{-0.340} \\
>80 \text{ to } 400 & 0.93 \text{Re}_\lambda^{-0.115} \\
>400 \text{ to } 1350 & 0.74 \text{Re}_\lambda^{-0.075} \\
>1350 & 0.43 \\
\hline
\end{array}
\]

**Fig. 5.6 Relationship Between the Longitudinal Integral Lengthscale, \( L \), and \( L_c \)**

(Pope, 2000)

A means of calculating a more accurate value of the Lengthscale, \( L_c \), is now available which is converged to within 0.01\% using an iterative procedure:

1. Use \( L \) and previously calculated \( \text{Re}_\lambda \) to re-calculate \( L_c \)
2. \( \text{Re}_{Lc} = (k^{1/2}/L_c)/v \)
3. \( \lambda_g = (\sqrt{10})\text{Re}_{Lc}^{-1/2}/L_c \)
4. \( \text{Re}_\lambda = u'\lambda_g/v \)

The longitudinal Taylor scale can be calculated using the appropriate assumption for isotropic turbulence \( (\lambda_d/\lambda_g = \sqrt{2}) \) and the Kolmogorov scales can also be calculated (Pope, 2000):

\[
\eta = L_c \text{Re}_L^{-3/4} \quad \tau_\eta = (v/\epsilon)^{1/2} \quad u_\eta = \eta/\tau_\eta
\]

And finally the dissipation rate is arrived at via the following equation:

\[
\epsilon = k^2/(\text{Re}_{Lc}v)
\]

This section has described the utilisation of quantities directly available from PIV data, to calculate a variety of important fluid dynamic properties, given the assumption of isotropy. The calculation process is given in the flowchart of figure 5.7.
Fig. 5.7 Flowchart of Turbulence Calculations
5.5 Closure

This chapter has discussed the calculation of the integral length scales from PIV data, and the measures necessary to ensure accuracy in their calculation across the measurement domain, especially at domain edges where curtailment of the spatial correlation can occur. The notation was also developed to allow proper definition of the nine possible integral length scales in three dimensional Cartesian space. Use of synthetic data showed that the integral length scale can be measured to within ±14% of its true value.

It has been shown that PIV can be used to calculate a number of quantitative fluid dynamic properties given some assumptions regarding the isotropy of the turbulence. Clearly isotropic turbulence does not exist in its purest form in practical engineering flows, but in many cases near isotropic turbulence does exist on a local scale - as hypothesized by Kolmogorov. Many theories have utilised this concept because it allows the derivation of other data to a reasonable degree of accuracy, e.g. Taylor's frozen turbulence theory. In this thesis, the process described is employed to calculate fluid dynamic data for the real experimental flows in the following chapters and gain valuable insight into the fluid dynamic characteristics. Without the assumptions of isotropy, this would not be possible and so the methodologies described here provide one of the few ways to obtain these type of fluid dynamic data from experimental results.

The formulae described in this chapter will also be used extensively in the following chapter regarding the effect of sub-grid filtering upon PIV data in turbulent flows.
Chapter 6

SUB GRID FILTERING
The PIV system employed throughout the research is a planar measurement technique, allowing series of instantaneous, two-component velocity data to be collected. Earlier chapters have demonstrated its power in quantifying instantaneous velocity fluctuations, correlation functions, and time average measurements, as well as a variety of fluid dynamic information that was previously extremely difficult to obtain experimentally. However, chapter 3 mooted the limitations associated with this technique, one of which is the important issue of Sub Grid Filtering (SGF). It will become clear in this chapter that the issue of sub-grid filtering has important consequences upon the calculated higher order statistics and lengthscales and that the PIV user must assess the effects of finite measurement volumes in their experiments.

6.1 What is Sub-Grid Filtering?

Sub-grid filtering occurs due to the way PIV calculates each individual velocity vector in the measurement domain. These velocities are calculated using the cross-correlation technique described in chapter 3. The calculation is performed over a finite area therefore computing a vector representative of the average (modal) translation of the cross correlation peak, as illustrated in figure 6.1. Note that the use of a light sheet of finite thickness means the local measurement domain for each vector calculation is in fact a volume. Although the discussion here neglects the third velocity component for simplicity, it should be easy to extend the arguments to three dimensions.

---

**Fig. 6.1** Illustration of Average Displacement Recorded by Each Interrogation Cell.
Providing that the particle images are of equal size and uniformly distributed within a 
given cell (which is usually approximately true), it is the modal particle displacement that 
is recorded for that cell, because the mode of the particle displacements produce the 
strongest correlation peak. Therefore, in essence the velocity calculated represents the 
average fluid movement within the measurement area, with the cell dimensions defining 
the upper limit of the spatial filter.

Indeed, virtually all velocity measurement techniques introduce an element of filtering by 
calculating the velocities over finite distances or times, but one must distinguish the type 
of filtering. The former paragraph introduced spatial filtering where an area (or volume) 
of fluid is taken as the measured value. The same comments could be made of pitot 
probe measurements, whereby it is the average pressure over the probe face that is 
recorded. There is also the issue of ‘dynamic averaging’ which chapter 3 discussed with 
respect to PIV measurements – the value of \( \Delta t \) having an impact on the averaging effect 
on particles following curved paths in space. Dynamic averaging in LDA measurements 
is dependant upon the distance (number of fringes required for a valid signal multiplied 
by the fringe spacing) over which the Doppler burst is recorded, and for Hot Wire 
Anemometry it is the time over which the voltage sample is recorded. LDA and Particle 
Tracking Velocimetry (PTV) do not suffer from spatial filtering because they consider 
only the movement of individual particles or ‘points’ of fluid. In contrast PIV performs 
calculations based upon the average movement of a group of particles, and hence the 
movement of a volume of fluid. Spatial averaging should not be confused with issues of 
positional accuracy of measurements: each calculated PIV vector is located at the centre 
of the respective cell, but in reality could be based upon particles from anywhere within 
that cell. An LDA measurement can be based upon particles passing anywhere through 
the control volume.

The effect of spatial filtering in PIV upon the calculated higher order statistics is 
dependant on the interrogation cell size relative to the turbulent scales present in the 
domain. In the PIV measurements documented herein, each finite interrogation cell is 
typically of order 1.0mm along each of its dimensions – a size not significantly smaller 
than the integral length scales present in these typical engineering flow problems. It 
should therefore be apparent why this effect must be acknowledged.
6.2 Experimental Illustrations

The effects of sub-grid filtering are particularly well visualised by considering the shear layer of the flow behind a bluff body, such as the flow over the step scenario introduced in chapter 4. Large scale turbulent motions are present in the wake region. These motions influence and shed into the free stream. Additionally in these flows much smaller scale structures occur within the shear layer itself.

![Fig. 6.2 Large FoV Time Average Example of a Typical Shear Layer Flow with a Variety of Example PIV Cell Sizes from Different FoVs Visualised](image)

![Fig. 6.3 Small FoV Time Average Example of a Typical Shear Layer Flow](image)
An example of the time average flow of such a scenario is given in figure 6.2, collected using a relatively large FoV (140mm). Due to the size of the data plane, the vectors shown are relatively sparse for clarity, but the example identifies the free stream in the region $y>10\text{mm}$ and shear layer occurring around $y=7\text{mm}$. In this case the typical integral length scale of the flow is 4.6mm (illustrated), and also superimposed upon the vector data are interrogation cell sizes for different FoVs, thus identifying the relative size of the measurement volume compared to the flow scales present. Figure 6.3 then shows time average data for the identical flow scenario collected at a small FoV (20mm). This clearly identifies that, even on a time average basis, a range of flow directions occurs within the interrogation cells of the larger FoVs.

![Fig. 6.4 Small FoV Instantaneous Example of a Typical Shear Layer Flow](image)

Figure 6.4 shows an instantaneous vector field collected at the small 20mm FoV and highlights the issue of sub-grid filtering; an eddy structure is present which the 140mm interrogation cell completely encompasses. Therefore such a cell could record any of the directions present as the peak (in the event that one particle exhibited a particularly strong ‘signal’), or a broad peak around zero displacement (the average displacement). It is therefore obvious that a measurement volume which is large compared to the scales of the flow structures cannot accurately record the relatively small displacements present in the flow. This fact has definite consequences on any flow statistics reported and this issue is discussed and quantified in the following sections.
6.3 Sub-Grid Filtering: Previous Work

It was identified in chapter 5 that the majority of the turbulent motions contributing to the flow statistics are in the small wavenumber, energy containing range of the spectrum. Pope (2000) states that the energy containing scales which are responsible for approximately 80% of the r.m.s velocities are in the range of motions greater than \((1/6)L\) in size. It is therefore a logical argument that in order to record the *majority* of the fluctuating velocities within the flow, the measurement volume should be of a similar geometry, or smaller. This notion is qualitatively supported by the instantaneous data in figure 6.4. Such a criterion is useful as a rule of thumb but is too loosely defined to assist in quantifying the effects of SGF.

The way in which the raw PIV image is discretised into interrogation cells for calculation is analogous to the discretised nature of a CFD domain. With increased computing power available, Large Eddy Simulation (LES) and Direct Numerical Simulation (DNS) have became feasible approaches for the modelling of complex unsteady flow scenarios. In DNS the Navier-Stokes equations are solved, therefore resolving all the scales of motion in the flow. The accepted criterion in DNS for good resolution of all turbulent scales in terms of cell size is \(\Delta X \leq 2.1\eta\) (Yeung and Pope, 1989) – a clearly expensive computation in terms of resources. A similar criteria was eluded to in chapter 3 regarding PIV cell sizes, but at the same time it was pointed out that such a small cell size was impracticable due to the Komlogorov scale often being of the same order as the particle diameter. LES bridges the gap between Reynolds-stress models, and the high resource demands of DNS by directly simulating the larger three-dimensional unsteady turbulent motions, and modelling the small scale sub-grid turbulence. For LES, the loose criteria described by Pope \((\Delta X \leq 1/6L)\) is more appropriate because it is accepted that scales smaller than the cell size will not be represented on an instantaneous basis. In the descriptions to follow the methods developed for PIV bear many similar features to the those of LES.

Despite the issue of SGF often being overlooked in experiments, it has been discussed by a number of authors in the PIV arena, with three distinct viewpoints:

a) Claim that all scales have been resolved via the use of sufficiently small interrogation cells.
Sub Grid Filtering

b) Attempt to recover the sub-grid scales by enhanced sub grid particle tracking algorithms.

c) Recognise the effect of SGF and attempt to estimate the loss of sub-grid fluctuating velocities.

Baldi et al (2002) carried out a PIV investigation in a vessel stirred by a hydrofoil impeller. They report an approximate definition for a global integral lengthscale of a tenth of the impeller diameter \((L_{11}=33.33/10=3.33\, \text{mm})\) and process their data with cell sizes equal to 2.46mm hence giving \(\Delta X/L_{11}=0.74\). It was claimed that “the data obtained with PIV measurements are very similar to those obtained with the LDA technique, and in most locations the differences are within the measurement errors. Therefore the PIV technique with the set up used proved reliable.” In fact their r.m.s. data shows underestimations of up to 30% in some locations, but overestimations in other areas of the measurement field. Some of the r.m.s. velocities were in agreement between the techniques, but the author feels that this is a trap fallen into by many experimentalists in the field, where the supposed correct answer is arrived at for the wrong reasons. In this example the fact is that there exist sub-grid fluctuations that could not be resolved: the similarity of the LDA and PIV r.m.s. velocities at the limited number of locations were probably coincidental. (A possible problem in impeller experiments is a lack of localised seeding due to centrifugal effects). Without sufficient seeding, the probability of valid detection is reduced and unless a comprehensive means of checking the quality of the data is incorporated, this noise contributes to the statistics and artificially increases the r.m.s., which then appears to match the alternate measurement technique. And one must also consider the accuracy of the alternate technique. LDA is susceptible to velocity gradient broadening effects which, unless taken into account, can significantly affect the statistics also. Baldi et al (2002) do, however, provide some lucid descriptions of their attempts to calculate dissipation rates using spatial derivatives of PIV vector data, arriving at findings in agreement with Piirto et al (2000) and Saarenrinne et al (2001) and those echoed in chapter 5 regarding spatial derivatives and the Taylor lengthscale: “The measured dissipation rate increased with the spatial resolution of the PIV measurements… lack of resolution at the dissipative scales was clearly the most likely reason for this discrepancy.”
Finzenhagen *et al* (1999) utilise PIV in their investigations to calculate small turbulent scales. They provide excellent detail of their methods to estimate the Taylor and Kolmogorov scales (which follow similar techniques to those outlined in chapter 5) and the assumptions of local isotropy which are necessary. They use spatial derivatives to calculate Taylor Length Scales of around 0.1mm, using cell sizes of order $\Delta X=1.0\text{mm}$. Although no integral scale is reported, it is often an order of magnitude greater than the Taylor Scale giving $\Delta X/L_1 \approx 1.0$. Their r.m.s. velocity data is lower than the equivalent LDA data, and the PIV spatial velocity correlation data exhibits higher values. These trends are typical of spatial filtering effects.

A significant number of PIV experiments documented do utilise sufficiently small cells relative to the integral lengthscales to record the vast majority of turbulent scales present. For example Lindken *et al* (2002) use a microscope PIV setup to enable cell sizes as small as $42\mu\text{m}$. In their investigation, the Kolmogorov scale was around $5\mu\text{m}$, and hence they would clearly record almost all of the turbulent motions. However, with such a challenging configuration they reported a maximum of only 80% ‘good’ vectors. It is suspected that it is for this reason that the PIV r.m.s. velocities are slightly higher than the equivalent DNS data (the relative error increasing the noise and hence r.m.s.). Although not strictly an experiment, Lecordier *et al* (2001) use DNS to produce synthetic PIV images with a known turbulence field to assess the treatment of PIV for turbulent flow studies. Their investigation is restricted to low Reynolds number and small virtual volume due to the computing power required for the DNS. With these conditions comes a large Kolmogorov length scale of $180\mu\text{m}$, which was approximately equal to the cell size. In such a case, it is clear that all turbulent motions are resolved. Unfortunately in real experiments, these kinds of ideal conditions seldom occur.

Where simple flow problems are studied experimentally with PIV and the cell size is quite small in relation the scales present, comparison of the flow statistics with alternate point measurement techniques often falls within the bounds of the experimental accuracy, and the issue of SGF is hence absorbed in these tolerances. In such experiments, the claims of PIV as a highly efficient ‘replacement’ to LDA or HWA may be justified, but these validation procedures cannot be extended to more complex flow scenarios where the relative size of the turbulent scales means SGF is a real issue which will significantly
Sub Grid Filtering

affect the higher order statistics. It is this type of flow scenario with complex recirculation zones, shear layers, and separated flow which is often most representative of real engineering problems, and is certainly the case in gas turbine combustors.

A large proportion of workers in the field have concentrated on recovering the sub grid fluctuations via so called ‘super-resolution’ PIV – a natural pursuance of improvement given PIV’s youth relative to the classical measurement techniques available. Super-resolution PIV essentially attempts to recover the sub-grid fluctuations by particle tracking within cells (Wernet, 1999; Wernet, 2001; Stitou and Riethmuller, 2001) or adaptive multipass gridding down to particle diameter sizes (Hart, 2000). The concept of super-resolution PIV was actually introduced by Keane et al (1995), and Stitou and Riethmuller (2001) base their efforts on this initial work using conventional cross-correlation PIV followed by sub grid particle tracking within individual cells. They report promising velocity data although state that no quantitative estimation of the quality of the measurement can be defined. They conclude their work stating that extensive validation is required to answer the question “what are we measuring?” Wernet (1999) takes the concept a step further by introducing fuzzy logic principles to determine the correct particle pairing on the sub-grid scale, although it was found that “in practice, the particle pair operation has a success rate of approximately 30%” which would have drastic consequences on the sub grid turbulence recorded in the event of false data being included in the statistics calculation. No r.m.s. data was presented in either of these sub-grid PTV documents.

The idea of sub-grid particle tracking is logical and the attempts have had some success but they encounter a common obstacle: PIV requires a certain number of particles (at least 5) within each interrogation cell to have a high enough confidence in the recorded correlation peak. Given a constant seeding density, the effect of reducing the cell size simply reduces the number of particles present in each cell, in turn reducing the strength of the correlation peak and hence the confidence in that peak and displacement accuracy. The additional risk is that, at these sub-grid scales, no particles exist to track. Therefore all of the methods result in higher errors on the data because of the effects of noise, and elevated probability of false particle pairing, with the consequence of inflated r.m.s.
values. One must also consider the re-mapping of inhomogenously distributed particle tracked vectors and their effect on the r.m.s.

The author feels that the only answer to the true recovery of sub-grid fluctuations lies with technology. Higher resolution CCD sensors will allow the often used 32x32 cell to represent a smaller area in the physical measurement plane – the cell hence becomes physically smaller and the sub-grid motions lost diminishes. However this would in turn necessitate the use of smaller particles to maintain the appropriate seeding density, and hence higher power pulsed lasers to illuminate them. It also requires higher capacity data transfer from the camera to the computer or dedicated image memory if any reasonable temporal resolution is to remain (an approach currently being pursued in high speed PIV systems). For the present, we are left with the following predicament: sub-grid filtering exists, so how do we cope with it? Fortunately a small number of authors have attempted to answer this question.

