An investigation of in-cylinder flows in a direct injection compression ignition engine using particle image velocimetry

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AN INVESTIGATION OF IN-CYLINDER FLOWS IN A DIRECT INJECTION COMPRESSION IGNITION ENGINE USING PARTICLE IMAGE VELOCIMETRY

By

Brian J. Stapleton
B.Eng. (Hons.)

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

November 2005

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Abstract

The Compression Ignition (CI) engine has been the engine of choice for heavy duty on and off-highway power generation due to its robustness and high fuel efficiency. CI engines also emit lower amounts of carbon dioxide (CO₂) than other prime-movers. Many governments are currently promoting the reduction of CO₂ emissions and hence the efficient and robust CI engine is expected to be used increasingly in many on and off-highway sectors. However, CI engines emit pollutants such as oxides of nitrogen (NOₓ), particulate matter (PM) and hydrocarbons (HC), and hence increasingly strict legislative limits to these are being phased in over the next 10 years. To aid the development of engines that meet this legislation, designers require a better understanding of the combustion and pollution formation processes in the engine cylinder. In-cylinder air charge motion is known to fundamentally affect the mixing and combustion of the injected fuel in CI engines and hence the emissions produced by the engine. Therefore, characterisation and quantification of the in-cylinder flow is an important step in the process of achieving the conditions necessary for optimal combustion.

This research has investigated, using optical measurement techniques, the in-cylinder air flow characteristics for three different CI engine inlet regimes. An optical engine was developed with directed and helical inlet ports, which were designed to provide similar swirl numbers as measured on steady-flow swirl test rigs. The aim of the research was to link the steady-flow test rig results with measurements taken, using Digital Particle Image Velocimetry (DPIV) at top dead centre (TDC). Testing involved two different piston bowls (deep and shallow bowl pistons) and three different engine speeds (800 rpm, 1200 rpm and 1600 rpm), to observe their affects on the charge air motion at TDC. A phenomenological model based on kinetic energy analysis of the flow was used in the current work for comparison with measured data. The model used measured data from the steady-flow swirl rig as initial conditions to model the flowfield of the charge air at TDC and these results were compared with measured data.