Sharp and Adrian (2001) calculate dissipation rates around a Rushton Turbine and use the Smagorinsky model, usually associated with sub-grid modelling in LES calculations, to estimate turbulent kinetic energy dissipation on the sub-grid scale. They find that in their experiment PIV captures at least 70% of the true dissipation. Saarenrinne et al (2001) use the Helland model spectrum to estimate the amount of TKE and dissipation rate recorded for a given cell size, finding that the spatial resolution should be around 20\eta in order to record 95% of the TKE. They point out that PIV is closely related to LES, and that PIV results are filtered velocity fields which can be expressed as follows:

\[ \tilde{U}(x, t) = \int G(r, x)U(x-r, t)dr \]

The filter function, G(r), is most appropriately described by the box filter function:

\[ G(r) = \frac{1}{\Delta} H\left(\frac{1}{2}\Delta - |r|\right) \]

where \( \Delta \) is the filter width, determined by the cell dimension, hence \( \Delta = \Delta \).

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The use of the above methods in PIV were published subsequent to work commencing under this project, which follows the method of Hoest-Madsen and Nielsen (1995) [to be referred to as ‘HMN’ throughout] who gave a theoretical examination of the problem of sub grid filtering and PIV, presenting a method by which the effect of filtering on the r.m.s. velocity could be related to the length scale and cell size for homogenous isotropic turbulence. Exactly isotropic turbulence does not exist in real flows (Hinze, 1959) but many flows have nearly isotropic local turbulence and many theories are based on the assumption of isotropy. The notion that a significant number of non-trivial flow problems have near isotropic turbulence on a local (i.e. sub-grid) scale is exactly what needs to be considered in the problem faced, and HMN state that the assumption of homogeneity and isotropy need only be true inside individual cells as the effect of filtering on individual vectors is that considered. They restrict their study to two-dimensional flows only, as many real flows are near two-dimensional. Defining the flow as homogeneous, isotropic, and two-dimensional results in a flow field which is statistically described by a stationary random field.

HMN quote formulae from Hinze (1959) which provide a statistical model of homogeneous, isotropic turbulence, on which they base their analysis. The analysis hinges on correlation functions relevant to the early and final stages of turbulence decay. These are given in terms of the integral length scale \( L_\parallel \) and Taylor micro scale \( \lambda_\parallel \) of the flow respectively:

\[
R_{\parallel} = e^{-\Delta x_i^1/L_\parallel} \quad R_{\parallel} = e^{-\Delta x_i^1/\lambda_\parallel}
\]

These equations are then used in formulae representing the volume weighting and random errors in PIV measurements to derive the two theoretical curves shown in figure 6.5 describing the relationship of true and measured r.m.s. The HMN technique is both logical and mathematically valid in its approach. Initial studies which compared PIV and LDA data suggested that the HMN curve based on the integral lengthscale provided a reliable means of correcting PIV statistics for the effect of sub-grid filtering. The initial studies are documented in Hollis et al (2001), and this chapter builds on those initial findings. It also provides a comprehensive validation of the HMN technique which Hoest-Madsen and Nielsen (1995) were unable to deliver in their documented work.
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Fig. 6.5 Theoretical Relationship of Measured and True r.m.s. Values (Hoest-Madsen and Nielsen, 1995)

6.4 Synthetic Data

To further understand the issue of sub-grid filtering and to make an independent comparison with the HMN theory for PIV data, calculations were carried out using a selection of synthetic velocity fields. Two separate synthetic data generation codes which produce random velocity fields with known correlation properties were used to create the sets of artificial data series.

6.4.1 1D Code

The first code developed by Kozintsev and Kedem (1999) allows the user to prescribe the form of the spatial velocity correlation across the velocity domain in terms of a one-dimensional function. This ‘1D code’ outputs a two-dimensional velocity field where each point is correlated with every other point in the domain (where points are separated by the vector, \( r \)), according to the correlation type chosen and its associated parameter values. Two correlation forms were considered: the rational quadratic model, and the exponential model. The related formulae are given below, followed by figure 6.6 which shows the correlation function forms. The two SVC types simulate different kinds of turbulent flows; the quadratic model emulates low Reynolds number homogeneous turbulence, and the exponential model is more like the correlation shown in figure 5.2 which is representative of higher Reynolds number turbulence.
Sub Grid Filtering

Rational Quadratic: \( R_{ij} = \left(1 + \frac{\Delta x_{ik}^2}{\theta_i^2}\right)^{-\theta_k} \)

Exponential: \( R_{ij}(x, \Delta x_k) = \theta_1 e^{\Delta x_k \theta_2} \)

Fig. 6.6 Comparison of Synthetic Correlation Models for 1D Code

Unlike the 2D code that was introduced in chapter 4, the integral lengthscale is not directly defined, and instead the coefficients (\(\theta_1\) and \(\theta_2\)) are prescribed in the correlation functions which in turn define the lengthscales. Two sets of synthetic data were generated for each correlation type, with integral lengthscales of 3.0 and 18.0 units.

6.4.2 2D Code

The second synthetic code used was that described in section 4.4. Only the inlet plane \((x_2, x_3)\) data from this ‘2D Code’ is necessary in this validation procedure. Two sets of data were generated, one with relatively small lengthscales (\(L_{22}=3.45\) and \(L_{33}=4.30\)) and one with large lengthscales (\(L_{22}=20.0\) and \(L_{33}=30.0\)).

The form of the correlation in this 2D code is almost identical to that of the 1D Quadratic model in figure 6.6. Due to the greater complexity involved with the 2D Code, it was not possible to easily modify this code to assume a correlation like the exponential form.
6.5 Validation: Synthetic Data

All synthetic data sets consist of series of 1024 planes of 62x62 velocity vectors. Each plane of velocity vectors can be described by $U_i,_{\text{real}}(x_1,x_2)$. The ‘real’ subscript is used to convey that each cell represents an exactly homogeneous fluid movement; it is the true displacement of simulated fluid.

The method to generate the SGF validation data was as follows:

1. Carry out spatial averaging on each synthetic data plane in the series by applying a local NxN matrix averaging algorithm, thus creating a filtered velocity field $\tilde{U}_i(x_1,x_2)$:

$$\tilde{U}_i(x_1,x_2) = \frac{1}{N^2} \sum_{j=1}^{N^2} U_{i,\text{real}}(x_1 + a, x_2 + b)$$

Where $a$ and $b$ are integers representing the range over which the spatial averaging is performed. I.e. for $N=3$ (the 3x3 spatial average); $-1 \leq b \leq 1$ and $-1 \leq a \leq 1$.

2. Calculate the ensemble average mean and r.m.s. velocity data across the domain for the entire spatially averaged data set series.

3. Calculate the integral length scales using the procedure outlined in chapter 5.

The NxN spatial filter (1) acts to create a local average over an NxN vector matrix for each instantaneous vector in the velocity field. Thus sub-grid filtering is simulated because each vector is then locally averaged in accordance with the matrix dimensions. This is a simplified version of the way in which the cross-correlation calculation of raw PIV images is performed because factors such as particle sparsity and non-uniform particle sizes are not taken into account. The analysis is therefore not specific to PIV but it does give valuable insight into the effect of spatial filtering as a process in its own right, without other noise being present in the data. It is also an effective controlled means of testing the HMN theory.

Time average quantities are calculated from ensemble averages across the entire domain and data series, i.e. utilising each cell of every instantaneous data plane – a valid approach given the synthetic flow is completely homogeneous. By doing so, high confidence is attained in the ensemble average quantities due to the effective large sample
size. The computational expense of calculating the integral length scales is much higher than the time average and r.m.s., and therefore only 64 integral lengthscales were calculated for each data set. This would give insight into the effect of sub-grid filtering on the integral lengthscale and hence SVC, an effect not yet reported by anyone in the field. The local spatial averaging used $1 < N < 17$, and hence a large range of cell size to length scale ratios were considered.

6.5.1 Synthetic Data: r.m.s.

The synthetic data are shown in figure 6.7, where it is compared to the HMN trends based upon the integral and Taylor lengthscales. The exponential synthetic data shows excellent agreement with the HMN integral curve. However the quadratic synthetic data only resembles the HMN Taylor curve in the early stages, tending towards the exponential one in the high relative cell size range.

![Fig. 6.7 Comparison of Synthetic Data with HMN curve – r.m.s.](image_url)

In order to fit trendlines to the synthetic measurements, it is necessary to split the data into $\Delta X/L < 1$ and $\Delta X/L \geq 1$ portions:

**Quadratic:**

$$u'_\text{meas}/u'_\text{true} |_{\Delta X/L < 1} = -0.0959 \left( \Delta X/L_{\text{true}} \right)^2 - 0.0186 (\Delta X/L_{\text{true}}) + 1.000$$

$$u'_\text{meas}/u'_\text{true} |_{\Delta X/L \geq 1} = -0.3315 \ln(\Delta X/L_{\text{true}}) + 0.8900$$

**Exponential:**

$$u'_\text{meas}/u'_\text{true} |_{\Delta X/L < 1} = e^{-0.3235(\Delta X/L_{\text{true}})}$$

$$u'_\text{meas}/u'_\text{true} |_{\Delta X/L \geq 1} = -0.2181 \ln(\Delta X/L_{\text{true}}) + 0.7501$$
It was stated in chapter 5 that the exponential correlation function most closely represents realistic engineering flows of high Reynolds number, and therefore one would expect the experimental data in the following section to closely resemble the exponential trend, and hence the HMN curve based on the integral lengthscale.

6.5.2 Synthetic Data: Lengthscales

Whilst the effect of Sub Grid filtering upon the r.m.s. values described in the previous section has been considered in the literature, the effect on the SVC and hence any integral lengthscales calculated, has to date been given no attention. It stands to reason that there will be some effect upon the SVC, and figure 6.8 illustrates this by comparing the original SVC with those calculated using N=5 and N=11 filtered velocities of the exponential synthetic data. Clearly, the effect of spatial filtering is to smooth the SVC and increase correlation values, and to delay the crossing of the x-axis. These effects will result in an increased calculated integral lengthscale. Figure 6.9 shows the corresponding trend for the measured integral lengthscales using the synthetic data, relative to the true lengthscale. Note that the inverse value is shown on the y axis compared to the r.m.s. plot – i.e. the true value divided by the measured value. This was done to show similarity in the r.m.s. and lengthscale trends.

![Figure 6.8 Comparison of SVC data with Increased Spatial Averaging](image_url)
Although there is greater scatter on the data, the trend falls within the expected error bands of the calculated integral lengthscales defined in chapter 5. Like the r.m.s. trends, it is split into two parts:

**Quadratic:**
\[
\frac{L_{\text{true}}}{L_{\text{meas}}} | \Delta X/L_{\text{true}} < 2.5 = -0.0189 (\Delta X/L_{\text{true}})^2 - 0.1274 (\Delta X/L_{\text{true}}) + 1.000 \\
\frac{L_{\text{true}}}{L_{\text{meas}}} | \Delta X/L_{\text{true}} \geq 2.5 = -0.3200 \ln(\Delta X/L_{\text{true}}) + 0.8600
\]

**Exponential:**
\[
\frac{L_{\text{true}}}{L_{\text{meas}}} | \Delta X/L_{\text{true}} < 0.65 = e^{0.5141(\Delta X/L_{\text{true}})} \\
\frac{L_{\text{true}}}{L_{\text{meas}}} | \Delta X/L_{\text{true}} \geq 0.65 = -0.2300 \ln(\Delta X/L_{\text{true}}) + 0.6230
\]

The data gives proof that the effect of spatial filtering not only has an impact on the r.m.s., but also has the effect of increasing the calculated lengthscale due to the impact on the calculated SVC. The fact that the calculated integral lengthscale is dependent on the relative cell size must be considered when applying a correction to the r.m.s. of the form showed earlier, because in order to apply such a correction the true lengthscale must be known. The trends identified here provide a means of correcting the lengthscale (albeit via an iterative method). It should be noted that, in the earlier r.m.s. trends, the true lengthscales were used to construct the plots because they were known *a priori*, due to the fact they were specified for the synthetic data sets.
6.6 Validation : Experimental Data

Having successfully obtained trends for the variation of r.m.s. and integral lengthscales from the synthetic data, this section describes similar efforts using real experimental measurements. Three sets of experimental data are used to construct the plots shown in this section:

1. Flow Over a Step; introduced in chapter 4.
2. Cylinder Wake.
3. Pipe Flow; uses data obtained from the water analogy rig calibration (chapter 2).

In all three cases, data was also collected using LDA to obtain the ‘true’ r.m.s. values which are needed to construct the trendlines. Where possible, the measurement volume of the LDA system was orientated to avoid velocity gradient broadening effects in the collected data. Where this was not feasible, the results were corrected using the method of Durst et al (1993). Note that L_true was obtained from a best fit of the PIV data shown in figure 6.14 and not from measured LDA timescales and Taylor’s hypothesis (which is inappropriate in the regions of interest). Table 6.1 shows a summary of the experimental parameters and the true values at the points of interest. The sections following describe each experimental setup.

<table>
<thead>
<tr>
<th>Point Magnification Range</th>
<th>ΔX/L Range</th>
<th>$u_1^*/U_{ref}$</th>
<th>$u_2^*/U_{ref}$</th>
<th>$L_{11}$ [mm]</th>
<th>$L_{22}$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow Over Step: $U_{ref}=0.340$ $Re=7581$ (based on channel height)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>0.114≤M≤0.622</td>
<td>0.0418≤ΔX/L≤0.8969</td>
<td>0.2705</td>
<td>0.2235</td>
<td>5.50</td>
</tr>
<tr>
<td>B</td>
<td>0.0921≤ΔX/L≤1.0045</td>
<td>0.4043</td>
<td>0.2705</td>
<td>2.50</td>
<td>2.50</td>
</tr>
<tr>
<td>C</td>
<td>0.0742≤ΔX/L≤0.9132</td>
<td>0.0620</td>
<td>0.0588</td>
<td>5.50</td>
<td>2.75</td>
</tr>
<tr>
<td>Cylinder Wake: $U_{ref}=0.552$ $Re=12308$ (based on cylinder diameter)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>0.065≤M≤0.430</td>
<td>0.1256≤ΔX/L≤1.1328</td>
<td>0.2861</td>
<td>0.2843</td>
<td>3.90</td>
</tr>
<tr>
<td>B</td>
<td>0.0652≤ΔX/L≤0.8033</td>
<td>0.3278</td>
<td>0.2898</td>
<td>5.50</td>
<td>10.0</td>
</tr>
<tr>
<td>Pipe Flow: $U_{ref}=0.503$ $Re=50468$ (based on pipe diameter)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>0.060≤M≤0.563</td>
<td>0.0817≤ΔX/L≤0.7658</td>
<td>0.0272</td>
<td>6.23</td>
<td></td>
</tr>
<tr>
<td>B</td>
<td>0.0863≤ΔX/L≤0.8090</td>
<td>0.0994</td>
<td>5.90</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 6.1 Sub Grid Filtering - Experimental Validation Setups
To achieve the variation in $\Delta X/L$, in addition to using four (five for the cylinder wake) different magnifications, the velocities were processed at 32x32 and 16x16 pixel final cell sizes. Although the 16x16 cell will contain lower particle density, the results are valid for the purpose of calculating trends. All processing of raw PIV images utilised adaptive gridding with 64x64 pixel initial cell size. Deformed grids were also used to eliminate velocity gradient effects, and second order correlations were employed to increase the effective signal to noise ratio and data quality (see chapter 3).

6.6.1 Flow Over a Step
The main part of the investigation of trends with real flows considered the flow over a wall mounted obstacle of square cross section, the initial findings of which were detailed in Hollis et al (2001). The experimental setup and three points of interest were identified in chapter 4. Many experiments detailing flow over steps are available in the literature where a variety of measurement techniques have been employed. Sousa et al (1998) investigated a similar problem using PIV – the flow around a surface mounted cube – in an attempt to prove the validity of obtaining meaningful turbulence data from PIV. However the flow was very low Reynolds number (Re=770) and direct comparisons were not made with alternate measurement techniques, hence it is difficult to determine the accuracy of their statistical data.

6.6.2 Cylinder Wake
This experimental data was acquired in the Water Analogy facility during a study of upstream obstacles in the annulus feed flow to the admission ports. Those blockages took the form of cylinders, in this case placed in line with the ports and therefore causing the wake to strongly influence the admission port feed flow. Test condition geometry and time average data are shown in figure 6.10, and an instantaneous example is shown in figure 6.11. Both illustrate the two locations of interest:

A. Along the symmetry line in the reverse flow of the wake
B. Within the wake shear layer
6.6.3 Pipe Flow

The third set of data was collected during the calibration work of the Water Analogy rig detailed in chapter 2, where the central core pipe is blank (no admission ports). The data were taken at a point on the centreline of the core pipe, and at \( r=40 \text{mm} \) which is close to the wall.

6.6.4 Experimental Data : Results

The calculated experimental r.m.s. values for the backward facing step (BFS), cylinder wake (CYL) and pipe flow (PIP) are shown in figure 6.12 compared to the true values at the relevant locations. Figure 6.13 compares the HMN and Synthetic trends with the
experimental data (grouped only by velocity component). As predicted, the trend is similar to the HMN curve based on the integral scale, which is in turn similar to the exponential synthetic data. The form of the experimental data trend is as follows:

\[
\frac{u'_\text{mean}}{u'_\text{true}} \mid \Delta X/L < 1 = 0.2381(\Delta X/L)^3 - 0.395(\Delta X/L)^2 - 0.1155(\Delta X/L) + 1.000
\]

\[
\frac{u'_\text{mean}}{u'_\text{true}} \mid \Delta X/L \geq 1 = -0.2181 \ln(\Delta X/L) + 0.7501
\]

(note that \(\Delta X/L \geq 1\) assumes a trend parallel to the exponential synthetic one due to a lack of data here)

There is some spread in the experimental data, but in all cases the deviation from the trendline is within the error bands associated with the number of samples (\(N=2400\)) acquired.