The measured data compared well with the modelled data for the shallow bowl piston, with a velocity profile that was between solid body rotation and a pre-determined velocity profile in the model. Results from the model for deep bowl piston geometry showed less agreement with the measured data and this was believed to be due to the effect of squish motion on the charge air motion at TDC. The velocity profile of the different inlet port geometries exhibited good repeatability across different engine speeds. This result showed how the steady-flow swirl rigs under-predict the charge air motion for combined inlet ports. Whole field datasets are valuable for the validation of computational fluid dynamic (CFD) models.
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Finally, my wife, Nicky Stapleton, whose love has supported me through challenging times. I dedicate this thesis to Nicky.
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<td>a</td>
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<tr>
<td>$A_c$</td>
<td>Cross-Sectional Area of Cylinder (m(^2))</td>
</tr>
<tr>
<td>$A_{E,L}$</td>
<td>Effective Leakage Area (m(^2))</td>
</tr>
<tr>
<td>$A_v$</td>
<td>Element of the Valve Open Area (m(^2))</td>
</tr>
<tr>
<td>B</td>
<td>Cylinder Bore Diameter (m)</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom Dead Centre</td>
</tr>
<tr>
<td>bmep</td>
<td>Brake Mean Effective Pressure (N.m(^{-2}))</td>
</tr>
<tr>
<td>bsfc</td>
<td>Brake Specific Fuel Consumption (g.kW(^{-1}).hr(^{-1}))</td>
</tr>
<tr>
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<td>$C_{ax}$</td>
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<tr>
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<td>Coefficient of Discharge</td>
</tr>
<tr>
<td>$C_{inst.}$</td>
<td>Instantaneous Speed (m.s(^{-1}))</td>
</tr>
<tr>
<td>$C_m$</td>
<td>Mean Speed (m.s(^{-1}))</td>
</tr>
<tr>
<td>$C_P$</td>
<td>Constant Pressure Specific Heat (kJ.kg(^{-1}).K(^{-1}))</td>
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<tr>
<td>$C_u$</td>
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<tr>
<td>$C_r$</td>
<td>Constant Volume Specific Heat (kJ.kg(^{-1}).K(^{-1}))</td>
</tr>
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<td>CCD</td>
<td>Charged Coupled Device</td>
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<tr>
<td>CFD</td>
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<td>CI</td>
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<td>$d_{pixel}$</td>
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$D$  Dimensionless Swirl Parameter
$D'$ Dimensionless Geometric Parameter of Swirl
$D_B$ Piston Bowl Diameter (m)
$D_k$ Gradient Diffusion of the Turbulence (J)
DI Direct Injection
DPIV Digital Particle Image Velocimetry
DSR Dynamic Swirl Ratio
$EA$ Ensemble Averaged
$f^a$ Ratio of Focal Length
$f_w$ Frequency (Hz)
$F_p$ Viscous Drag Force (N)
FFT Fast Fourier Transform
$G$ Bump Clearance (m)
$h_B$ Height of Piston Bowl (m)
$h_v$ Absolute Valve Lift (m)
HDD Hard Disk Drive
HPIV Holographic Particle Image Velocimetry
HWA Hot Wire Anemometry
$\dot{i}$ Angular Momentum Flux (kg.m$^2$.s$^{-1}$)
$I_c$ Moment of Inertia (kg.m$^2$)
$I_{cylinder}$ Moment of Inertia for Cylinder (kg.m$^2$)
$I(i,j)$ Intensity Distribution in Cross Correlation
$I(r)$ Intensity Distribution in a Bessel Function
IC Internal Combustion
IVC Inlet Valve Closure
$j$ Cycle Number
$J_i$ Flux of Angular Momentum into Cylinder (kg.m$^2$.s$^{-1}$)
$J_{1}(r)$ $1^{st}$ Order Bessel Function
$k$ Turbulence kinetic Energy (J)
$KE_{rotational}$ Rotational Kinetic Energy (J)
$l_t$ Integral Length Scale (m)
$L_V$  Valve Lift (m)
LDV  Laser Doppler Velocimetry
LIF  Laser Induced Fluorescence
$m$  Mass (kg)
$m$  Mass Flow Rate (kg.s$^{-1}$)
$m_s$  Mass of Charge Air in Bowl (kg)
$m_c$  Mass of Charge Air (kg)
$m_p$  Mass of Particle (kg)
$M$  Image Magnification
$m_{ep}$  Mean Effective Pressure (N.m$^{-2}$)
$n$  Crankshaft Speed (rad.s$^{-1}$)
$n_d$  Swirl Speed (rad.s$^{-1}$)
$N$  Crankshaft Speed (rpm)
NO  Nitric Oxide
NO$\text{x}$  Oxides of Nitrogen
$p_{\text{squish}}$  Turbulent Squish Production per unit mass (J.kg$^{-1}$)
$p_{\text{swirl}}$  Turbulent Swirl Production per unit mass (J.kg$^{-1}$)
$P$  Pressure (N.m$^{-2}$)
$P_{v}$  Volumetric Production of Turbulence (J.m$^{-3}$)
pixel  Picture Element
PIV  Particle Image Velocimetry
$q$  Normalised Diameter (m)
$r$  Radial Spatial Co-ordinate (m)
$r_b$  Radius of Piston Bowl (m)
$r_c$  Compression Ratio
$R$  Gas Constant (J.kg$^{-1}$K$^{-1}$)
$R_{\text{y}}(x,y)$  Cross Correlation of Intensity Distribution
$R_{\text{cyl}}$  Radius of Cylinder (m)
$R_{\text{pixel}}$  Ratio of Image Size to Pixel Size
$R_{\text{gen}}$  Gas Molar Constant for Air (J.kg$^{-1}$K$^{-1}$)
RAM  Random Access Memory
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<td>Re</td>
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<tr>
<td>RMS</td>
<td>Root Mean Square</td>
</tr>
<tr>
<td>rpm</td>
<td>Revolutions per Minute (rad.s(^{-1}))</td>
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<td>s</td>
<td>Stroke (m)</td>
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<tr>
<td>(s_{ratio})</td>
<td>Ratio of Connecting Rod Length to Crank Radius</td>
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<tr>
<td>(\bar{S}_p)</td>
<td>Mean Piston Speed (m.s(^{-1}))</td>
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<td>(S_t)</td>
<td>Swirl Ratio</td>
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<tr>
<td>(S_{sutherland})</td>
<td>Sutherland Constant</td>
</tr>
<tr>
<td>sfc</td>
<td>Specific Fuel Consumption (g.kW(^{-1}).hr(^{-1}))</td>
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<tr>
<td>SI</td>
<td>Spark Ignition</td>
</tr>
<tr>
<td>(St)</td>
<td>Stokes Number</td>
</tr>
<tr>
<td>(t)</td>
<td>Time (s)</td>
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<tr>
<td>(T)</td>
<td>Temperature (K)</td>
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<td>(T_d)</td>
<td>Driving Torque (N.m)</td>
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<tr>
<td>(T_f)</td>
<td>Torque due to Wall Friction (N.m)</td>
</tr>
<tr>
<td>(T_r)</td>
<td>Restraining Torque (N.m)</td>
</tr>
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<td>(T_s)</td>
<td>Torque exerted by the Shear Forces (N.m)</td>
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<td>TDC</td>
<td>Top Dead Centre</td>
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<tr>
<td>(u)</td>
<td>Fluctuating Velocity (m.s(^{-1}))</td>
</tr>
<tr>
<td>(u')</td>
<td>Turbulence Intensity (m.s(^{-1}))</td>
</tr>
<tr>
<td>(U)</td>
<td>Instantaneous Velocity (m.s(^{-1}))</td>
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<td>(\bar{U})</td>
<td>Mean Velocity (m.s(^{-1}))</td>
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<td>(U_0)</td>
<td>Amplitude (m)</td>
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<tr>
<td>(v_p)</td>
<td>Particle Velocity (m.s(^{-1}))</td>
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<tr>
<td>(v_{sq})</td>
<td>Squish Velocity (m.s(^{-1}))</td>
</tr>
<tr>
<td>(\dot{V})</td>
<td>Volume (m(^3))</td>
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<tr>
<td>(\dot{V})</td>
<td>Volumetric Flow Rate (m(^3).s(^{-1}))</td>
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<tr>
<td>(V_p)</td>
<td>Volume of Piston Bowl (m(^3))</td>
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<tr>
<td>(V_c)</td>
<td>Clearance Volume (m(^3))</td>
</tr>
<tr>
<td>(V_{ cyl})</td>
<td>Swept Cylinder Volume (m(^3))</td>
</tr>
</tbody>
</table>
$V_d$  Displaced Volume (m$^3$)
$V_\theta$  Tangential Velocity (m.s$^{-1}$)
$V_R$  Radial Velocity (m.s$^{-1}$)
$z_0$  Distance between Image Plane and Lens (m)
$Z_0$  Distance between the Lens and Object Plane (m)
$z_p$  Distance between the Piston Crown and the Cylinder Head (m)
$\alpha$  Crank Angle (rad)
$\beta$  Density Ratio
$\gamma$  Specific Heat Ratio
$\varepsilon$  Dissipation of Turbulence kinetic Energy (J)
$\eta_{f,i}$  Theoretical Indicated Fuel Efficiency
$\theta$  Crank Angle in Degrees (degrees)
$\mu$  Dynamic Viscosity (N.s.m$^{-2}$)
$\Omega$  Angular Momentum of Charge at BDC (kg.m$^2$.s$^{-1}$)
$\rho_F$  Fluid Density (kg.m$^{-3}$)
$\rho_P$  Particle Density (kg.m$^{-3}$)
$\phi$  Phase Lag
$\lambda$  Laser Wavelength (m)
$\chi$  Coefficient
$\sigma$  Particle Density Ratio
$\phi$  Segment Angle of Interest (rad)
$\omega$  Angular Velocity (rad.s$^{-1}$)
$\Gamma_c$  Angular Momentum of Charge (kg.m$^2$.s$^{-1}$)
$\tau_{air}$  Shear Stress between Adjacent Fluid Layers (kg.m$^2$.s$^{-2}$)
$\tau_p$  Time Constant (s)
$\tau_w$  Shear Stress at Fluid-Solid Interfaces (kg.m$^2$.s$^{-2}$)
$\nu_t$  Turbulent Viscosity (s$^{-1}$)
$\Psi$  Angle of Radius ($r$) to Horizontal (rad)
$\psi$  Angle of Measured Vector ($U$) to Horizontal (rad)
$\kappa$  Angle between Tangential Vector ($V_\theta$) and Measured Vector ($U$) (rad)
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Chapter 1 Thesis Introduction