**Fig. 6.12** r.m.s. Trend for Experimental Data

**Fig. 6.13** Comparison of Experimental, HMN, and Synthetic r.m.s. data Trends
Figure 6.14 shows the experimental data lengthscale trend, and Figure 6.15 compares it with that of the synthetic data. Again there is a spread in the data, but this conforms with the expected error bands. Like the r.m.s. there are clear similarities with the synthetic exponential data. The trendlines are described by the following formulae:

\[
\begin{align*}
\text{Exp Data} : & \quad L_{\text{true}}/L_{\text{meas}} |_{\Delta X/L \leq 1} = e^{-0.4001(\Delta X/L)} \\
& \quad L_{\text{true}}/L_{\text{meas}} |_{\Delta X/L \geq 1} = -0.2300 \ln(\Delta X/L) + 0.6700
\end{align*}
\]
The experimental data shows clearly identifiable trends in the r.m.s. and lengthscale data, and provides the means to correct PIV data for the effects of sub grid filtering, giving estimates of these fluid dynamic quantities which are significantly better than the raw data. The error bands of the trends are typically around 6%, and are therefore reasonable for the type of data.

### 6.6.5 Shear Stress Trends

The effect of sub grid filtering on the shear stresses, $u_i u_j$ (where $i \neq j$), was not considered in detail. To do so would require detailed data for flows with known shear stresses, which was beyond the scope of the project. Additionally, the LDA setup that was used to measure the ‘true’ r.m.s. data is a one-dimensional system, and therefore unable to acquire shear stress information. However, data is plotted for the flow over the step in figure 6.16, where the true value has been replaced with the value from the smallest field of view (and the lengthscale replaced with the average of those referring to the two velocity components).

![Shear Stress Trends](image)

**Fig. 6.16 Shear Stress Trends for the Flow Over the Step**

The figure shows a trend similar to the HMN-type curves calculated earlier, albeit with more spread on the data points. Therefore, until a comprehensive study of sub grid...
filtering effects on shear stresses from PIV data can be conducted, the author recommends that they should be corrected in the same way as the r.m.s. data (although the correction coefficient will be squared to reflect the shear stress being the product of the two fluctuating components). However, corrected shear stresses should be treated with caution until proper trends are substantiated. The study of shear stress trends highlights an area of future investigation.

6.7 SGF Correction Methodology

In order to calculate an estimate of the true r.m.s., an estimate of the true lengthscale is required. The correction of the integral lengthscale apparently requires the true lengthscale in order to calculate a value of itself (the true value being on both axes of the curve). Therefore this calculation should be solved iteratively where the first iteration assumes $L_{\text{true}} = L_{\text{meas}}$ in order to obtain a position on the $\Delta X/L_{\text{true}}$ axis. This yields an $L_{\text{true}}/L_{\text{meas}}$ ratio which in turn gives a new estimation for $L_{\text{true}}$. Performing this iteration for 10 cycles usually results in a value converged to within 0.0001%, with the convergence more rapid for the smallest $\Delta X/L$ ratios. However, in the rare event of a ratio in excess of 1.5, there is potential for the iteration to diverge. It is therefore recommended that a lower correction limit of $L_{\text{true}}/L_{\text{meas}} = 0.3$ be applied. Similar recommendations apply to the correction of the r.m.s. values.

6.8 Closure

Synthetic data has been used to validate the HMN-type correction method for sub grid filtering effects on PIV data. This gave a curve for the ratio of measured to true r.m.s. velocities in relation to the size of the PIV interrogation cell relative to the local integral lengthscale. In addition to this, trends were identified for the ratio of the true and measured integral lengthscale, an effect not documented until now. A comprehensive set of experimental data for a variety of flow phenomena was then used to calibrate the correction method resulting in an empirical curve. Preliminary shear stress data suggest they too exhibit similar trends due to sub grid filtering. This is a potential area of future study.
Sub Grid Filtering

Vector Fields, \( U \) → Fluctuating velocity components \( U = u - \bar{U} \) → Spatial Autocorrelation

Mean Velocities \( \bar{U} \) → 'Measured' RMS \( u'_{\text{meas}} \) → TKE, \( k \)

Estimate \( \varepsilon \) → Define char length, \( L \)

Calculate \( R_e_L_c \) → Calculate \( L_e \) → Calculate \( \lambda_e \) → Calculate \( R_e_L_e \)

Calculate \( k_{\text{true}} \) → True TKE, \( k'_{\text{true}} \) → True RMS \( u'_{\text{true}} \) → Correction of r.m.s. due to sub grid filtering

True Integral Lengthscales, \( L_{11\text{true}} \) and \( L_{22\text{true}} \)

Correction of integral lengthscales due to sub grid filtering

Iterate

Calculate \( R_e_L_e \) → Re-calculate \( L_e \) → Calculate \( \lambda_e \) → Calculate \( R_e_L_e \)

Calculate \( \lambda_f \) → Calculate \( \eta \) → Calculate \( \tau_\eta \) → Calculate \( u_\eta \) → Calculate \( \varepsilon \)

Fig. 6.17 PIV Data Calculation (including r.m.s. and lengthscale correction)
Sub Grid Filtering

The empirical trends derived should be used to correct PIV data in turbulent flows to give more reliable estimates of calculated r.m.s. and integral lengthscales. The correction can be implemented in an automated fashion, and a modified version of the processes by which the lengthscales, r.m.s. and associated fluid dynamic data are calculated from PIV data (first seen in figure 5.7) is given in figure 6.17. This includes the necessary correction methodologies developed in this chapter.

The correction methodology is successfully applied to the turbulent flows of the gas turbine combustor geometries in the following chapters, where it is proven that more accurate data is obtained via the use of this method in practical engineering flows.
Chapter 7

EXPERIMENTAL RESULTS
WATER ANALOGY FACILITY
In this chapter the results from the generic combustor geometry of the water analogy facility are presented and discussed. The preceding chapters developed methodologies for the analysis of PIV data in turbulent flows and highlighted the wealth of fluid dynamic information that is calculable, such as conditional averages, correlation maps, and integral lengthscales. These methodologies enabled the efficient identification of the most important features in the water analogy facility. They have also allowed data to be presented in a form which maximizes the information content. Chapter 6 introduced a technique for the correction of sub grid filtering effects in PIV data, and this facilitated accurate estimates of the turbulence statistics to be provided from relatively large fields of view.

The datum set-up represents a primary jet condition for the plain port geometry, which was described in chapter 2. Additionally, flow conditions were varied to represent the dilution jet case as well as intermediate configurations. The test matrix for the data presented in this chapter is given in table 7.1, together with reference velocities and jet Reynolds numbers.

<table>
<thead>
<tr>
<th>Flow Conditions</th>
<th>R</th>
<th>B</th>
<th>$U_c$ (m/s)</th>
<th>Re$_j$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary [datum]</td>
<td>5.0</td>
<td>50%</td>
<td>0.198</td>
<td>22580</td>
</tr>
<tr>
<td>Primary (increased R)</td>
<td>7.0</td>
<td>50%</td>
<td>0.158</td>
<td>25220</td>
</tr>
<tr>
<td>Primary (increased R - varied bleed)</td>
<td>7.0</td>
<td>Varied</td>
<td>≤0.158</td>
<td>≤25220</td>
</tr>
<tr>
<td>Dilution</td>
<td>2.0</td>
<td>80%</td>
<td>0.366</td>
<td>19870</td>
</tr>
<tr>
<td>Dilution (varied bleed)</td>
<td>2.0</td>
<td>Varied</td>
<td>≤0.366</td>
<td>≤19870</td>
</tr>
</tbody>
</table>

Table 7.1 Water Analogy Facility Test Matrix

Chapter 2 discussed the choice of R and B to be representative of primary and dilution configurations. Although the values are more characteristic of older gas turbine combustion systems, the underlying flow topology and fluid dynamic characteristics are still transferable to modern systems. Using the flow conditions for the primary and dilution configurations detailed in table 7.1 was also necessary in order to allow direct comparisons with the LDA data collected by Spencer (1998). However in addition to these, data were also collected for a primary setup with increased jet to cross flow ratio (the reasons for which are detailed in section 7.3), and varied bleed flows, in order to identify trends.
All experimental tests were carried out in accordance with the best practice optimisation procedures outlined in chapter 3. The seeding is Polyamid particles, which are known to faithfully follow the flow, and an average of 8 particles were present in the interrogation cells resulting in very good signal to noise ratios ($Q>2.5$, see section 3.3.1). The camera was carefully positioned perpendicular to the laser light sheet, and focussed to make certain of a sharp image with negligible peak locking (Peak Lock $<0.06$, see section 3.3.1) in the resultant vector maps. The inter-frame timing was adjusted to ensure optimum in plane particle displacement (with a maximum shift of 8 pixels) for each FoV investigated whilst at the same time maintaining first choice vectors accounted for typically 95% of the velocity measurements. Vector processing was carried out as described in chapter 3: adaptive gridding with 64x64 pixel initial cell size and 32x32 pixel final cell size, deformed grid to reduce correlation peak broadening due to velocity gradients across cells, and second order correlation to enhance signal to noise ratio.

Four different FoVs were utilised for most of the data, ranging from 140mm (full diameter) to small FoVs concentrating on the port exit. The half diameter (80mm) FoV provided the best compromise allowing identification of important flow structures, whilst at the same time providing accurate statistical information via the use of the sub grid filtering correction procedure. Measurement planes in the radial (x-r) and pseudo circumferential planes (x-z) were obtained, but capture of data in the (r-0) was not possible due to the test facility layout. Each measurement typically comprised 4 sets of 600 instantaneous vector fields to ensure good accuracy in the statistics from the large sample size. With the matrix of flow configurations studied and different FoVs, a huge amount of quantitative data was available, and this created a significant challenge in terms of data management. This was handled using the strategy detailed in section 3.7.

This chapter proceeds with a detailed analysis of the datum configuration, which includes comparisons with the LDA data of Spencer (1998) and application of the sub-grid filtering correction. It continues by providing a selection of additional turbulence quantities for the datum configuration, using the methodologies detailed in chapter 5. The subsequent sections concentrate on the most important flow phenomena from all flow setups, highlighting specific attributes via the use of analysis and presentation techniques described in the earlier chapters. These features are introduced below with
reference to the location of the phenomena within the flow field (note areas of interest are highlighted on only one side of the flow field for simplicity).

**Annulus Modes**

An instability at bleed ratios of order 50% leads to fluctuating flow topology in the annulus. Conditional ‘mode locked’ ensemble averages are calculated to highlight the effects of each annulus mode.

**Through Port Vorticity (developing in the annulus)**

At lower bleed ratios in particular, the flow which separates from the outer annulus wall forms a vortical motion orientated parallel to the through port axis. The dependence of this ‘bath plug’ effect upon bleed ratio is studied.

**Jet Shear Layer Vortices**

The shearing action between the jet column and core cross flow generates vortices within the shear layer. These are identified using decomposed vector fields, correlation maps and Laser Induced Fluorescence (LIF) techniques.

**Core Near Wall Vortices**

The core recirculation zone varies in size instantaneously and sometimes undergoes asymmetric collapse. During this breakdown the reverse flow column can adhere to the core wall, causing embedded vortices within the boundary layer.
Bimodal Behaviour in Core Recirculation Zone

The recirculation zone grows and shrinks in size with some low frequency periodic features. The underlying fluid dynamics are studied with the aid of conditional averages and correlation maps.

Unlike the preceding ones, subsequent figures are presented at the end of this chapter, with only tables being embedded in the text. Where appropriate, data is normalised by the average core inlet velocity (at $x=-150\text{mm}$) for the relevant flow setup, $\overline{U}_c$, where $\overline{U}_c=0.198\text{m/s}$ for the datum configuration. In accordance with the argument put forward by Spencer (1998), geometric information is presented in dimensional form. Although the natural choice for normalisation may be the port diameter (20mm), the annulus height and core diameter also play important roles in determining the fluid dynamic characteristics. The data shown concentrates on the radial plane through the port centreline ($\theta=0^\circ$), with additional data included where necessary.

7.1 Time Average Data

The datum configuration is firstly used to demonstrate agreement of the PIV data with the LDA data collected by Spencer (1998), followed by an in depth look at the fluid dynamic statistics.

The flow conditions for the datum (R=5.0, B=50%) at inlet ($x=-150\text{mm}$) and outlet ($x=150\text{mm}$) are shown in figures 7.1 and 7.2. The first figure shows the core and annulus inlet velocity and turbulence profiles non-dimensionalised by their respective velocity maxima, i.e. at the core and annuli centrelines. This figure shows a symmetric core velocity profile, but a slight difference in the annuli profiles in terms of maximum velocity, with the left hand side exhibiting an axial velocity 3% lower than that of the right hand side. This slight asymmetry was not evident with the blank test section and is therefore attributed to a slight influence of the port presence on the upstream flow. The r.m.s. velocities have been corrected using the procedures outlined in chapter 6, and compare well with those measured by Spencer (1998), being between 3% and 4% in the core centreline region and around 5% at the annulus centreline.
Experimental Results: Water Analogy Facility

Figure 7.3 shows the time average velocity vectors in the radial plane. In this figure, the PIV velocity data has been interpolated onto the LDA measurement grid for a more suitable comparative view. Because the presence of the port caused an unavoidable deterioration of the image quality, it was never possible to obtain velocity data inside the port, or in close proximity to the port exit. Therefore figure 7.3 shows port 'exit' velocities at \( r = 44 \text{mm} \), the closest possible data available being obtained using the 32mm FoV. As expected, the topology of the flow is very similar. The streamwise annulus feed flow enters the port to form the majority of the jet column, with some reversed flow from out of plane feeding the very rear of the port. The jet forms in the core and bifurcates at the impingement point, with some fluid forming a column of axially reversed flow, and the remainder passing downstream. The reversed flow meets the incoming core flow at the head of the recirculating region, resulting in a toroidal recirculation zone upstream of the jets. Some fluid passes around the recirculation zone, accelerating between it and the core walls where it meets the leading edge of the jet column. It passes around the jet column into the wake region, subsequently passing downstream and out of the core.

The only detectable differences in the flow topology of the PIV and LDA data are at the rear edge of the port in the annulus, the magnitude of the reverse flow along the core centreline, and a slight difference in lee-side jet angle. Figure 7.4 and 7.5 show comparisons along \( r = 0 \text{mm} \) and \( x = -20 \text{mm} \) in profile form. The higher reverse flow magnitude of the LDA data along the centreline suggests it was taken at a slightly greater jet to cross flow ratio. The profile at \( x = -20 \) shows some slight differences in the annulus inlet profile, suggesting that the inlet in the LDA data is being influenced by the downstream port earlier than that from the PIV data. These small anomalies are most likely due to small differences in mass flow rates through the system, which can only be set with an accuracy of \( \pm 2\% \). The core flow condition was especially sensitive to \( R \), and very slight differences in overall mass flow rates combined with the delicately balanced jet impingement system has probably resulted in the differences observed. Therefore comparisons between PIV and LDA data can only be made to an accuracy equal to the accuracy of the rig set up (\( \pm 2\% \)).

Figures 7.6 and 7.7 give the raw (uncorrected) normal stress data, and the longitudinal integral lengthscales \( (L_{11} \) corresponding to the streamwise velocity, and \( 2L_{22} \)
corresponding to the radial velocity – see chapter 5) from the PIV data, where the lengthscale data has been corrected in accordance with the procedures in chapter 6. Large integral lengthscales are evident upstream of the impinging region of the core flow indicating large scale turbulent motions associated with the recirculating region. Much smaller values are seen in the jet shear layers and at the jet impingement indicating the dominance of small scale turbulent structures. The radial lengthscales are significantly larger than the streamwise ones where the topology is most influenced by the jet, due to the dominant radial velocity of the jet column. Spencer (1998) reports the integral lengthscale at the point of impingement as being 15mm, compared to the 1.74mm calculated using PIV. The former value was calculated using an integral timescale obtained via an autocorrelation of the time history, and Taylor’s hypothesis together with an estimated convection velocity (which was assumed to be equal to the r.m.s. value due to the impingement point having zero mean velocity). This 15mm lengthscale appears very high given the small scale structures which are responsible for the turbulence at the impingement point. Based on the assumption that Taylor’s hypothesis is applicable at the point of interest, Spencer (1998) states that “close to impingement where local velocity is close to zero, the integral time scale will increase dramatically”. However, given the high levels of turbulence present in this region, the rate of mixing will be increased and hence the residence time of pockets of fluid will be reduced resulting in decreased time and length scales. It is the author’s opinion that the above statement was incorrectly applied because Taylor’s hypothesis is only appropriate in well behaved low turbulence conditions. It is also possible that the much reduced LDA data rates in the impingement region resulted in the calculation of an incorrect timescale, upon which the calculated lengthscale depended.

Small integral lengthscales mean that the sub-grid compensation factors are higher. For accurate capture of the turbulence quantities with the interrogation cell sizes of 1.74mm employed at this FoV, the integral length scales should be in excess of 8.7mm in order to capture 95% of the turbulent motions. It is clear that this is not the case, and in regions such as the impingement point and shear layers a significant increase in the corrected normal stresses is seen in figures 7.8 and 7.9, where they are compared to the LDA data. (Note that correction of normal stresses uses the squared value of the r.m.s. correction factor.) The corrected PIV data shows better agreement with the LDA values, as would
be expected based on the evidence of chapter 6. Some differences are present between the corrected PIV and LDA data, but these are primarily in the jet shear layer region, where the LDA data was acquired using a relatively coarse measurement grid and hence the shear layers are poorly resolved. An r.m.s. velocity profile is provided in figure 7.10 at the r=12.0mm location to highlight the applicability of the sub grid correction, without clouding the performance of the methodology with grid dependency issues. Measured and corrected r.m.s. data are given for the largest (140mm) FoV, along with measured small FoV (25mm) data and LDA values. The profiles highlight the fact that the large FoV PIV data is improved by the correction method, with the corrected values more closely matching those of the small FoV and LDA data. The data shows that even in this flow containing high anisotropy and a variety of flow mechanisms, the sub-grid correction method works well. This gave further confidence in the procedures, and applicability of the method to all obtained data. The data in section 7.2 utilises corrected values.