1.1 Introduction

The Internal Combustion (IC) engine is of major importance in transport and power generation applications and is expected to be so for the foreseeable future. Compression Ignition (CI) (i.e. Diesel) engines have been the engine of choice for both heavy duty on and off-highway vehicles and power generation due to their robustness, high fuel economy and potential maximum size. They are becoming increasingly popular for passenger car applications due to their high fuel economy, low CO₂ emissions and high torque characteristics. CI engines have become highly refined over recent years, with particular attention having been focussed on fuel injection and the inlet charge air systems, both of which have led to significant improvements in engine performance and emissions.

As emissions legislation becomes more stringent and the necessity for clean, efficient engines, the complex interaction between fuel and charge air has to be better understood. This is because charge air and fuel mixing is of fundamental importance in the combustion process.

The aim of the research presented in this thesis was to measure in-cylinder charge air motion in a real geometry motored CI engine using particle image velocimetry (PIV) techniques, with the goal of understanding further the behaviour of charge air motion. The objective was to obtain dynamic flow data to validate phenomenological zero-dimensional swirl models and provide datasets for potential computational fluid dynamic (CFD) model validation.

Chapter 1 gives a brief introduction to in-cylinder charge air motion in CI engines, the basic theory of charge air motion during the compression stroke and how this charge air motion is generated during the induction process of the engine. This is followed by a chapter-by-chapter overview of the thesis.
1.2 In-cylinder air charge swirl

Engine in-cylinder air swirl is defined as the organised rotation of the charge about the cylinder axis. Swirl is created during the engine’s induction process due to the inlet port (or valve) geometry giving the intake air angular momentum. The swirl structure is then modified during the compression stroke. Although swirl exists throughout most of the engine cycle, what is of particular interest to the diesel engineer is the charge swirl velocity at the end of the compression stroke when fuel injection process takes place. Swirl is of fundamental importance as it affects the mixing process between the air and the fuel, hence the speed and completeness of the combustion process. This directly affects the engine’s power output, fuel economy and exhaust emissions.

The air charge angular momentum (swirl) is modified during the compression stroke by the design of the piston. Most direct injection (DI) diesel engines have the combustion chamber machined into the piston crown, commonly referred to as “bowl-in-piston”. The design of this bowl-in-piston has an important influence on the swirl level at the time of injection.

1.3 Basic swirl theory

The swirl momentum that is generated during the induction stroke continuously loses angular momentum due to friction effects and turbulent dissipation during the compression stroke, with expected losses of between 25-35% (Heywood 1988). During actual engine running, the swirling flow velocity during the induction period is not uniform and is constantly changing. The first half of the induction stroke has relatively high velocities of the intake charge, whilst in the latter half of the induction stroke the velocities are substantially lower. During the compression process, in a bowl-in-piston design combustion chamber, the swirling charge filling the engine cylinder is forced into the smaller bowl radius where the angular momentum of the charge is maintained (but subjected to the friction and turbulent dissipation losses as described previously) and hence the tangential velocity of the charge is increased with
decreased radius. The piston bowl usually has a ‘pip’ to aid in centring the flow in the cylinder about the cylinder axis. For the work completed in this investigation, a ‘pip’ design was used, as shown in Figure 1.1.

Figure 1.1 Piston with ‘bowl-in-piston’ combustion chamber and centring pip

The most common way of quantifying the swirl of an engine is by the swirl ratio, defined as the swirl angular velocity normalised by the engine crankshaft speed.

\[
Swirl \ Ratio = \frac{Swirl \ Angular \ Velocity}{Engine \ Crankshaft \ Angular \ Velocity}
\]  

(1.1)

This is usually the defined ratio of swirl taken from steady-flow swirl rigs normalised by the equivalent engine crankshaft speed at bottom dead centre (BDC).

1.4 Types of swirl inlet geometry and effect on emissions

Swirl is most commonly generated by the design of the inlet ports in the cylinder head. Inlet port design is discussed in detail in Chapter 2, here we shall simply focus
on the affect different swirl levels have on engine performance so that the motivation for its measurement can be justified.

As stated previously, the importance of quantifying swirl will aid in understanding the mixing of the fuel and air during the fuel injection period. Ricardo and Hempson, as cited by Stone (1999), researched the effects of swirl on an engine where the swirl ratio could be varied. The variation in the swirl rate at constant fuel flow rate was investigated to observe the affect on brake mean effective pressure (bmep), the brake specific fuel consumption (bsfc), exhaust temperature and maximum cylinder pressure. Results from this study are presented in Figure 1.2.

![Figure 1.2 Variation of engine performance with swirl ratio at constant fuelling rate (Stone 1999)](image)

Ricardo and Hempson found that there was an optimum swirl ratio (i.e. 10.5) for maximum bmep and minimum bsfc, however, due to the increased maximum pressure and the rate of pressure rise, the engine was found to be 'rough' and noisy.

Various investigations have been carried out which link the emissions of an engine to the inlet swirl regime. Early work by Khan et al. (1972) used shrouded inlet valves to investigate various swirl rates. This work showed a direct link between increased swirl and increased in-cylinder temperatures, resulting in higher NOx emissions and
A further finding in the work by Khan et al. was that changes in the swirl had a much stronger effect on smoke emissions than the NO$_x$ emissions (in the region of 2-3 times the effect for similar engine running conditions). Work by Shimada et al. (1986) also studied the affect of swirl on emissions. In this research a specially designed double inlet system was used to alter the swirl entering the engine by means of butterfly valves. This work tested different engine speed parameters with varying load and inlet swirl conditions. One of the main contributions from this work was showing that there is not an optimal swirl condition for all engine operating conditions. It was reported that for the full load condition at a higher engine speed (2200 rpm), the swirl was lowered (swirl number = 2) to achieve low bsfc and smoke emissions. For full load at medium engine speed (1400 rpm), the swirl achieved the lowest bsfc and smoke with an increased swirl (swirl number = 2.5 – 3.3). For full load at lower engine speed (600 rpm), the high swirl case (swirl number = 4.3) was found to produce the lowest bsfc and smoke emissions. The NO$_x$ levels were found to be directly affected by the swirl, where the NO$_x$ increased with increasing swirl. Further tests involved part loading the engine, this returned interesting results in that for an engine load less than 50% at low engine speed, increasing swirl increased the bsfc. This was attributed to heat transfer losses in the engine under low load. The test matrix devised for this work led to a design for operating the butterfly valves to control the inlet swirl for varying parameters of the engine (based on engine speed and loading), resulting in improved low speed torque characteristics and lowered bsfc without increasing NO$_x$ emissions. Other areas of engine performance could also be improved by the use of the variable swirl inlet, in particular cold startability and white smoke emissions.