With its higher grid resolution the PIV data in figures 7.8 and 7.9 highlight normal stresses along the jet column shear layers, as a result of their interaction with the core cross flow. The data in the port itself has to be treated with great caution due to the interpolated nature in this region. It is perhaps surprising to see the relatively low normal stresses within the jet core, indicating a lack of mixing of this flow until it reaches the impingement and bifurcation point at the core centreline, where the radial normal stresses are logically very high. It can also be seen that the streamwise normal stresses are somewhat elevated at the head of the recirculation zone, where the reversed flow meets the incoming core flow. The annular flow exhibits relatively low normal stress values upstream of the port where it is essentially an undisturbed channel flow. The values rise in closer proximity to the port, due to the separation of fluid from the outer annulus wall and impingement of flow at the rear edge of the port at the inner annulus wall.

Despite the support given for the successful application of the correction method, utilising smaller FoVs should always provide more accurate statistical data, to a higher level of confidence. However, in this type of project there is interest in both the large and small scale structures. In an idealised investigation both large and small FoV data would be collected for all flow scenarios. However, this is often unfeasible from a timescale
Experimental Results: Water Analogy Facility

and data management perspective. Therefore, as is often the case in research projects, some compromise has to be found. The half diameter (80mm) FoV provides that compromise, and is indeed utilised for a large proportion of the analyses carried out.

7.2 Additional Turbulence Quantities

Using the methods and formulae described in chapter 5, it is possible to derive flow quantities in addition to those shown in the previous section. The turbulent kinetic energy, $k$, is presented in figure 7.11 and calculated assuming that the out of plane r.m.s. velocity is similar to the radial r.m.s., and hence $k=1/2(u^2+2v^2)$. This is a standard approach for a flow with a dominant flow direction, and was identified by Spencer (1998) as providing the most accurate representation of $k$. Although this calculation of the turbulent kinetic energy is considered appropriate in the majority of the flow, the reader must recognise its limitations in localised regions such as impingement, where it may be the case that the radial r.m.s. velocity dominates the true value of $k$, due to it undergoing a rapid deceleration in that direction. As stated, the value of $k$ is calculated directly from the normal stress components, and therefore similar comments apply to those made earlier with reference to figures 7.8 and 7.9: high levels of $k$ are seen at the impingement point and across the shear layers in this data.

The dissipation rate of the turbulent kinetic energy is shown in figure 7.12, normalised by the square of the reference velocity, $\overline{U_c^2}$. It can be seen that the dissipation rate varies by 3 orders of magnitude, from less than 1/s in the low turbulence annulus feed flow, to over 1000/s at the jet impingement where the highly turbulent and rapid mixing occurs. Relatively high values also exist in the jet shear layers, but within the jet are an order of magnitude lower, indicating that the initial jet column is ‘protected’ from the cross flow by the shear layers which shroud it until close to the point of impingement. Whether these estimations of dissipation rate are accurate is difficult to ascertain due to a lack of data for comparison, which is partly due to the methods in chapter 5 providing one of the few means of estimating dissipation rate. What can be said is that the values are of the same order as those given by the well known broader estimate inherent in the initial calculations of chapter 5, where the dissipation is rate given by (see section 5.4);
\[ \varepsilon = A \left( \frac{u^3}{L} \right) \]

If the jet impingement is considered, figure 7.12 reports the maximum figure of \( \varepsilon/\text{U}_\text{ref}^2 = 2511/s \) at \((x,r) = (15,0)\). Using the above equation with local values of \( L = \text{L}_c = 4.35 \text{mm} \) and \( u = (\%k)^{1/4} = 0.663 \text{m/s} \), gives \( \varepsilon/\text{U}_\text{ref}^2 = 1711/s \). This suggests that the broad estimation is an underestimation of the dissipation rate, if the methodology in chapter 5 provides an accurate value.

<table>
<thead>
<tr>
<th>Position</th>
<th>( x \text{ [mm]} )</th>
<th>( r \text{ [mm]} )</th>
<th>( \eta \text{ [\mu m]} )</th>
<th>( \text{L}_c \text{ [mm]} )</th>
<th>( \text{L}_c/\eta )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annulus feed flow</td>
<td>-35.0</td>
<td>60.0</td>
<td>138.7</td>
<td>11.49</td>
<td>83</td>
</tr>
<tr>
<td>Core recirculation</td>
<td>-20.0</td>
<td>20.0</td>
<td>25.10</td>
<td>31.00</td>
<td>1235</td>
</tr>
<tr>
<td>Jet upper shear layer</td>
<td>-2.0</td>
<td>32.0</td>
<td>17.53</td>
<td>5.34</td>
<td>305</td>
</tr>
<tr>
<td>Jet core</td>
<td>3.0</td>
<td>36.0</td>
<td>45.32</td>
<td>7.91</td>
<td>175</td>
</tr>
<tr>
<td>Port rear edge (annulus)</td>
<td>10.0</td>
<td>52.0</td>
<td>19.50</td>
<td>3.48</td>
<td>179</td>
</tr>
<tr>
<td>Core wall (upstream of jet)</td>
<td>-30.0</td>
<td>40.0</td>
<td>47.69</td>
<td>27.82</td>
<td>583</td>
</tr>
<tr>
<td>Jet wake</td>
<td>20.0</td>
<td>34.0</td>
<td>14.69</td>
<td>3.43</td>
<td>233</td>
</tr>
</tbody>
</table>

Table 7.2 Comparison of Characteristic and Kolmogorov Lengthscales

The characteristic eddy sizes, \( \text{L}_c \), are shown in figure 7.13 but being closely related to the integral lengthscales the values naturally echo the same trends. The smallest turbulence producing eddies are at the impingement point and within the shear layers, and are approximately 5mm in diameter, whereas the upstream core region has motions of up to 40mm in size. In contrast, the Kolmogorov scales are presented in figure 7.14. Chapter 5 stated the assumptions made in calculating these (the smallest of flow scales), and one must take these isotropic assumptions into account when assessing the data in this section, especially in regions of strong anisotropy where the assumptions are less accurate. In reality it is practically impossible to directly measure Kolmogorov length scales in 'real' engineering flow scenarios using PIV, or any other experimental measurement technique. The data provided here is considered to be one of the few viable methods of obtaining legitimate and valid indications of these smallest dissipative scales. Within the majority of the core relatively small Kolmogorov length scales exist, with the smallest values being in the jet shear layer and at the impingement point. The fact that small Kolmogorov scales but large characteristic lengthscales (figure 7.13) occur in the recirculation region indicates that the range of turbulent scales is large. Selected characteristic eddy size and Kolmogorov scale data are given in table 7.2 to highlight the range of scales present. The table confirms that the core recirculation contains the largest
range, and indicates that the cascade of energy is a relatively lengthy process. The Kolmogorov time scales (figure 7.15) echo the pattern of their lengthscale counterpart.

### 7.3 Annulus Modes

At the condition of 50% bleed, the annulus flow towards the rear of the port was observed to switch between two distinctive ‘modes’ as shown in the instantaneous examples of figure 7.16. These modes exhibit in-plane flow structures which mimic bleed ratios both higher, and lower than the time average value of 50%, and will be referred to as ‘mode A’ and ‘mode B’ respectively. Of the data collected for the various jet to cross flow conditions, the highest (R=7) appeared to have the most even balance of occurrence of mode A and B topology. At this point it should be highlighted that the flow topology at the R=7 (B=50%) condition is very similar to the datum configuration, and still representative of a primary port flow condition; the main difference is a higher reverse flow magnitude along the core centreline and hence larger recirculation region.

Initially a series of vector fields at R=7 were manually categorised as containing annulus mode A, mode B, or some other non-definable mode ‘X’. From these instantaneous vector maps, pseudo phase locked, or ‘mode-locked’ ensemble average fields were calculated. This categorisation suggested mode A topology occurred with a frequency of 23%, mode B at 45%, and mode X at 32%. In order to quantify any consequences due to the occurrence of each mode, a less subjective method of categorising the topology had to be devised, rather than the time-consuming manual selection process. This was achieved by considering an area of axial velocity to the rear of the port; the loci of which is shown in figure 7.17. An algorithm was written to acquire the axial velocity data at every vector position in the region, assigning a value of +1 where positive axial velocity occurred, and -1 where negative velocity occurred. These values were summed and normalised by the total number of points, $P_{tot}$ (136 in this case) thus resulting in minimum and maximum normalised values, $M_{norm}$ of +1 and -1. A value close to +1 indicated a flow with higher bleed ratio like characteristics (mode A), and -1 indicated mode B. This automated selection process was applied to the data and very successfully categorised the annulus mode. However, it was found that the range $-0.2 < M_{norm} < 0.2$ could contain modes of either description and intermediate modes not falling into either category. Therefore the following classifications were defined:
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Mode A = \text{Mode}_{\text{norm}} \geq 0.2
Mode B = \text{Mode}_{\text{norm}} \leq -0.2
Mode X = -0.2 < \text{Mode}_{\text{norm}} < 0.2

The mode locked average fields using the algorithm are shown in figure 7.17, where A=18%, B=65%, X=17%. It is interesting to note the positions of the core recirculation centres for each mode: All are close to r=24mm, and in mode B the axial location is at x=-24mm, but for mode X it is at x=-14mm. Mode A, however, exhibits two recirculation centres at x=-24mm and at x=-14mm. It would therefore appear that the annulus mode has a direct effect on the core flow topology, thus supporting the notion of coupling between the annulus and core flow.

The primary reason for occurrence of the different modes is related to the annulus feed flow impingement at the port rear edge, which in turn appears to be dependent upon the upstream separation point at the outer annulus wall. In mode A (mimicking higher bleed ratios) all of the flow feeding the jet originates from upstream in the annulus. For mode B the fluid which feeds the lee side of the jet originates from the reversed flow region downstream of the port trailing edge in the annulus. The main visible effect of mode B on the core topology is a higher initial jet angle in the rear portion of the jet and a narrower jet column.

Although the flow structures mimic alternate bleed ratios, there are visibly recognisable similarities between the mode average fields and the whole set time average data at R=5.0 and R=7.0 (shown in figure 7.18); mode A is similar to R=5.0, and mode B like R=7.0. The data therefore could suggest that instantaneously, the jet to cross flow ratio is varying rather than the bleed ratio. However, alternate conclusions could be drawn by considering varied bleed ratios at fixed jet to cross flow values, as illustrated in figure 7.19 which shows bleed ratios of 55% and 45% at R=7.0.

Whether the annulus mode phenomena is related to an instantaneously varying jet to cross flow ratio or bleed ratio could be clarified by considering the jet characteristics at port exit. Calculations showed very little variation in absolute velocity across the jet column (-6mm < x < 10mm), and the jet mass flow rate based upon this in-plane velocity profile indicates less than 1% difference between modes. This fact, combined with core
and annulus mass flow rates which are known to be constant at inlet suggests that, although the annulus mode topologies are analogous to either varying jet to cross flow ratios or bleed ratios, the reality is that the effect is localised and does not significantly influence the jet or outlet mass flow rates. This implies that the jet cross sectional shape could be varying as a result of the behaviour. Attempts made to relate instantaneous changes in the jet column behaviour in terms of the streamwise velocity of the port exit (x-z) plane and radial (x-r) plane gave some support to this idea, but the links made could not provide a convincing explanation. To truly test this hypotheses, dual plane (e.g. Adrian, 2003) or holographic PIV (e.g. Hinsch, 1999) would be required to simultaneously obtain radial and pseudo-circumferential planes.

If the modal behaviour were driven by the core it must logically be as a result of the recirculation zone behaviour and subsequent effect upon the jet in cross flow. This hypothesis would suggest that at a low jet to cross flow ratio (and 50% bleed) where no recirculation zone exists, the modal behaviour would vanish. However, modal behaviour is still present at R=2 as shown in figure 7.20. This implies that the instability is a natural phenomena originating in the annulus at configurations with bleed ratios close to 50%. Although correlation calculations were unable to directly support this notion, those shown in figure 7.21 for the dilution setup (R=2, B=20%) do suggest some connection between the instantaneous behaviour in the annulus around the mode switching location, and the rear portion of the jet. The figure shows a series of temporal offsets (τ=−2ΔT, 0ΔT, +2ΔT) and identifies positive correlation in the axial velocity between the annulus and rear portion of the jet flows. This implies a downstream shift in the rear portion of the jet column is associated with a transition to mode A behaviour in the annulus, giving some support to the notion of a change in jet shape being associated with the mode transition in the annulus. Spectral analysis of the PIV data identified no dominant frequency, indicating that the triggering mechanism for mode transition is a random event. However this analysis is limited by the temporal resolution of the system, and can therefore only confirm a lack of very low frequency periodicity.

The data indicates the phenomena of the annulus mode is most likely a random event associated with the unstable impingement of the annulus flow at the port trailing edge. The switching of the mode is evident for all jet to cross flow ratios studied where the
bleed ratio is close to 50%, and annulus mode switching occurs regardless of the presence of a recirculating region in the core. Figure 7.22 shows the relative occurrence of each mode with bleed ratio. The changing of the annulus mode has no significant impact on mass flow rates through the system, and is instead thought to manifest itself as small changes in the jet shape.

7.4 Through Port Vortices

Spencer (1998) found that with annulus bleed flow set to near zero (<10%), through port vortical motion was evident on a time average basis, regardless of jet to cross flow ratio. Carrotte and Stevens (1990) also observed time average through port vortex motion in their studies of dilution ports at zero bleed settings. The same findings were observed in the current investigation and vector data illustrates the vortex motion for R=2.0 and R=5.0 at zero bleed ratio in figure 7.23. The previous works also found that neighbouring ports exhibited vortices rotating in alternating directions, and Spencer stated that this phenomena was due to the vortex interaction between ports setting up a stabilised circulatory motion. In agreement with the current investigation, Spencer found no evidence that through port vortex formation existed on a time average basis at higher bleed ratios. Without the ability to quantify instantaneous planar vector data, Spencer was only able to mention that through port vortex motion appeared to be evident on an instantaneous basis in the flow visualisations performed. It is thought that at the lowest bleed ratios, the port wake 'system' in the annulus stabilises to allow the vortex motion to be of a certain rotational sense, thus resulting in the motions identified on a time average basis. At higher bleed ratios, the flow past the ports in the annulus is less stable, hence no time average vortex motion is evident due to the sign of the vortex rotation changing on an instantaneous basis. The current investigation addresses the ability to identify instantaneous vortices, and examines in detail the occurrence of vortex motion primarily for the dilution configuration (R=2.0, B=20%). The topic of through port vortices is returned to later in this chapter and in the next chapter regarding the investigation in the fully featured combustor geometry.

Figure 7.24 shows pseudo circumferential (x-z plane) instantaneous data illustrating the occurrence of through port vortices in the annulus and close to the port exit plane. The examples in the figure all show anticlockwise vortices, although clockwise vortices and
other topology did occur, examples of which are shown in figure 7.25. It should be realised that the examples of the anticlockwise vortices at the several radial locations in figure 7.24 were obtained on different test runs due to the planar limitation imposed by the PIV system. Nevertheless, the occurrence of strong anticlockwise vortices at the different radii suggests that a single vortex could be strong enough to extend from its (assumed) origin at the outer annulus wall through the entire annulus and into the jet column. The vortices were physically largest and most frequently occurring in the planes furthest from the port ($r \geq 57.5\text{mm}$) which implies that they can be destroyed by the local turbulence intensity or shed downstream in the annulus before they can be convected through the port. It is interesting to note in figure 7.25 that the Counter-rotating Vortex Pair (CVP) can be of the conventional form, whereby the mainstream flow (either side of the port) is caused to reverse in the downstream region and results in reverse flow along the centreline of the port (looping ‘inwards’), or they can be of the opposite sense where the centreline flow is positive (looping ‘outwards’). It should be appreciated that the vectors shown are raw instantaneous data that has not been decomposed in an attempt to further highlight vortex properties. This is because the local convection velocity is near zero with the vortices passing perpendicularly out of the plane viewed. Therefore subtracting a convection velocity provides no further insight, although it may highlight other secondary vortex systems in the measurement plane.

If one considers the pseudo circumferential vector data shown along the port centreline, the motions observed could go part way to explaining the modal behaviour observed in the radial plane (section 7.3): despite reverse flow being evident off centreline at the largest two radii of figure 7.24, if we were only able to see the flow on the centreline (as is the case for the $x-r$ planes), no reverse flow would be evident. This might suggest that the modal phenomena observed in the radial plane were in fact purely a product of through port vortex motion. However, this cannot be the case because the modal behaviour was first discussed for the primary setup which shows scant evidence of through port vortex motion at its 50% bleed ratio.