Recent work by Benajes et al. (2004) further investigated the effect of swirl on emissions for varying load and engine speed. Similar to the findings above, it was concluded that there was not an optimum swirl number which would suit all conditions. This work used butterfly valves in the inlet manifold to control which inlet port (helical or directed) delivered the air charge, allowing for variation of swirl during running conditions. The findings showed that with increasing swirl, up to a certain level, a more intense pre-mixed combustion phase was observed and an improved diffusion controlled combustion phase. This increased the NO$_x$ formation but reduced the smoke production. When the swirl was increased beyond the
optimum, the pre-mixed combustion phase always improved, but the excess swirl was seen to affect detrimentally the diffusion controlled combustion phase, resulting in increased smoke emission and fuel consumption. This work concentrated on mapping optimum swirl levels to load and engine speed conditions, as was the focus of the work by Shimada et al. (1986).

1.5 Swirl measurement and modelling

It is current industrial practise to measure swirl on steady-flow swirl rigs. This method of swirl measurement is achieved by attaching the cylinder head to a dummy cylinder and creating a pressure drop across the inlet port. A honeycomb flow straightener (or paddle wheel) is mounted axially in the dummy cylinder and measures torque exerted upon it from the swirling airflow. This measured torque combined with flow measurement and pressure drop across the cylinder head gives an indication of the swirl strength that will be experienced in the engine at BDC. This is the basis of the Tippelmann (1977) steady-flow swirl rig and will be discussed in further detail later. Swirl rigs are used to determine the general level of swirl an engine cylinder head creates, but they do not provide detailed spatial information of the swirl and no information on the swirl occurring at TDC under fully motored conditions. In modelling the flow from BDC to TDC, the results from these steady-flow swirl rigs are often used as the initial charge air motion boundary conditions.

The above findings set out the motivation for measuring swirl in CI engines. Later, in Chapter 2, a more detailed literature review is presented.

1.6 Thesis Overview

The aim of this thesis is to better understand the air flow structures that occur at TDC in CI engines and to use this information for model validation. As the modelling of engines becomes more powerful, it is important to validate the computer predicted results. For this purpose a research engine was developed in this project which was
optically accessible to allow 2-D Particle Image Velocimetry (PIV) measurements to be made, thus permitting actual velocity maps to be obtained of the air flow at TDC.

The engine was tested using three different inlet port configurations, three different engine speeds and two different piston bowl designs. The three port configurations consisted of a helical port, a directed port and combination of both. Having a wide test envelope and comprehensive dataset has provided further understanding of how the swirl is affected by altering these main parameters.

The following describes the thesis content chapter-by-chapter:

Chapter 1 has provided a basic introduction to swirl, how swirl is normally introduced into the engine and the importance of swirl with respect to combustion and modelling. It then summarises the thesis content including the major contributions to knowledge.

Chapter 2 discusses the fundamentals of CI engines with particular emphasis on the charge air motion and the factors that affect it. It introduces how a CI engine works in principle and practice and how swirl is generated during the induction stroke. Other aspects of in-cylinder flow and their affects on the swirl are discussed, introducing the effect of squish, turbulence and cyclic variability.

Chapter 3 reviews and discusses non-contact measurement techniques that can be employed in measuring in-cylinder engine flows. The PIV technique is discussed in detail showing the advantages and limitations of this method of measurement. Flow seeding is discussed in detail since optical measurement techniques infer the motion of the fluid by the use of small seeding particles introduced into the flow.

Chapter 4 presents a literature survey of work that has been carried out using non-contact measurement techniques in IC engines. This literature survey is primarily aimed at DI CI engines.

Chapter 5 explains in detail the main conventional industrial method employed for measuring steady-flow swirl. It then reviews previous work carried out for predicting swirl spin-up in the engine during the compression stroke and develops a model to
determine in-cylinder swirl at TDC. This mathematical model incorporates more complex models of the swirl spin-up, including squish, turbulent dissipation, kinetic energy, air temperature and air density.

Chapter 6 reports the experimental set-up used in this research and reviews the experimental procedure employed to obtain data. It is explained how the engine was made optically accessible for the PIV measurement equipment, instrumentation and control of the engine is also described and how this interfaced with the PIV system. The PIV system used is described and how it is arranged to measure the engine charge air flow, with further information to the experimental limitations and accuracy of this system. Seeding of the engine charge air flow is discussed regarding which seeding type was preferable, the reasons for that choice and the method for introducing the seed into the engine inlet ports. The validation of these velocity vector maps is also explained.

Chapter 7 presents the validated PIV datasets that were obtained from the engine. A variety of vector maps are displayed showing both instantaneous and mean results for all 18 different test points in this research. The raw data presented in this chapter is further analysed by using a new analysis routine to interpret the results. It was found that the best method of presenting data was using a dimensionless unit called the 'dynamic swirl ratio'. The swirl of the measured data is considered to rotate about the geometric centre of the piston bowl pip and a 'normalised dynamic swirl ratio' determined. The results of the measured data are then compared with the modelled data that was presented in Chapter 5. A discussion of these results follows. The results are then considered for the effect of engine speed and inlet ports. Using the non-dimensional parameter of normalised swirl ratio, the velocity profile for different conditions were directly compared and discussed. The planar 2-D PIV result was then compared with a 3-D PIV result obtained using Holographic PIV, which has also been employed for measuring flows on the engine in this research. The novel Holographic PIV technique is outlined and the advantages and disadvantages of this advanced measurement technique is discussed.

Chapter 8 presents the conclusions from this research and provides suggestions for future work.
1.7 Contributions to knowledge from this research

From this research the following important contributions to knowledge have been made:

1. The development of a production geometry optical diesel engine with typical compression ratio. A method of creating an optical piston allowing for a piston bowl pip.
2. A comprehensive dataset of PIV measurements in a CI engine in the combustion bowl at varying speeds, deep and shallow piston bowl designs and different inlet geometries.
3. Comparison of dynamic swirl ratio to steady-flow swirl rig results and identification of trends and bulk flow motion in the piston bowl.
4. Comparison of 2-D PIV with 3-D Holographic PIV measurements in a diesel engine piston bowl.

The results obtained have improved both experimental methods and basic understanding of swirl in a CI engine.

1.8 Summary

This chapter has introduced the basic concept of swirl, its importance in IC engines and has illustrated the necessity of research in this area. An overview of the thesis was given. Chapter 2 will consider CI engines in greater detail and relevant factors that affect the in-cylinder flow.
Chapter 2    CI engine operation and charge air motion

2.1 Introduction

This chapter reviews the fundamental aspects of a CI engine with particular emphasis to the in-cylinder charge air motion. An introduction to fuel injection and piston bowl design are also given.

2.2 Overview of the CI engine

The reciprocating internal combustion engines are the most common form of engine or prime mover currently in use. This work is interested in the CI engine (more commonly known as the diesel engine after the inventor Rudolph Diesel) which uses the rise in temperature from the compression of the inducted charge air to initiate ignition of the injected fuel. The CI engine most commonly operates on a four-stroke cycle, which comprises two crankshaft revolutions and is described in the following sequence of events:

**Induction Process:** The inlet valve of the engine is opened and the piston moves away from TDC and draws air into the cylinder.

**Compression Process:** The piston moves from BDC towards TDC and the inlet valve closes. The charge air that was trapped in the cylinder is now compressed and a rapid rise of pressure and temperature occurs. Fuel is injected towards the end of the compression process.

**Power/Expansion Process:** Combustion of the air charge and the injected fuel creates irreversible expansion of the gases within the cylinder forcing the piston down the cylinder with excess pressure. Towards the end of this stroke, the exhaust valve opens and the products of combustion begin to flow out of the cylinder.
Exhaust Process: The exhaust valve remains open whilst the piston travels up from BDC expelling the remaining gases from the combustion chamber. Towards the end of the exhaust stroke, the inlet and exhaust valves are typically open simultaneously for a certain period. During this time the remaining exhaust products are scavenged from the cylinder by the incoming charge air.

The four-stroke cycle is illustrated in Figure 2.1, it is shown in mechanical terms as the piston moves up and down the cylinder.

Figure 2.1 Annotation of 4-stroke operating cycle (www.tpub.com 2005)

The four-stroke cycle can be represented on a pressure-volume diagram as shown in Figures 2.2 and 2.3. The theoretical CI engine is represented by a dual-cycle graph (Joel 1992) which is more closely representative of a CI engine cycle since heat is partly released at constant volume and partly released at constant pressure.
In contrast to the idealised dual cycle, a typical operating engine pressure-volume graph shows that induction, combustion and exhaust processes differ as the properties of the working fluid (air) change. For example, a pumping loop can be seen in Figure 2.3, which is typically small in CI engines due to no throttling of the inlet charge air and also the compression and expansion processes being non adiabatic.

This is a basic description of how a CI engine operates, there are many parameters involved for effective running and there have been many advances over the past 100 years. Specific designs of engines vary according to their end use requirement. Applications include automobile, locomotive, power generation and marine. The main variations of the traditional design are the combustion chamber (direct injection and indirect injection), air inlet conditions (normally aspirated, turbocharged and supercharged), fuelling pressures, timing and in-cylinder flows. The in-cylinder charge air flow is the primary area of investigation of this research. In this present work the engine used was a Perkins Trailblazer CI four-stroke single cylinder test engine with a swept volume of 1 litre. It was of the direct injection (DI) type where the combustion chamber was machined within the piston bowl.
2.3 Compression Ratio

As previously described, ignition of fuel in CI engines is achieved by the approximate adiabatic compression of the charge air, which increases the temperature of the charge air beyond the minimum ignition temperature of the injected fuel.

The compression of the charge air is fundamentally affected by the compression ratio, \( r_c \), which is defined as

\[
r_c = \frac{\text{max. cylinder volume}}{\text{min. cylinder volume}} = \frac{V_d + V_c}{V_c}
\]  

(2.1)

where \( V_d \) is the displaced volume in the cylinder

\( V_c \) is the cylinder clearance volume.