The descriptions thus far provide essentially qualitative information regarding the instantaneous through port vortex motion. Vorticity (see section 3.5.2) is the traditional means of quantifying the amount of rotation of a volume of fluid. Due to the form of the
vorticity definition, identical distributions are obtained from both the time average vector field and the time average of the instantaneous vorticity values (obtained from the instantaneous vector plots) and hence either can simply be referred to as the 'average' vorticity, i.e.:

\[-\frac{\omega}{\omega} = \frac{\partial U}{\partial z} - \frac{\partial W}{\partial x} = \left( \sum_{i=1}^{N} \frac{\partial u}{\partial z} - \frac{\partial w}{\partial x} \right) / N\]

The average vorticity in the r=57.5mm plane is shown in figure 7.26 (where clockwise motion is positive). It suggests that the vortices are centred around x=13mm, somewhat downstream of the port, whereas the instantaneous vector data shows vortices clearly located above the rear half of the port itself. The reason that the location of the average vorticity is further downstream is because the vorticity quantity identifies shearing motion as well as vortex cores (Adrian et al, 2000). In chapter 3 the use of swirl strength ($\lambda_{ci}$) was presented as a means of identifying vortex cores. Unlike vorticity, swirl strength does not identify areas containing significant vorticity that are absent of swirling motion (i.e. shear layers). The ensemble average swirl strength, $<\lambda_{ci}>_N$ (where N=260) at the r=57.5mm plane is shown in figure 7.27 and the location of the highest values is more coincident with the actual location of the vortex centres seen in the instantaneous data. The swirl strength of the time average field (not shown) is different to and less meaningful than the ensemble average. And because swirl strength has only positive values, it gives no information about rotational sense unlike vorticity. Therefore as an alternative, the loci of maximum positive ($\omega_{\text{max}}$) and negative vorticity ($\omega_{\text{min}}$) was recorded for every instantaneous planar measurement. The sum of the occurrences of the maxima for each grid cell (of area $\Delta^2$) for the entire set of $N_{\text{tot}}$ instantaneous measurement planes was then calculated. This value was then normalised by the total number of samples multiplied by the grid cell area, and therefore the values express the expected relative number of occurrences per unit (1m²) area. The reason for including the grid cell area in the normalisation is to enable comparisons between data recorded for different FoV sizes, as a larger grid cell for the same experiment will theoretically contain more occurrences of the maxima (a comment that ignores sub grid filtering effects). Notice that we are considering the grid cell size, and not the interrogation cell size: these values...
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are related by the overlap used in the processing scheme (which is 50% in this case). The maximum vorticity values corresponding to the loci are shown in table 7.3.

<table>
<thead>
<tr>
<th></th>
<th>Vorticity, $\omega$ [1/s]</th>
<th>Swirl Strength, $\lambda_{st}$ [1/s²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average of Max Positive Values</td>
<td>114.5</td>
<td>3683.7</td>
</tr>
<tr>
<td>Standard Deviation of Max Positive Values</td>
<td>25.0</td>
<td>1747.4</td>
</tr>
<tr>
<td>Average of Max Negative Values</td>
<td>-116.7</td>
<td>N/A</td>
</tr>
<tr>
<td>Standard Deviation of Max Negative Values</td>
<td>26.1</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Table 7.3 – Vorticity and Swirl Strength; Average of Maxima at r=62.5mm

The loci of the vorticity maxima was seen to agree well with the ensemble average swirl strength in figure 7.27 and showed that although the vorticity quantity is influenced by regions of shear, the vorticity maxima tends to coincide with regions of truly vortical motion. However, it has been shown that in a number of instantaneous velocity maps there exists more than one vortex, and in some cases no vortex at all. Therefore, in the same way that the loci of the maximum vorticity was recorded, the loci of vortices above a certain strength threshold were also recorded. This allows the identification of more than one ‘strong’ vortex for each instantaneous plane where a CVP is present for example. It also accounts for instantaneous planes where zero ‘strong’ vortices exist. The threshold value had to be chosen to provide a direct and useful comparison between radial locations, and other flow configurations. The value chosen was $|\omega|=100.0$/s, and therefore the loci of $\omega>100.0$/s and $\omega<-100.0$/s are shown in figure 7.28 for r=62.5mm.

In the normalisation process, $N_{tot}$ is still the total number of instantaneous measurement planes. These new loci show slightly smoother distributions, but are similar in pattern to the $\omega_{\text{max}}$ loci data (not shown).

As the vortices pass closer to the port their vorticity and swirl strength increase, which disagrees with the computational simulations of Spencer (1998), who reported that the vorticity strength remain constant with decreasing radius. By the time the vortices reach the r=52.5mm plane, the number of vortices present exceeding the $|\omega|=100.0$/s threshold has greatly increased. Figure 7.29 shows that at r=52.5mm the loci of strong vortices is more concentrated (notice the scale is five times that of the previous figure), with the average of the maximum positive and negative vorticity in this plane being 187.4/s and 187.2/s respectively, and the average maximum swirl strength being 9616/s² – almost three times the value of that at r=62.5mm. Data for the same plane and bleed ratio at
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R=7.0 is not shown, but exhibits similar patterns and quantities to those for the dilution port set-up. In fact, there is little difference in average maximum values for different jet to cross flow ratios. However there is a distinct exponential trend with bleed ratio which is illustrated in figure 7.30. It shows that the occurrence of swirling motion in the x-z plane in the annulus is almost entirely dependent upon bleed ratio, and that the levels increase rapidly as the bleed ratio decreases. It is worth noting that at (or near to zero bleed) the vortex loci becomes skewed due to the stabilised wake at this zero bleed situation, which results in the vortices being seen in the time average flow fields as seen earlier, and by Spencer (1998) and Carrotte and Stevens (1990).

The previous works also describe coupling between adjacent ports leading to the vortices rotating in opposing senses. Figure 7.31 shows time average data for the dilution port case at different azimuthal locations. It identifies that flow is separated and recirculating at the outer annulus wall (in the x-r plane) at $\theta=+30^\circ$, but not at $\theta=-30^\circ$, suggesting that the recirculation extends between the port at $0^\circ$ and $+60^\circ$, but not between $0^\circ$ and $-60^\circ$. Figures 7.32 and 7.33 show instantaneous pseudo-circumferential (x-z plane) examples in the annulus for the dilution case, and primary jet case with zero bleed. These also show the flow separation at the outer annulus wall to be biased towards the positive azimuthal plane, again indicating that the recirculation at the outer wall is connected between ports. Figure 7.33 shows the particularly skewed nature of the annulus flow due to the asymmetric separation from the outer wall about the port centreline, which essentially results in the strong through port vortices, already discussed.

Compiling the data discussed in this section together with flow visualisations of Spencer (1998) allowed an interpretation of the annulus topology with varying bleed to be sketched in figure 7.34. At bleed ratios of greater than 55% (not illustrated), no flow separation from the outer annulus wall on a time average basis occurs (although it may momentarily separate between $55%<B<65\%$) because the amount of fluid demanded by the port is supplied entirely by the upstream flow. Between 30% and 55% bleed, in order to provide sufficient flow to the port, it must take part of its fluid from the reversed flow region which is established toward the rear of the port. The separation at the outer annulus wall is symmetric, resulting in a CVP at the rear of the port. For bleed ratios of 10% to 30% a significant amount of fluid must feed the port from downstream (having
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passed by the port in an offset azimuthal plane. The recirculation which ensues is relatively strong at the outer wall, causing successive pairs of ports to have connected recirculation regions. On an instantaneous basis, this linked recirculation can alternate sides between ports, especially for bleed ratios of 20% to 30%. For bleeds of less than 20% the situation stabilises in the paired arrangement shown in figure 7.34 (b) and (c). For bleed ratios of less than 10% the recirculation is very strong, resulting in one of the vortices of the CVP dominating the through port flow. The weaker of the two vortices is ‘leant’ over and sets up a similar recirculating motion to that at the outer wall, with the alternate adjacent port. This is believed to be the motion observed by Spencer in his flow visualisations, and occurs towards the inner wall. Unfortunately no PIV data was collected for zero bleed ratios between ports, hence the lack of quantitative data to substantiate this qualitative observation of Spencer.

7.5 Jet Shear Layer Vortices

Evidence of ‘ripples’ in the jet upper shear layer were present in the instantaneous velocity data prompting further investigation and analysis, including the use of decomposed vector fields to identify dominant structures. Figure 7.35 (a) shows a typical instantaneous vector field which, on close inspection, shows evidence of the rippling in the upper shear layer. The visualisation of the structures was greatly enhanced by LES decomposition, which subtracts a locally calculated average convection velocity from each vector. The DaVis software allows the user to specify over how many equivalent cells the local average is calculated (see section 3.5.1). Implementing a $\Delta^2 = 4$ filter identified the vortex structures within the shear layer in figure 7.35 (b). A short study of this phenomena utilising Laser Induced Fluorescence (LIF) was carried out in parallel to the PIV work to shed further light on the shear layer motions. In this investigation, Fluoroscene dye was added to the annulus flow and the raw images were captured using the PIV system to indicate scalar mixing. Despite the basic, uncalibrated nature of the set-up, figure 7.36 shows an example of the LIF image clearly identifying the rolling vortices of the jet shear layer. Lim et al (2001) gave lucid insight into jet in cross flow structures (described in chapter 1) and the illustration reproduced in figure 7.37 from that work shows significant agreement with the mechanisms present. Lim et al (2001) conducted their experiment for a single jet in a uniform cross flow, whereas the flow here
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encompasses large scale turbulent motions upstream of the jet which one could expect to
destroy the coherence in the jet shear layer. However, the structures clearly exist, and
such motions could have significant consequences in production gas turbine engines,
potentially resulting in incomplete mixing, increased emissions and fuel consumption.

In the case of the dilution port setup, the large scale turbulence associated with the
recirculation region is absent, and so the shear layer vortices become even more visible,
and survive further downstream allowing some quantitative wavelength (and frequency)
analysis to be performed. Figure 7.38 shows an example of a typical instantaneous
measurement plane. The contours and streamtraces represent the raw instantaneous
velocity, with LES decomposed vectors highlighting the vortices marked A to D. There
is regular spacing (wavelength) between the vectors which appear to originate from the
separation point at the leading edge of the port in the annulus. In this sense, it is not
actually the jet in cross flow set-up that initiates the vortices, but the flow around a sharp
obstacle. However, shear layer vortices have been identified in standard JICF
configurations so it would seem that the JICF flow downstream of the initial separation
assists in enhancing and sustaining a shear layer able to contain the vortices.

Although the temporal resolution of the PIV system prevents the possibility of directly
measuring the frequency of vortex generation for this setup, it is possible to use the
information available to calculate a frequency. If we consider the early vortex, B, on
which to base the calculation, the average wavelength (of A-B and B-C) is \( \lambda_o = 13.94 \text{mm} \)
and the convection velocity, \( U_o = 0.583 \text{m/s} \). The period of the vortex motion is therefore;

\[
T_o = \frac{\lambda_o}{U_o} = 0.0239 \text{s}
\]

This gives a frequency, \( f_o \) of 41.84Hz. We can then relate this frequency to a Strouhal
number, where the characteristic velocity is taken as the average jet velocity,
\( \bar{U}_j = 0.887 \text{m/s} \). The characteristic length is more difficult to define but will logically be
related to the shear layer thickness of around 4mm, which is 20% of the port diameter.
The Strouhal number of the shear layer vortices is thus:

\[
St_o = 0.2Df_o/\bar{U}_j = 0.189
\]
Performing the same analysis on the primary flow field shown in figure 7.35 gives $St_0 = 0.198$. An alternate approach to deriving a shear layer Strouhal number is to utilise the shear strain rate, $\partial |U|/\partial x$, the reciprocal of which may be used to replace $0.2D/\bar{U}_j$ in the above equation. For the dilution port setup, $\partial |U|/\partial x = 237.5s^{-1}$ giving $St_0 = 0.176$, which is very similar to the earlier value. However, this cannot be calculated without detailed knowledge of the flow across the shear layer, unlike the more versatile use of the port diameter to calculate an estimate of the shear layer thickness.

Blanchard et al (1999) made studies of jets issuing from a thin rectangular slit (20mm x 2mm) into cross flows at very low Reynolds numbers – $Re_j < 100$. (Although the journal paper makes reference only to the inlet Reynolds number, it is thought more pertinent to convert those to jet Reynolds number). The data of Blanchard shows that the frequency of the shed vortices increased linearly with $\bar{U}_j$ almost independent of jet to cross flow ratio, $R$, clearly indicating the greater importance of the jet velocity compared to that of the cross flow. The frequency data presented in the aforementioned document has been converted to a Strouhal number using the method described earlier (where $D = 2mm$, and $\bar{U}_j$ and $f_0$ vary) and is presented in figure 7.39. Also shown in this figure is dilution and primary jet data obtained in current work. Despite the significant differences in configurations (rectangular vs circular hole, perpendicular jet vs annular feed), a logarithmic trendline of the form defined below fits the two sets of data well. Nevertheless, there is a distinct lack of values in the mid Reynolds number range which would need to be obtained (for a variety of hole geometries) to confirm the trend:

$$St = 0.0324 \times \ln(Re_{jet}) - 0.1294$$

Figure 7.40 show correlation maps with the origin located within the upper shear layer of the jet, the patterns of which echo the periodic nature of the flow. It is pertinent to compare these correlation maps with those shown earlier where the origin was located in the annulus, and note the huge difference in correlated flow region. This is because the scale of the fluctuations in the shear layer is relatively small, which is confirmed by the PDF Conditioned (Reynolds decomposed) vectors superimposed on the $R_{21}$ plot. PDF conditioning was introduced in chapter 3, and chapter 4 illustrated its power as a correlation visualisation tool for the flow over a step. In this case the conditioning used
the radial velocity at the correlation origin as the subject, and an r.m.s. coefficient of \( K=1.5 \) (see section 3.5.1 for terminology). The PDF conditioned vectors serve to explain the flow topology of the correlation map, and illustrate the periodicity of the shear layer vortices, identifying a ‘chain’ of vortices along the upper shear layer of alternating sign. LIF and LES data only identified vortices rotating in the clockwise direction. It is believed that the anticlockwise vortices in the PDF conditioned vector map are secondary motions induced between the primary vortices.

### 7.6 Core Near Wall Vortices

By examining the instantaneous vectors of the primary setup, it was noticed that as the core recirculation zone grew and collapsed (discussed further in the following section), the asymmetry of the reverse flow column could sometimes cause the peak of it to ‘adhere’ to the core walls, and appear to shed part of itself into the boundary layer of the core wall flow. This vortical structure appears to become embedded in, and convected downstream with the boundary layer, existing for up to 8 time steps (approximately 0.5 seconds) before it reaches the port leading edge where it is destroyed by the local turbulence intensity of the shear layer vortices. The rotation of the vortex in the boundary layer is of the opposite sense to the large scale recirculatory rotation of the core (seen in the time average and instantaneous vector fields). It would therefore appear as though the hairpin vortices which naturally occur in boundary layer flow which are shed into the free stream under normal conditions (Piomelli, 1995) are enhanced by the large scale circulation of the core flow, and contained within the boundary layer because of this.

Figure 7.41 shows velocity vectors at the preceding two time steps \((t-2\Delta T\text{ and } t-\Delta T)\) to a vortex being shed into the boundary layer, followed by the velocity vectors illustrating the actual vortex motion within the boundary layer at \(t\), \(t+2\Delta T\) and \(t+4\Delta T\) in figure 7.42. These vortex motions embedded in the boundary layer are similar to those seen when a moving wake is shed across a turbine blade, as is the case for a rotor stage passing a stator stage in a gas turbine engine (Stieger et al, 2003), and can cause significant pressure oscillations. Luton et al (1995) showed that a minimum in surface pressure coincides with the location of the vortex centre. Such pressure perturbations could have significant impact on the performance of combustor film cooling flows, exposing the
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flame tube to excessive temperatures. Figure 7.41 showed the behaviour of the core flow preceding the occurrence of the boundary layer vortex, and clearly showed the outer jet to carry more (in plane) mass flow at the leading edge than the opposing jet. This mechanism appears to have caused an imbalance of the impinging jets, resulting in the asymmetric collapse of the core recirculation zone, and consequently the generation of the boundary layer vortex.