With reference to Figure 2.2, the compression and power strokes of this idealised cycle are adiabatic and reversible. Hence the efficiency can be calculated from the constant pressure and constant volume processes. The theoretical indicated fuel efficiency of a dual cycle engine is related to the compression ratio by

\[
\eta_{f,d} = 1 - \frac{1}{r_c} \left[ \frac{r_p \alpha^r - 1}{(r_p - 1) + \gamma r_p (\alpha - 1)} \right]
\]  

(2.2)

where \( \gamma \) is the ratio of gas specific heat capacities, defined as \( \gamma = \frac{C_p}{C_v} \)

\( r_p \) is the pressure ratio during constant volume heat addition \( (r_p = \frac{P_1}{P_2}) \)

\( \alpha \) is the volume ratio during constant pressure heat addition \( (\alpha = \frac{V_1}{V_3}) \)

Figure 2.4 shows the effect of indicated fuel efficiency for varying values of compression ratio for constant-volume, constant-pressure and limited-pressure cycles.
The numbers on the figure are representative of $\frac{P_2}{P_1}$ for the limited pressure cycle.

![Figure 2.4](image)

**Figure 2.4** Fuel conversion efficiency as a function of compression ratio, for constant-volume, constant-pressure and limited-pressure ideal gas cycles. (Heywood 1988)

Typical CI engines have a compression ratio ($r_c$) in the region of 15-20, achieving a minimum ignition temperature in excess of 600°C.

### 2.4 In-cylinder flows and fuel injection

CI engines have two main fundamental requirements for successful combustion; these are control of the fuel injection process and the charge air motion into and within the cylinder. The DI engine used in this study employed various types of inlet geometry (helical and directed ports) and bowl-in-piston designs. There are multiple types of piston bowl shapes available. For this project, both shallow and deep bowls were investigated to compare different piston designs and their affect upon air motion within the combustion chamber. The following sections consider the generation of charge air motion, the fuel injection event and different piston bowls. These factors
are shown to have a considerable affect on the emissions and fuel conversion efficiency of an engine.

2.4.1 Port generated induction swirl

Swirl is defined as the rotation of the intake air about the cylinder axis and is achieved by inlet port shape and orientation. The air enters the cylinder with an angular momentum, which persists through to TDC on the compression stroke. The importance of swirl is to promote rapid mixing between the inducted air charge and the injected fuel, aiding combustion, turbulent burning velocities and emission control. The preferred method of achieving swirl is via carefully designed inlet ports. There are two main types of port to create swirl during induction, these are:

i. **Directed:** The air is discharged tangentially from the valve towards the cylinder wall, where it is deflected sideways and downwards in a swirling motion. The port passage is essentially straight (Figure 2.5) which can create a flow restriction within the cylinder head due to space requirements, thus resulting in relatively low discharge coefficients.

ii. **Helical:** In this method the air charge produces swirl about the valve axis prior to entering the cylinder through specially designed inlet ports (Figure 2.5). These inlet ports typically have a higher coefficient of discharge for equivalent levels of swirl since the entire area of the valve is used, leading to an increased volumetric efficiency for the general case (Heywood 1988).
The induction stroke is of high importance to achieving good combustion during the power stroke. The angular momentum of the charge air introduced through induction decays, due to frictional losses, during the induction and compression strokes (approximately 30% (Heywood 1988)), but remains sufficiently high up until TDC where it develops a mean flow pattern, which is assumed to approximate solid body rotation.

Figure 2.6 compares steady-flow test rig results for different types of inlet, showing the swirl rig coefficient against a normalised valve lift. This rig swirl coefficient increased with the valve lift, which is indicative of the increasing impact of the port shape and the decreasing impact of the flow restriction between valve and cylinder. The normalised swirl numbers were as follows:

i. Plain directed port normalised swirl of 2.5
ii. Helical shallow ramp port normalised swirl of 2.9
iii. Helical steep ramp port normalised swirl of 2.6
As shown in Figure 2.6, helical ports normally have an increased angular momentum at medium valve lifts compared to directed ports, however, as the valve lift approaches maximum, the directed inlet shows an increased swirl. This result is confirmed by Williams and Tindal (1975), as shown in Figure 2.7.
Williams and Tindal also found the air charge to represent solid body rotation in the outer regions (approximately 32 – 92% of cylinder radius) of the flow, but the central core (0 – 32% of cylinder radius) to be a disorganised flow. Their work went on to consider the effect of swirl rate on engine performance and emissions. A single cylinder research engine was fitted with a masked inlet valve (similar to that illustrated in Figure 2.8 (b)) to achieve a variable swirl rate. The engine was a CI DI bowl-in-piston type engine, running at compression ratio of 13.6:1, with a two-hole injector. It was run at constant speed (1600 rpm) with constant fuelling. The angle of the masked valve was altered during running and the smoke, NOx and bmep (brake mean effective pressure) were recorded. Results from the steady-flow testing of the cylinder head were superimposed onto the emissions and bmep data and showed a strong link between swirl rate, engine performance and emissions. This can be seen in Figure 2.9.