7.7 Core Recirculation Zone

This section gives an analysis of the flow behaviour in the core. Spencer (1998) observed bimodal behaviour in the axial velocity on the core centreline at x=-57mm. This bimodal behaviour was associated with the varying size of the recirculation region within the core, with flow visualisations identifying switching between two distinct modes of significantly different size. This behaviour was also observed using the PIV system, and typical instantaneous large and small recirculation zones are shown in figure 7.43. A PDF at x=-55mm (being slightly downstream compared with the LDA due to the slightly lower value of R) is provided in figure 7.44 quantifying the bimodal behaviour. Instantaneous data show that the recirculation region grows, and then collapses, giving rise to the different recirculation sizes. Spencer (1998) reported that spectral analysis identified no periodicity to the phenomena, but said that observing the flow visualisations indicated a possible low frequency motion. Using the PIV data to calculate an autocorrelation of the axial velocity $R_{11}(x,\tau)$ at $(x,r)=(-55,0)$ shows some evidence of this low frequency periodic behaviour, and is illustrated in figure 7.45 with background levels included and calculated according to section 4.4.4. The notion of a low frequency periodic behaviour is confirmed by a Power Spectrum Density (PSD) plot provided in figure 7.46 highlighting a broad peak around 0.60Hz. Bernero and Fiedler (2000) identified a periodic frequency of 0.40Hz in their study of a jet in counterflow (which is essentially what the reverse flow column represents). No other dominant frequencies could be found at R=5.0. At R=7.0, although no significant peak in the spectra was evident along the core centreline (possibly due to a lack of data far enough upstream), a relatively sharp peak was present at 1.65Hz at the recirculation centre as shown in figure 7.47, although the reason for this remains unknown.
Spatial Velocity Correlations were calculated at the point where the bimodal PDF occurs and these showed strongly correlated flow in the recirculation region (figures 7.49 and 7.50). Section 3.5 introduced the method of PDF conditioning to extract interesting flow phenomena, with chapter 4 demonstrating its ability as a correlation visualisation tool. The reader should refer back to these sections for the mathematical background. The velocity criteria chosen (where the streamwise velocity is the velocity subject) had to isolate the strongest motions present in order to explain the correlation maps. The r.m.s. coefficient which gave the best insight together with sufficient velocity maps for the ensembles was K=1.5, therefore essentially isolating the vector fields corresponding to the tails of the PDF in figure 7.44. Figure 7.48 shows the PDF conditioned velocity fields, and essentially represents pseudo phase locked averages of the example instantaneous vector fields in figure 7.43. The Reynolds decomposed versions of the PDF conditioned vector maps were obtained using the same criteria, and are shown superimposed on two of the possible four correlation maps (R_{11} and R_{12}) in figures 7.49 and 7.50. R_{11}, shows strong correlation of opposite sign with the flow at the core walls, suggesting a perturbation at the origin is associated with movement in the opposite sense at the core walls. The correlation of the streamwise velocity at the origin with the radial velocity everywhere else in the domain is shown in the R_{12} SVC. It exhibits a 'cloverleaf' pattern which implies rotational motion. The PDF conditioned vectors comprehensively describe the topology suggested by the correlation patterns. They show that a perturbation at the origin results in a very large scale rolling vortex between the reverse flow column and flow at the walls. In the case of a perturbation against the incoming core flow, the vector field mimics a starting jet type flow topology. Conversely, a perturbation representing a decrease in the reverse flow magnitude shows the circulatory motion occurring in the opposite sense. Clearly there is a battle between the incoming core flow and the reverse flow column produced by the impingement and bifurcation of the incoming jets, and when one momentarily overcomes the other, the reverse flow column either bursts upstream against the flow to produce a larger recirculation zone, or collapses under the momentum of the core inflow. The instantaneous (fluctuating component) structures associated with this are large scale vortices between the reverse flow column and flow close to the core walls.
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This phenomena was first identified by Spencer (1998) as a bimodal PDF. PIV has allowed a detailed quantitative description of the phenomena via the use of instantaneous vector fields, conditional averages, correlation maps, and PDF conditioned ensemble vector maps. The very large scale structures that exist will contribute to the r.m.s. and turbulent kinetic energy measurements. These large scale fluctuations contain substantial energy but are not 'traditional' turbulence, and so they will not contribute to the scalar mixing in the same way. This has potential implications on the efficiency of the combustion process with combustors, as well as affecting the downstream temperature distribution.

7.8 Closure

Through the use of best practice optimisation procedures and correction techniques developed in this thesis, high quality PIV data has been obtained in this generic combustor geometry. The methodologies developed for the analysis of PIV data has allowed the identification of instantaneous flow field structures not seen previously - phenomena which are invisible in the time average descriptions and have potential impact on combustor performance. Proof has been supplied that knowledge of the instantaneous flow is vital if we are to understand the mixing processes within the combustor, and therefore be able to design out flow behaviour which may increase harmful emissions.

Time average data for the datum case has been presented and compared with the LDA results of Spencer (1998). The mean velocities compare well, and the corrected r.m.s. velocities are also in close agreement, adding validity to the sub-grid filtering correction procedure outlined in the chapter 6. The main feature of the datum flow are the impinging jets which bifurcate to create a column of reverse flow which acts against the incoming core flow to result in the large toroidal recirculation zone. High levels of normal stress are evident in the jet shear layers and at the impingement point; the turbulence in the shear layers is particularly well resolved compared to the LDA data. Large integral lengthscales are evident in the recirculation zone due to the large scale motions, whereas much smaller scales are experienced in the jet shear layers, where shear layer vortices are responsible for the turbulent production mechanisms. Additional statistical quantities were provided in the form of Kolmogorov lengthscales and dissipation rates, providing insight into the flow that would be very difficult and time-consuming to measure directly.
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consuming to obtain with point based measurement techniques. The quantities calculated would provide useful information for the validation of CFD calculations.

Several instantaneous phenomena have been identified and quantified via the use of decomposed velocity fields, correlation maps, and conditional averaging. Modal behaviour is present in the annulus, regardless of jet to cross flow ratio at the 50% bleed condition. Although the annulus mode has negligible impact upon the mass flow rates through the system, it is instead thought to manifest itself in small changes of the jet shape and angle. This, together with correlation data, has supported the notion of annulus-core coupling effects.

Shear layer vortices have also been identified for both primary and dilution flow splits. These are especially prevalent in the leading edge shear layer. Decomposed vector fields and Laser Induced Fluorescence served to highlight the vortical motion. The fact that these vortices exist in the highly turbulent environment illustrates the strength of the motions, and could have important consequences on jet-core mixing processes within the combustor.

The core recirculation zone exhibits periodic behaviour for the datum configuration, with bimodal PDFs and conditional averaging identifying the mechanisms behind the process. The flow is analogous to a jet in counterflow, and the instabilities arise from the reverse flow column battling against the incoming core flow. This can result in the asymmetric collapse of the recirculation zone, which in turn can cause vortices to be shed into the core wall boundary layer flows. These near wall vortices become trapped, and can exist for significant time durations as they are convected downstream. This process could cause the breakdown of liner cooling flows in actual engine configurations.

Through port vortices were identified as being present, especially at lower bleed ratios. The establishment of the vortices with decreasing bleed ratio was interpreted and illustrated using radial and pseudo circumferential data. The loci of maximum vorticity magnitude were plotted, and this data was used to establish distinct trends in the occurrence of strong through port vortices with bleed ratio.
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Fig. 7.1 Inlet Conditions (x=150mm); Normalised Velocity Profiles
(Normalised by maximum velocities in respective flow)
R=5.0 Bleed=50%

Fig. 7.2 Inlet [x=150mm] (a) Outlet [x=150mm] (b) Conditions Normalised by $U_c$
R=5.0 Bleed=50%
Fig. 7.3 Time Average Velocity Vectors – PIV (top) and LDA (bottom)

$R=5.0$ Bleed=50%
Experimental Results: Water Analogy Facility

Fig. 7.4 Time Average Velocity Profiles along Core Centreline \((r=0)\)

\(R=5.0\) Bleed=50%

Fig. 7.5 Time Average Velocity Profiles in Annulus \((\mathbf{x}=-20\text{mm})\)

\(R=5.0\) Bleed=50%
Fig. 7.6 Uncorrected PIV Streamwise Normal Stresses (top) and Corrected Streamwise Longitudinal Integral Lengthscales (bottom)

瞒 = 5.0 Bleed = 50%

[masked port area in top figure indicates that data is interpolated in this region due to optical distortion, hence should be treated with caution. However, it is not removed from subsequent plots because most import interpolated data provides reasonable estimates]
Fig. 7.7 Uncorrected PIV Radial Normal Stresses (top) and Corrected Radial Longitudinal Integral Lengthscales (bottom)

R=5.0 Bleed=50%
Fig. 7.8 Streamwise Normal Stresses: Corrected PIV (top) and LDA (bottom)

R=5.0 Bleed=50%
Fig. 7.9 Radial Normal Stresses: Corrected PIV (top) and LDA (bottom)

R=5.0 Bleed=50%
**Fig. 7.10** Comparison of measured and corrected PIV data at 144mm FoV vs. Small FoV measured PIV [25mm] and LDA values at r=12.0mm plane (θ=0°)

R=5.0 Bleed=50%
Experimental Results: Water Analogy Facility

Fig. 7.11 Turbulent Kinetic Energy, $\frac{k}{U_c^2}$ [$k=\frac{1}{2}(u'^2+2v'^2)$]

R=5.0 Bleed=50%

Fig. 7.12 Normalised Turbulent Kinetic Energy Dissipation Rate, $\frac{\varepsilon}{U_c^2}$

R=5.0 Bleed=50%
**Fig. 7.13** Characteristic Eddy Sizes

R=5.0 Bleed=50%

**Fig. 7.14** Kolmogorov Length Scales

R=5.0 Bleed=50%
Fig. 7.15 Kolmogorov Time Scales

R=5.0 Bleed=50%
Fig. 7.16 Instantaneous Annulus ‘Mode’ Topology
Mode A (top) and Mode B (bottom) - Streamlines and regions of $U<0$ (shaded)

R=5.0 Bleed=50%
Fig. 7.17 Modes Classified via Automated Process

Mode A (top) [18%], Mode B (mid) [65%], Mode X (bottom) [17%]

R=7.0 Bleed=50%
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Fig. 7.18 Time Average Flow Topology – Streamlines and regions of U<0 (shaded)
R=5.0 Bleed=50% (left); R=7.0 Bleed=50% (right)

Fig. 7.19 Time Average Flow Topology – Streamlines and regions of U<0 (shaded)
R=7.0 Bleed=55% (left); R=7.0 Bleed=45% (right)

Fig. 7.20 Modes Classified via Automated Process
Mode A (left) [54%], Mode B (right) [29%]
R=2.0, Bleed=50%
Fig. 7.21 Cross Correlation Maps for Origin in Port Rear Feed Region
Streamwise Velocity Correlations: $\tau=-2\Delta T$ (top); $\tau=0\Delta T$ (middle); $\tau=+2\Delta T$ (bottom)
$R=2.0$ Bleed=20%
Experimental Results: Water Analogy Facility

Fig. 7.22 Bleed Ratio and Mode Trends

R=7.0
Experimental Results: Water Analogy Facility

Fig. 7.23 Time Average Pseudo-Circumferential Data at \( r = 52.5 \text{mm} \) (near port inlet)

Plain Port Dilution Setup, \( R = 2.0 \) Bleed=0% (top)

Plain Port Primary Setup, \( R = 5.0 \) Bleed=0% (bottom)
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Fig. 7.24 Instantaneous Pseudo-Circumferential Data - Through Port Vortices
R=2.0 Bleed=20%

Fig. 7.25 Instantaneous Pseudo-Circumferential Data - Alternate Topology
R=2.0 Bleed=20%
Fig. 7.26 Average Vorticity – r=62.5mm
R=2.0 Bleed=20%

Fig. 7.27 Ensemble Average Swirling Strength – r=62.5mm
R=2.0 Bleed=20%
Fig. 7.28 Loci of Vorticity; \( \omega > 100/s \) (top) and \( \omega < -100/s \) (bottom) (normalised quantities) \( r=62.5\text{mm} \)

\[ R=2.0 \text{ Bleed}=20\% \]
Fig. 7.29 Loci of Vorticity; $\omega > 100.0/s$ (top) and $\omega < -100.0/s$ (bottom) (normalised quantities) $r = 52.5\text{mm}$

$R = 2.0$ Bleed = 20%
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Fig. 7.30 Variation with Bleed Ratio at r=52.5mm
Average Maximum Vorticity per instantaneous measurement plane (top)
Average Maximum Swirl Strength per instantaneous measurement plane (bottom)

R=2.0 Bleed=20%
Fig. 7.31 Time Average Velocity Maps at Varying Azimuthal Locations

R=2.0, B=20%
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Fig. 7.32 Instantaneous Velocity Map centred at r=57.5mm
R=2.0, B=20%

Fig. 7.33 Instantaneous Velocity Map centred at r=52.5mm
R=7.0, B=0%
Fig. 7.34 Illustration of Annulus Topology with Varying Bleed Ratio
(a) 30%<B<55%; (b) 10%<B<30%; (c) B<10%
Fig. 7.35 Example of Instantaneous Jet Shear Layer Structures;
(a) raw instantaneous PIV Data and (b) LES decomposed data ($\bar{x}' = 4$)

R=5.0 Bleed=50%
Fig. 7.36 LIF image showing jet shear layer structures
R=5.0 Bleed=50%

Fig. 7.37 Illustration of JICF observations of Lim et al (2001)
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Fig. 7.38 Instantaneous Example of Shear Layer Vortices
Streamtraces show instantaneous velocity / Vectors show LES decomposed velocity (normalised quantities) \( r = 52.5 \text{mm} \)
\( R = 2.0 \) Bleed = 20%

Fig. 7.39 Strouhal Number vs Reynolds Number
Comparison with Low Reynolds Number Data of Blanchard (1999)
Fig. 7.40 Spatial Velocity Correlations in Jet Upper Shear Layer

$R_{21}$ includes PDF Conditioned (Reynolds decomposed) vectors superimposed (ca-)

$R=2.0$ Bleed=20%
Fig. 7.41 Instantaneous Velocity Vector Data Illustrating Asymmetry
Prior to Vortex Shedding into Boundary Layer; t-2ΔT and t-ΔT
R=5.0 Bleed=50%
Fig. 7.42 Core Wall Boundary Layer Vortex; t, t+2\Delta T, t+4\Delta T

R=5.0 Bleed=50%
Fig. 7.43 Instantaneous Velocity Vectors Illustrating Small and Large Recirculation zones within the core

R=5.0 Bleed=50%
Fig. 7.44 Bimodal PDF as a result of Alternating Core Recirculation Zone Size 
(red line shows bimodal Gaussian fit to the data) 
\((x, r)=(-55, 0) - \text{R}=5.0 \text{ Bleed}=50\%\)

Fig. 7.45 Autocorrelation Function at Point of Bimodal PDF 
\((x, r)=(-55, 0) - \text{R}=5.0 \text{ Bleed}=50\%\)
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Fig. 7.46  Spectra near Point of Bimodal PDF - \((x, r)=(-55, 0)\)
Ensemble Average of 4x600 samples - \(R=5.0\) Bleed=50%

Fig. 7.47  Spectra at Recirculation Region Centre - \((x, r)=(-22.6, 24.0)\)
Ensemble Average of 8x260 samples - \(R=7.0\) Bleed=50%
Fig. 7.48 Pseudo-Phase Locked Average Velocity Vectors

\((x, r)=(-55, 0) - R=5.0 \text{ Bleed}=50\%\)
Fig. 7.49 Spatial Velocity Correlations at Point of Bimodal PDF with PDF Conditioned (Reynolds decomposed) vectors superimposed (ca+)

\((x,r)=(-55,0) - R=5.0\) Bleed=50%
Fig. 7.50 Spatial Velocity Correlations at Point of Bimodal PDF with PDF Conditioned (Reynolds decomposed) vectors superimposed (ca-)

$$(x, r) = (-55, 0) - R=5.0 \text{ Bleed}=50\%$$
Chapter 8

EXPERIMENTAL RESULTS
VULCAN SECTOR RIG
This section discusses results from the VULCAN sector rig. The geometry of the rig (as described in chapter 2) and fact that air is the fluid medium, made collection of data in the rig a challenging final test for PIV measurements in an actual gas turbine combustor. Where possible, repetition of data available in Griffiths (2000) is avoided, but some of that data must be included here to give a complete overview of the flow characteristics, and comparison of the measurement techniques. Methodologies developed in earlier chapters for the analysis of PIV data are used throughout, and flow phenomena compared to those observed in the water analogy facility. All PIV data was collected using the best practice procedures outlined in chapter 3.

The chapter begins with details of the measurement planes obtained using the PIV system, followed by comparisons with LDA data. Subsequent to this, comparison of the data from different FoVs is made, with attention given to the ability of the sub-grid filtering correction methodology in this challenging turbulent environment. The chapter then proceeds to present time average and statistical data from the three local Cartesian orientations, concentrating upon measurements previously unobtainable with point based techniques (e.g. fuel injector swirler exit characteristics). The sections subsequent to this present data regarding the most prevalent instantaneous flow phenomena. The four main areas covered are through port vorticity, jet shear layer vortices, primary jet impingement, and fuel injector exit flow precession. Analysis of these phenomena utilises methodologies developed in earlier chapters, and subsequently demonstrated on the water analogy facility data.

8.1 Measurement Planes and Basic Flow Topology
Data was collected in three local Cartesian measurement orientations described in chapter 2: radial (x-r), pseudo-circumferential (x-z), and axial (r-z). Note that in this description of the orientation planes, the r axis extends radially outwards from the imaginary centre of the engine. However, the radial datum (r=0mm) is taken as the centerline of the combustor for convenience (see figure 2.12). Also the x-z plane is always perpendicular to the local radial axis, and hence z location is local to the relevant radial axis (see figures 2.21 and 8.1).
A selection of the measurement plane orientations are illustrated in figure 8.1 where the two x-r planes azimuthally offset relative to the global Cartesian x-y plane are through the secondary, and the sector edge primary port centre lines. Although not shown, data collected through the secondary and primary sector edge for the positive offsets were collected. These showed that the flow behaviour is symmetric about the datum x-r plane, therefore allowing collection of data to be concentrated on one half of the sector. The half chosen was for the negative azimuthal offset because the rig positioning meant that it afforded more convenient optical access.

Most data was collected in the x-r planes as these provide the most interesting information, and allowed the best optical access. Although figure 8.1 gives a taste of the time average flow topology, it is of limited use for quantitative analysis due to the proximity of the different measurement planes and lack of geometry acting as physical reference points. Therefore the data presented in the following sections are restricted to more practical 2D contour and 1D line plots. All data is normalised by the reference velocity (pre-diffuser centerline value) of $U_{ref}=41.29\text{m/s}$.

### 8.2 Comparison with LDA Data

Figures 8.2 and 8.4 compare the basic topology at the primary ports to that measured using LDA (Griffiths, 2000). The first figure shows the sector centre data ($\theta=-0.5^\circ$) and the latter figure the sector edge ($\theta=-7.9^\circ$). Figure 8.3 shows the secondary port plane ($\theta=-4.35^\circ$) from the PIV data alone. The flow topology recorded by the two techniques are very similar, with only very slight differences detectable. The figures highlight the large difference in sector centre flow characteristics downstream of the injector compared to the sector edge data. At the sector centre the primary jets fail to impinge on a time average basis as would be expected, with the misalignment resulting in the outer primary jet dominating the recirculation zone. In contrast, at the sector edge the system appears to be more balanced with the jets impinging as anticipated. The secondary port flows, also effectively in the absence of upstream fuel injector influences, impinge in a predictable manner. Griffiths (2000) performed calculations on the time average data that supported the idea that the primary jet impingement is influenced by the pressure field generated by the flow exiting the fuel injector swirler. The impingement of opposing jets is known to be a delicately balanced system and Griffiths argues that slightly different feed flows to
the inner and outer primary ports cause the low pressure within the swirl of the injector exit flow to draw the outer primary jet upstream. This causes the misalignment of the jets seen on a time average basis and is likely to have important consequences on the mixing characteristics of the flow.

Figure 8.5 compares the time average mean velocities near the inner primary port (sector centre) jet exit. Although the data is at slightly different radial locations, the agreement in the velocity profiles is generally very good. The PIV profiles are closer to the combustor centerline because reflections from the sector edge primary ports in the background of the x-r plane measurements rendered data very close to the port exit unusable.

8.3 Sub-Grid Filtering

The nature of the rig set up, seeding and optical access requirements made PIV data acquisition a challenging proposition during this phase of the project. Therefore the time available to obtain data for analysis of sub-grid filtering effects was limited. Having proved the validity of the correction technique in the previous chapters, only a brief consideration is given here. Figure 8.6 shows the streamwise r.m.s. values near the inner primary port exit (r=-47.5mm) compared to LDA data. The 30mm FoV data shows uncorrected values, as the correction factors for this data are typically greater than 0.95. Data from a large FoV [125mm] shows that the corrected data closely matches the LDA and small FoV data, and gives further confidence in this technique for application to real engineering flows. The slight differences seen between measurement techniques are believed to be due to the small differences in radial location. As the corrected data can be seen to give a reasonably accurate estimate of the statistical characteristics, it was deemed acceptable to use the measurements from the large FoV to look at variations in flow characteristics across the entire combustor height (section 8.4.1).

8.4 Time Average and Statistical Data

This section begins with a comparison of the statistical characteristics of the flow from the full height x-r plane at the 3 different azimuthal locations. Following this are sections
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presenting detailed fuel injector swirler and port exit plane time average statistics from the smallest FoV data.

8.4.1 Full Height x-r Plane Data
The reader should realise that these largest FoV data are at the limits of those possible with the current system, and providing adequate seeding was extremely challenging due to the velocities and upstream geometry. Therefore, close to the combustor walls, the data should be treated with caution (where reflections were strong). Detailed statistical information for selected measurement planes are provided in the following two sections.

Figure 8.7 shows comparison of the primary jet turbulent kinetic energy at the sector centre and sector edge. Unlike the water analogy facility where the streamwise velocity dominates the majority of the flow, the values of $k$ were calculated from the two available normal stresses in the x-r plane assuming $k = \frac{1}{2}(u'^2 + v'^2)$. The misalignment of the primary jets causes the peak values to be closer to the inner flame tube wall. The higher TKE values here are likely to be due to larger scale interaction of the jets shearing with, rather than impinging against, each other. Lower values of TKE are seen at the head of the flame tube at the sector centre due to this lack of impingement and hence lack of conventional recirculation zone. The dissipation rate plots of figure 8.8 show higher levels within the jet flows at the sector edge, compared to the sector centre. They also exhibit higher levels towards the head of the flame tube due to the recirculation created by the more balanced impinging jets.

The streamwise integral lengthscales (figure 8.9) are typically between 4mm and 8mm for both sector centre, and sector edge: the latter having slightly higher values. In contrast, the radial lengthscales (figure 8.10) in the primary zone of the sector centre are large in comparison to the sector edge, due to the jet misalignment and resultant lack of small scale turbulence production.

The secondary port flow turbulent statistics are shown in figure 8.11, and are similar in magnitude and distribution to the sector edge primary port flows. The integral lengthscales (figure 8.12) are generally slightly higher at the secondary port plane compared to those at the primary planes.
8.4.2 Fuel Injector Swirler Exit Plane

Figure 8.13 shows time average velocities at an r-z plane close to the fuel injector exit (x=6mm), and is the closest possible axial measurement plane because of the presence of the shroud. Cartesian velocities are not necessarily the most intuitive to present for the injector dominated flow, and for that reason the local polar velocity contours \((V_r\) and \(V_\theta\)), relative to the fuel injector centerline, are presented in figure 8.14. It is important to note in both figures that an area of poor quality vectors is evident around the shroud on the left hand side. This is due to high intensity reflections in this region which significantly reduced the signal to noise ratio of the particle images. However the effect is localized and does not affect the validity of the majority of the data. The polar velocities show the radial distribution to be relatively axi-symmetric, although they do show that the swirl centre has been translated from the injector centerline by \((r,z)=(0,+2)\) – a significant shift only 6mm downstream of the exit plane. The tangential velocities \((V_\theta)\) are less symmetric, with the higher values biased towards the inner wall. This is due to the dominating outer primary jet fluid traversing across towards the inner flame tube wall before reversing direction across the face of the injector exit. Figure 8.15 gives a comparison of the raw time average velocity field and a radially averaged symmetric profile. Although the exit flow is not axi-symmetric, radially averaged data is presented for subsequent quantities because it provides a more intuitive view of the flow characteristics (using polar quantities). The profile shows the bulk of the flow exiting the swirler occurs between \(0.4<r/R_{inj}<0.8\). However, the combustor flow field is clearly having a dramatic effect on the exiting flow, as the streamwise direction is entirely negative (or near zero). This indicates that the flow exiting the injector is immediately forced towards the flame tube walls along the radial axis, and that the fuel injector pilot zone is long in the radial direction, but very short in the streamwise direction. It should be appreciated that no fuel injection or combustion is taking place in these measurements, and the flow local to the fuel injector is likely to be that most affected by the introduction of fuel flow and combustion.

Figure 8.16 shows the time average velocity distribution at \(x=11.5\)mm. At this short distance downstream of the injector (of 50mm diameter) there is scant evidence of swirling flow from the fuel injector in the upper half of the combustor can; the flow is dominated by the outer primary jet recirculation in this region. At \(x=11.5\)mm the polar
velocities have lost almost all axi-symmetric features, and the 'swirl' centerline has been translated \((r,z)= (+4.3, +6.7)\), giving an average angle between the injector centerline axis, and swirl 'axis' of approximately 35°. Figure 8.17 shows the circumferential velocity contours and vectors close to the outer primary port, with the left hand plot confirming the complete dominance of the outer primary jet at this plane. The vector directions suggest that the jet fluid probably bifurcates in the \(z\) direction at somewhere around \(r= -30\)mm, i.e. close to the inner primary jet exit. The top right hand figure moots a possible residual swirl effect from the injector flow upstream of the jet column. The trailing edge also exhibits a significant circumferential velocity component not evident in the jet column.

Figure 8.18 and 8.19 show the radially averaged polar normal stresses and turbulent kinetic energy distributions. The data shows that the TKE value peaks off the centerline at around \(|r|=10\)mm, and that the turbulence is anisotropic at this location with the radial normal stress component contributing twice that of the other two components. It is possible that this distribution is due to a precessing vortex core in this swirling flow, which would have important implications on what is conventional turbulence and what is due to large scale unsteadiness. The average of the three longitudinal Cartesian integral lengthscales is also provided in figure 8.19 and indicates the typical lengthscale to be 5% of the fuel injector diameter. The average Cartesian lengthscale is quoted because at the time of writing it was not possible to calculate a polar lengthscale, and the Cartesian values in isolation are asymmetric and not considered a fair representation of the size of turbulent eddy structures present. Figures 8.20 and 8.21 show the injector exit shear stresses, which again highlights a peak value around \(|r|=10\)mm.

8.4.3 Port Exit Data

Figures 8.22 to 8.25 concentrate on profiles close to the port exits where port geometries are included for clarity. Abbreviations in the legend are used for compactness and are described in table 8.1. The time average port exit velocities are shown in the first of the four figures. The streamwise velocity upstream of the ports is higher for the outer primary ports than the inner one. This is due to a positive streamwise component in the larger recirculation zone upstream of the outer ports. However, the inner and outer ports at the sector centre experience similar peak streamwise velocities, whereas the outer
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primary port at the sector edge has much lower peak velocities, similar to that of the outer secondary port. The secondary port flows experience lower streamwise velocities through the jet column than the sector centre primary port flows, with the inner port exhibiting reversed streamwise velocities towards the trailing edge.

<table>
<thead>
<tr>
<th>Key</th>
<th>Location</th>
<th>r [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>IPC</td>
<td>Inner Primary sector Centre</td>
<td>-47.5</td>
</tr>
<tr>
<td>OPC</td>
<td>Outer Primary sector Centre</td>
<td>50.0</td>
</tr>
<tr>
<td>OPE</td>
<td>Outer Primary sector Edge</td>
<td>44.0</td>
</tr>
<tr>
<td>ISC</td>
<td>Inner Secondary sector Centre</td>
<td>-48.0</td>
</tr>
<tr>
<td>OSC</td>
<td>Outer Secondary sector Centre</td>
<td>41.0</td>
</tr>
</tbody>
</table>

The secondary ports are not strictly sector centre, but for the purposes of the description, the notation indicates that they are downstream of the injector and azimuthally inboard of the sector edge primary ports

Table 8.1 Profile Locations close to the Port Exits

Both streamwise and radial velocities show that the inner primary port jet column is narrower (in the x direction) than the outer columns. This is due to a greater approach velocity in the inner annulus as a result of smaller cross sectional area and similar mass flow rate, discussed in detail by Griffiths (2000). This greater approach velocity means that there is a larger separation off the inner chute upstream edge. This difference in jet feed characteristics allows the injector swirl field to predominantly influence the outer port flow, with its lower initial streamwise velocity. The secondary port radial velocity distributions exhibit 'bimodal' peak values, possibly due to cooling slot flow entrainment at the trailing edge.

The inner ports show little sign of significant circumferential velocity magnitude, whereas the outer ports at the sector centre show more variation, with the primary flow exhibiting very high W component velocity magnitudes up to 20% of the reference velocity. The distribution indicates strong time average through port vorticity. The outer secondary port also shows a similar pattern, although with smaller magnitudes. Table 8.2 shows the port diameters and annulus heights at the port locations, height to diameter ratios, and estimated local bleed ratios. Spencer (1998) states that through port vortices are most likely at low bleed ratios and low height to diameter ratios. This has been the experience of other workers and was discussed in the previous chapter, where it was also confirmed that through port vorticity on a time average basis was only identifiable for
very low bleed ratios. However, in this data, the outer primary port exhibits the greatest through port vorticity, yet has a very high bleed ratio and relatively large (compared to a value of 1.0 for the water analogy measurements) height to diameter ratio. This suggests that another factor is responsible for the through port vorticity, a notion confirmed by the fact that the sector edge outer primary port, with similar geometrical and mass flow features to the sector centre, experiences no time average through port swirling motion. The outer secondary port also experiences through port vorticity, and does have a low h/D ratio, and lower bleed ratio. However, the strength of the vorticity is lower than the primary port and, at similar conditions in the water flow rig, time average through port vortices were not evident. This points towards a physical feature upstream of the sector centre ports in the outer annulus causing the vorticity. It is therefore likely that the fuel injector burner arm is shedding vortices in the outer annulus passage which are being fed through the outer primary and secondary ports. This could result in dominant shedding frequencies being passed into the core flow as a result. Any dominant frequencies in the flame tube are undesirable as they can cause flame instability.

<table>
<thead>
<tr>
<th>Port</th>
<th>Port Diameter, D (mm)</th>
<th>Annulus Height, h (mm)</th>
<th>h/D</th>
<th>Bleed Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner Primary (Sector Centre)</td>
<td>13.1</td>
<td>27.9</td>
<td>2.13</td>
<td>0.85</td>
</tr>
<tr>
<td>Outer Primary (Sector Centre)</td>
<td>14.5</td>
<td>23.6</td>
<td>1.63</td>
<td>0.82</td>
</tr>
<tr>
<td>Inner Secondary</td>
<td>16.8</td>
<td>22.0</td>
<td>1.31</td>
<td>0.69</td>
</tr>
<tr>
<td>Outer Secondary</td>
<td>20.1</td>
<td>19.4</td>
<td>0.97</td>
<td>0.52</td>
</tr>
<tr>
<td>Outer Primary (Sector Edge)</td>
<td>14.5</td>
<td>24.0</td>
<td>1.66</td>
<td>0.82</td>
</tr>
</tbody>
</table>

Table 8.2 Port Properties

Figure 8.23 shows the port normal stress exit profiles. Most of the profiles exhibit two peaks due to the highest turbulence levels being within the jet column shear layers. In terms of the primary ports, the sector centre outer port seems to be the exception to this rule, with much higher streamwise values over the rear half of the exit flow. This coincides with high values in the circumferential value of this port and is therefore most likely associated with large scale through port vortical motions. The streamwise normal stresses of the secondary ports do not exhibit the two peaks either, with the outer port having a plateau of high values, and the inner port having a single peak, falling to very low values at the rear of the port. These very low values at the rear of this port coincide with very high radial normal stresses, showing that the flow is highly anisotropic in this region. The turbulent kinetic energy profiles, based on all three available normal stress
Experimental Results: VULCAN Sector Rig

components, are shown in figure 8.24, and indicate values typically around $k/U_{ref}^2 = 0.06$. The radial normal stress is clearly the main contributor to the TKE, highlighted by the peak values in the shear layers.

Typical lengthscales (not shown) are around a quarter of the port diameter, with the main exception to this being the inner primary port radial integral lengthscales. In this region the values are half the port diameter at the leading edge, where the flow is essentially dominated by fluid originating at the outer primary port.

The available shear stress profiles from the x-r and x-z planes are shown in figure 8.25. Like the normal stress profiles, peak uv stresses occur within the jet shear layers, and again the data highlights the relative narrowness of the inner primary jet. In the uw stress profiles, the occurrence of the through port vortices results in the highest shear stresses occurring for the outer primary port, although the peak magnitude is approximately a quarter of the uv stress value.

8.5 Through Port Vorticity

It was identified in section 8.4.3 that the sector centre outer primary port showed the most evidence of through port vorticity on a time averaged basis. As would be expected, this trend is echoed in the instantaneous vector fields which ultimately make up the time average picture. Strong anti-clockwise vortices dominate over 30% of the instantaneous vector maps, an example of which can be seen in figure 8.26. However, also shown in figure 8.26 is an example of an instantaneous clockwise vortex — a rarer event but nonetheless present in around 5% of instantaneous data planes. Some vortices are also evident for the secondary outer port, but the occurrence at the other ports are few and far between. Figure 8.27 shows the time average vorticity field, highlighting that positive values dominate the through port flow, with the maximum (positive) vorticity being 7396/s. Chapter 7 showed alternate means of quantifying the occurrence and strength of vorticity and this is used to provide the data in figure 8.28 which shows the loci of vorticity exceeding 30000/s (normalised by the product of the total instantaneous samples used, 600, and the grid cell area, 0.1945mm$^2$). This alternate view of the fluid dynamics indicates that in 7% of the instantaneous vector fields, the vorticity is four times that of the time averaged value. These observations support the notion that the fuel injector feed
arm situated upstream in the outer annulus is generating the vortices which are seen to pass through the air admission ports.

8.6 Jet Shear Layer Vorticity

It was identified in previous chapters that shear layer vorticity is a common phenomena with jets in cross flow. This section shows that these mechanisms also exist in the complicated jet in cross flow regimes within the fully featured combustor. Although vortices are visible in the raw data of the outer primary port, figure 8.29 uses LES decomposition of an instantaneous vector field to highlight the flow characteristics. Vortices are clearly identifiable in both the upper and lower shear layer, with wavelengths of approximately 1.5 and 1 times the port diameter. If the average total velocity of 0.45U_{ref} at the location of the vortices is assumed to be the convection velocity, this gives shedding frequencies in the upper and lower shear layers of approximately 850Hz and 1300Hz respectively. Using the formula in section 7.5 gives a Strouhal number of 0.082, which is significantly lower than the water analogy results, the reason for which may be related to the port being chuted. Dominant frequencies within the combustor can are undesirable because they can lead to combustion instabilities, increased acoustic excitation and hence noise, and can also threaten the structural integrity of the system. To fully identify the mechanisms and dominant frequencies, a higher temporal resolution, such as that offered by High Speed PIV would be required, as demonstrated on a fuel injector swirler in Hollis and Spencer (2004).

In the shear layer of the inner primary port, large scale rolling vortices were evident in the raw instantaneous data, as shown in figure 8.30. The motions seen here are most likely related to the shearing action between the outer primary and inner primary flows, caused by the misalignment of the jets at the flame tube centerline. In some instantaneous examples, flow originating from the outer primary port can be seen to travel across the can height and be re-ingested by the upstream part of the inner primary port (see figure 8.31). This undesirable flow feature is aided by the large separation of the feed flow from the upstream edge of the chute inlet in the annulus. Such a phenomena could entrain hot gases and carry them towards the leading edge of the inner port, thus degrading the life of the port due to the elevated local temperatures.
8.7 Jet Impingement

The time average data has shown that the outer primary jet dominates the flow topology at the sector centre. Whilst the time average topology is seen in many instantaneous vector fields, the variation of the flow is quite dramatic. Figure 8.31 illustrates the differences seen in instantaneous flow patterns, showing one example where the outer jet completely dominates the primary zone and actually traverses the entire flame tube height, and another example where the desired balanced impingement is experienced. In the majority of cases where the outer jet dominates, the residence time of the fuel-air mixture within the hot primary zone would be extended, resulting in the likelihood of higher NOx emissions, a highly undesirable effect outlined in the introductory chapter.

Griffiths (2000) identified several locations within the combustor where bimodal velocity PDFs were evident. The positions included a location just upstream of the small recirculation above the inner primary jet. A location close to this is highlighted in figure 8.32 along with the relevant radial velocity bimodal PDF data, replicated from Griffiths. Figure 8.33 shows PIV measurements corresponding to a point close to that of the LDA data, including the bimodal type PDFs for both the radial and streamwise velocities. There are differences in the shape of the PDFs from the two techniques, but this is attributable to the slight difference in position, and much smaller sample size for the PIV data. To assess the fluid dynamic behaviour at this location, SVC maps were generated. The radial correlation is shown in figure 8.34. The positively correlated fluid extends over a large proportion of the can height, and in addition, the map shows strong negative correlation between the flow originating from the outer primary port and the inner port, suggesting that when the jets become misaligned, the resistance provided by the opposing jet is reduced, and the absolute radial velocity from the two ports increases as the jet columns instantaneously miss each other.

Figure 8.35 utilises the PDF conditioning technique to explain the correlation topology. This technique was introduced in chapter 3 and its value was demonstrated in chapters 4 and 7. Figure 8.35 uses K=1.5 to calculate the PDF conditioned (Reynolds decomposed) velocity vectors. The vectors clearly explain the 'cloverleaf' pattern of the SVC map, and show that misalignment of the columns results in a starting jet type flow, with large vortex structures above and below the outer primary jet column. This correlation and
PDF conditioned vector topology is similar to that identified for the bimodal core recirculation behaviour in section 7.7. The analysis clearly explains the mechanisms behind the bimodal PDFs at this location seen both in the current work, and in that of Griffiths (2000).

8.8 Fuel Injector Exit Precession

The time average fuel injector exit characteristics were discussed in detail in section 8.4.2, and highlighted that the combustor flow field has a dramatic effect on the flow exiting the swirler, causing the cone angle to effectively be 90° a very short distance downstream of the exit plane. It was also highlighted that a precessing vortex core could be responsible for the turbulent kinetic energy pattern. Temporally resolving precession in the flow was not possible given the limitations of the PIV system, but instantaneous examples of the translation of the swirl centre are provided in figure 8.36. The notion that precession is evident is supported by the correlation maps in figure 8.37, which exhibit strong positive correlation in the region of high TKE, suggesting a periodic motion is present and responsible for the turbulence production mechanisms. Periodicity at the fuel injector could be particularly detrimental to combustor performance because of the likely impact on flame stability.

Griffiths (2000) hypothesized that a combination of the sensitivity of the jet impingement balance and the low pressure generated by the recirculation downstream of the fuel injector swirler, caused the outer primary jet at the sector centre to be drawn upstream, resulting in the misaligned jets. This section and the previous one have proven that instantaneous fluctuations at the jet impingement and fuel injector are significant. It is therefore possible that the precessing vortex core downstream of the fuel injector results in fluctuations in the associated pressure field, and hence drives the instantaneous fluctuating misalignment of the primary jets. To confidently state this connection, it would be necessary to investigate the correlation between directly measured pressure, and PIV measured velocity fields, or via dual plane PIV.
8.9 Closure

This chapter has presented PIV data collected from a fully featured combustor geometry. With air as the fluid medium, and the difficult optical access, this phase of the project provided a real challenge in terms of carrying out PIV measurements. Despite this, comprehensive high quality data were obtained, which provided details of fuel injector swirler and port exit profiles, the likes of which were not feasible with previous point based techniques. Comparisons between LDA and PIV again showed the success of the sub grid filtering correction methodology developed in chapter 6, and the analysis techniques for instantaneous data demonstrated in earlier chapters again proved invaluable in being able to describe underlying fluid dynamic behaviour, and therefore understand likely impact on combustor emissions performance.

One of the main features of the flow is the misalignment of the primary jets at the sector centre. This is believed to be due to the pressure field associated with the swirling flow exiting the fuel injector. The swirling flow shows signs of vortex precession, which would in turn cause fluctuations in the pressure field experienced by the primary jets. This is thought to be the likely cause of the instantaneous changes in jet flow topology. The jets themselves exhibit shear layer vortices like those seen in the water analogy data. The inner primary port (sector centre) data shows that large scale recirculation exists upstream of the jet as a result of the outer primary flow dominating this region.

Through port vortices are a dominant feature of the sector centre outer port flows, with the primary port in particular experiencing strong swirling motions. These are believed to be due to vortex shedding from the fuel injector feed arm and could cause instabilities within the combustor primary zone, leading to flame instabilities and problems associated with that such as increased noise.
Experimental Results: VULCAN Sector Rig

Fig. 8.1 Illustration Data from Selected Measurement Planes
Experimental Results: VULCAN Sector Rig

Fig. 8.2  Sector Centre ($\theta=0.5^\circ$) Time Average Flow Topology Comparison
PIV (left); LDA (right - Griffiths, 2000)

Fig. 8.3  Secondary Port Plane ($\theta=4.35^\circ$) Flow Topology; PIV

Fig. 8.4  Sector Edge ($\theta=7.9^\circ$) Time Average Flow Topology Comparison
PIV (left); LDA (right - Griffiths, 2000)
Experimental Results: VULCAN Sector Rig

Fig. 8.5 Inner Primary Port Exit Time Average Velocities Comparison

Fig. 8.6 Inner Primary Port
Comparison of r.m.s. data from PIV (r=-47.5mm) and LDA (r=-54.8mm)
Fig. 8.7 Turbulent Kinetic Energy (Primary Port Planes)
Sector Centre (top); Sector Edge (bottom)
Experimental Results: VULCAN Sector Rig

Fig. 8.8 Turbulent Kinetic Energy Dissipation Rate (Primary Port Planes)
Sector Centre (top); Sector Edge (bottom)
Fig. 8.9 Streamwise Integral Lengthscales (Primary Port Planes)
Sector Centre (top); Sector Edge (bottom)
Experimental Results: VULCAN Sector Rig

Fig. 8.10 Radial Integral Lengthscales (Primary Port Planes)
Sector Centre (top); Sector Edge (bottom)
Experimental Results: VULCAN Sector Rig

Fig. 8.11 Secondary Port Plane

Turbulent Kinetic Energy (top); TKE Dissipation Rate (bottom)
Fig. 8.12 Secondary Port Plane Integral Lengthscales
Streamwise (top); Radial (bottom)
Experimental Results: VULCAN Sector Rig

**Fig. 8.13** Fuel Injector Exit (r-z) Plane, x=6.0mm

*Time Average Cartesian Velocities [looking along positive x axis]*

**Fig. 8.14** Fuel Injector Exit (r-z) Plane, x=6.0mm

*Local Time Average Polar Velocities [relative to injector centreline]*
Experimental Results: VULCAN Sector Rig

Fig. 8.15 Fuel Injector Exit Velocity Profiles at x=6mm

Full diameter raw data showing asymmetry (top);
Symmetric (averaged) data (bottom)
(Black data indicates measurements from the xr plane; red data indicates rz plane)
Experimental Results: VULCAN Sector Rig

Fig. 8.16 Downstream of Injector Exit Plane, x=11.5mm
Local Time Average Polar Velocities [relative to injector centreline]

Fig. 8.17 Axial Planes at Outer Primary Jet
Port centreline, x=37mm [composite plot of two FoVs] (Left);
Upstream port edge, x=29mm (top right); Trailing edge, x=44mm (bottom right)
Experimental Results: VULCAN Sector Rig

**Fig. 8.18** Fuel Injector Exit Normal Stress Profiles (x=6mm)
Symmetric (averaged) data

**Fig. 8.19:**
Left: Turbulent Kinetic Energy at Injector Exit Plane (using vv and ww only) (x=6mm)
Below – Turbulent Kinetic Energy (using all three normal stresses); Total Velocity; Average Integral Lengthscale - at Injector Exit
Fig. 8.20 Polar Shear Stresses at the Injector Exit Plane

Fig. 8.21 Fuel Injector Exit Shear Stress Profiles (x=6mm)

Symmetric (averaged) data
Experimental Results: VULCAN Sector Rig

Fig. 8.22 Port Exit Time Average Velocity Profiles
Streamwise Velocity, $U/U_{\text{ref}}$ (top); Radial Velocity, $V/U_{\text{ref}}$ (middle);
Circumferential Velocity, $W/U_{\text{ref}}$ (bottom)
Experimental Results: VULCAN Sector Rig

Fig. 8.23 Port Exit Normal Stress Profiles
Streamwise, $uu/U_{ref}^2$ (top); Radial, $vv/U_{ref}^2$ (middle);
Circumferential, $ww/U_{ref}^2$ (bottom)
Experimental Results: VULCAN Sector Rig

![Fig. 8.24 Port Exit TKE Profiles](image1)

**Fig. 8.24 Port Exit TKE Profiles**

![Fig. 8.25 Port Exit Shear Stress Profiles](image2)

**Fig. 8.25 Port Exit Shear Stress Profiles**

$uv/U_{ref}^2$ (top); $uw/U_{ref}^2$ (bottom)

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Fig. 8.26 Instantaneous Examples of Through Port Vorticity

Anti-Clockwise (top); Clockwise (bottom)

Outer Primary Port, Sector Centre
Experimental Results: VULCAN Sector Rig

Fig. 8.27 Time Average Through Port Vorticity
Outer Primary Port, Sector Centre

Fig. 8.28 Loci of Through Port Vorticity, $\omega > 30,000$/s
Outer Primary Port, Sector Centre
Experimental Results: VULCAN Sector Rig

Fig. 8.29 LES Decomposed Example of Shear Layer Vorticity
Outer Primary Port, Sector Centre

Fig. 8.30 Instantaneous Examples of Shear Layer Vortex
Inner Primary Port, Sector Centre
Fig. 8.31 Instantaneous Examples of Primary Jet Behaviour (Sector Centre)
Outer Jet Dominating (top); Balanced Impingement (bottom)
Experimental Results: VULCAN Sector Rig

Fig. 8.32 Bimodal PDF at Weak Jet Impingement (Griffiths, 2000)
(inset shows positions where bimodal PDFs were obtained)

Fig. 8.33 Bimodal PDFs near weak Jet Impingement; at (x,r)=(38.0,-9.5)
[in Small Recirculation Zone Upstream of Inner Primary Jet]
Fig. 8.34 Spatial Velocity Correlation Map of Radial Velocity in Small Recirculation Zone Upstream of Inner Primary Jet

Fig. 8.35 PDF Conditioned (Reynolds Decomposed) Vectors Superimposed on SVC Map ($R_{21}$)
Experimental Results: VULCAN Sector Rig

Fig. 8.36: Instantaneous examples of vortex precession at fuel injector exit

Fig. 8.37 SVC Maps; Fuel Injector Exit Plane
Radial Velocity, $R_{22}$ (top); Circumferential Velocity, $R_{33}$ (bottom)
Chapter 9

CONCLUSIONS AND RECOMMENDATIONS


Conclusions and Recommendations

9.1 Conclusions

Through the use of best practice optimisation procedures and correction techniques developed in this thesis, Particle Image Velocimetry has been successfully applied to two gas turbine combustor representative geometries. The methodologies developed for the analysis of PIV data has allowed the identification of instantaneous flow field structures not seen previously, phenomena which are invisible in the time average descriptions and have potential impact on combustor performance. Proof has been supplied that knowledge of the instantaneous flow is vital if we are to understand the mixing processes within the combustor, and therefore be able to design out flow behaviour which may increase harmful emissions. The PIV techniques and methodologies developed during this project have proved highly successful in efficiently providing information about the underlying fluid dynamics of the flow in gas turbine combustors. These methods will prove equally valuable for any turbulent flow where PIV is utilised as the measurement technique.

A method for calculating the correlation functions from velocity data obtained using PIV has been developed and validated through the use of synthetic data with known correlation characteristics. The algorithm has been used extensively in the examination of the flows studied experimentally. The ability of the PDF conditioned velocity fields to efficiently explain the topology associated with the correlation maps was also established as an invaluable analysis tool. These methodologies could equally be applied to other planar velocity data, such as that from LES and DNS computational simulations.

Integral lengthscales have been calculated for whole field data, and methods to overcome problems associated with the calculation of these values close to domain edges have been successfully implemented. Additionally methodologies have been presented to calculate fluid dynamic properties that were previously extremely difficult to obtain. From the integral lengthscales and some assumptions of isotropy, data including turbulent kinetic energy dissipation rate and Kolmogorov length scales have been calculated. These methods provide one of the very few means of obtaining this type of fluid dynamic information from experimental data.
Conclusions and Recommendations

Sub grid filtering occurs in PIV analysis due to the low pass filtering effect of the finite interrogation cells which make up the measurement domain. This reduces the measured r.m.s. and results in elevated correlation values. Synthetic data has been used to validate a method to correct statistical PIV data for the effects of sub grid filtering. The method was then calibrated using a comprehensive set of experimental data, and enables accurate estimates of r.m.s. velocities to be obtained, regardless of the size of the PIV field of view. In addition to this, trends were identified for the ratio of the true and measured integral lengthscale, an effect not documented until now. These techniques were applied to the data collected from the combustor representative geometries, where it was shown that application of the corrections resulted in better estimates of the flow statistics.

Detailed data sets were obtained for the generic water analogy facility and the fully featured combustor sector airflow rig, the latter providing a particularly challenging turbulent environment for the application of PIV. Comprehensive data sets were collected in both facilities, providing spatial data resolution unheard of with point based techniques, and measurement planes previously unobtainable. In addition to the wealth of statistical data, several instantaneous phenomena have been identified and quantified via the use of decomposed velocity fields, correlation maps, and conditional averaging.

Modal behaviour was identified in the annulus of the water analogy facility due to unstable impingement of the flow at the rear of the admission port. This occurred at the 50% bleed condition regardless of jet to cross flow ratio and is thought to manifest itself in small changes of the jet shape and angle. This also led to evidence of coupling between the annulus flow and rear portion of the ensuing jet via the use of correlation maps.

Shear layer vortices between the jet column and core flow were identified in both test facilities and were especially prevalent in the leading edge shear layer. This vorticity had distinct wavelength and hence frequency, identified via decomposed vector fields. Laser Induced Fluorescence served to highlight the vortical motion. The fact that these vortices exist in such a highly turbulent environment illustrates the strength of the motions, and could have important consequences on jet-core mixing processes within the combustor.
Conclusions and Recommendations

Periodic behaviour was identified in the water analogy facility core recirculation zone. PDF conditioning and bimodal PDFs were utilized to identify the mechanisms behind the instantaneous flow topology. The process is analogous to a jet in counterflow and the instabilities arise from the reverse flow column battling against the incoming core flow. This can result in the asymmetric collapse of the recirculation zone, which can cause vortices to be shed into the core wall boundary layer. These near wall vortices become trapped in the boundary layer and survive as they are convected downstream. Such a process could cause the breakdown of liner cooling films in gas turbine combustors.

PDF conditioning and correlation maps were successfully applied to the data from the fully featured combustor and used to identify the flow topology associated with misalignment of the primary jets downstream of the fuel injector. Like the bimodal recirculation zone, the instantaneous misalignment results in starting jet type topology. Instances were shown where the outer primary port flow completely dominates the primary zone, traversing the flame tube and being re-ingested by the forward portion of the inner primary port. In an actual combustor, this would result in hot gases around the chute leading edge, which would reduce the combustor life and efficiency. The misalignment is believed to be due to the pressure field associated with the swirling flow exiting the fuel injector, which itself shows signs of a precessing vortex core. As the core precesses, changes in the associated pressure field could be responsible for influencing the primary jet flow, causing the instantaneous misalignment. The primary jet misalignment causes large integral lengthscales and reduced turbulent kinetic energy. This could lead to incomplete mixing of the combustion products and increased emissions.

Through port vortices developing in the annulus were identified as being present. In the water analogy facility these vortices were strongest at lower bleed ratios. The loci of maximum vorticity magnitude were plotted and used to establish distinct trends in the occurrence of strong through port vortices with decreasing bleed ratio. At very low bleed ratios flow coupling between adjacent ports was interpreted and believed to be responsible for recirculation in the radial plane at the outer annulus wall between ports. In the airflow facility, through port vortices were only evident at ports downstream of the burner arm which protrudes through the outer annulus. It is believed that vortex shedding
from the arm caused the particularly strong vortices seen in the outer primary port. This may result in dominant frequencies being convected into the primary zone of the flame tube, which could cause flame instability and therefore increased noise and incomplete combustion.

9.2 Recommendations

This project has applied PIV to the highly turbulent flow within gas turbine combustors. The sub grid correction method for r.m.s. and integral lengthscale quantities has been proven to be a very successful technique and further investigation of the effect on shear stresses, vorticity, etc. should now be carried out. Initial findings suggest shear stresses are affected in a similar way to r.m.s velocities.

During the project PIV methods have advanced, with stereoscopic, dual plane, and high speed possibilities now being realistic. Stereoscopic PIV at locations such as the fuel injector exit plane would provide valuable information on the highly three dimensional exit characteristics. Dual plane PIV within the VULCAN rig (in addition to better optical access for the annular flows) would allow correlations between annulus and core to be examined. Alternatively, dual measurements could be employed to study correlations, such as hot wire anemometry probes in the annulus combined with PIV within the flame tube. High speed PIV would also provide invaluable information on the temporal development of the flow. Laser induced fluorescence has been briefly used in this project and the possibility of simultaneous PIV and LIF would enable the study of scalar mixing processes.

The experimental results detailed in this document have provided a wealth of information, but carrying out experiments remains expensive, so efforts to improve computational methodology must be pursued. LES (and even DNS) are more viable nowadays due to increased computing power and the ability of these techniques to predict the instantaneous fluid dynamic behaviour seen within this project would give confidence in these simulation tools. During the project several instantaneous fluid dynamic phenomena have been identified, but it is difficult to say whether these are all degrading to the combustor performance. Studies should be made into any benefits large scale unsteadiness might bring, rather than assuming all are undesirable.
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