![Figure 2.8](image)

**Figure 2.8** Shrouded inlet valve and masked cylinder head approaches for producing net in-cylinder angular momentum (Heywood 1988)

Figure 2.9 shows the correlation between emissions and swirl, where NOx increases with swirl rate, and smoke decreases with increased swirl rate. There is a deviation in the bmep results with a third peak occurring at approximately 160° of the masked valve orientation. Williams and Tindal associated this increase in bmep to be due to a reversal in the swirl direction due to the mask opposing the incoming flow from the directed port.
2.4.2 Fuel injection event

The main focus on the work presented in this thesis is air charge motion. Fuel injection was not employed in this work since the increased pressure and temperature rates due to the heat release of fuel were too high for the optical engine used. However, since the air charge motion and the fuel injection event are the two main factors that affect emissions, a basic introduction of fuel injection will be given.

The aim of the fuel injection event is to provide the compressed charge air with atomised fuel. The accuracy of this fuel delivery is fundamental to the emissions produced by the engine. Typical CI engines have fuel injection pressures in the
region of 200 – 2000 bar depending on type of engine and fuel pump delivery system. Such high pressures are required to achieve high fuel jet velocities, so that:

i. There is good charge utilisation ensuring that the injected fuel has enough velocity to cross the combustion chamber in the correct time, thereby guaranteeing improved mixing with the charge air.

ii. Good fuel atomisation occurs, whereby the injected fuel is reduced to small droplets (thus increasing the surface area to volume ratio) and therefore facilitating rapid evaporation, heat transfer and combustion.

Determining the correct type of injection is dependent upon the final application of the engine, hence there is not an ideal fuel injection type. These properties of the fuel injection event are reliant on many parameters and much research into the area of fuel injection is ongoing. The fundamental processes of the main spray parameters can be sub-divided into four main areas, as shown in Figure 2.10:

i. Spray tip penetration (charge utilisation)
ii. Spray angle (charge utilisation)
iii. Break-up length (fuel atomisation)
iv. Droplet size distribution (fuel atomisation)

Figure 2.10 Schematic of CI DI engine fuel-spray defining major parameters (Heywood 1988)
In Figure 2.10, as the fuel liquid exits the nozzle of the injector it becomes turbulent and spreads out facilitating mixture with the charge air. Initial jet velocities at the nozzle exit are seen to be greater than \(100 \text{ m.s}^{-1}\). The outer surface of the jet breaks into smaller droplets in the order of \(10 \text{ } \mu\text{m}\) diameter. The liquid column exiting the nozzle disintegrates over a finite length, known as the break up length, into different droplet sizes. In modern multi-valve CI DI engines, the injector is usually mounted central to the cylinder axis. As the radius of the piston bowl increases, the mass of air that the fuel spray interacts with increases, the spray diverges, the width of spray angle increases and the velocity of the fuel decreases (Heywood 1988).

The air swirl is used to promote the fuel-air mixing rates. Figure 2.11 shows an example of the interaction of fuel spray and swirl motion in a DI engine. The interaction of the fuel air becomes much more complex with the addition of swirl. The spray will entrain the air to a much greater degree with swirl and bends the spray in the direction of the swirl. For similar injection parameters, the spray will penetrate less for an engine with swirl than without. The design aim of the fuel injection event is to limit the amount of liquid fuel that impinges on the piston bowl wall, ensuring complete mixing of the fuel and air charge.

![Figure 2.11 Schematic of fuel spray injected radially from chamber axis into swirling flow (Heywood 1988)](image)

With more stringent emissions legislation, the pressure of the fuel injection event has substantially increased and nozzle diameters have decreased, resulting in better atomised fuel distribution.
2.4.3 Piston bowl design

The piston bowl design increases the amount of air swirl during the compression process. It is based on the conservation of angular momentum, whereby the swirling charge air is forced into a smaller radius as the piston approaches TDC. This conservation of angular momentum will be discussed fully in Chapter 5. Many different designs of the bowl-in-piston combustion chamber exist, dependent on the application, engine size and inlet port geometry.

Figure 2.12 presents the most common type of combustion chamber designs. The four piston bowls shown in Figure 2.12 are claimed to give similar emissions and fuel conversion efficiency, despite the differences in the bowl shape. Stone (1999) suggests that the piston bowl shape is less critical than the careful design of the charge air motion and the fuel injection system.

![Figure 2.12 Different types of DI combustion chambers (Stone 1999)]

(a) Hemispherical combustion chamber
(b) Shallow bowl combustion chamber
(c) Shallow toroidal combustion chamber ($d/h \approx 4$)
(d) Deep toroidal combustion chamber ($d/h \approx 2$)
In this current research, two types of piston bowl were studied; shallow bowl and deep bowl designs. Since the squish and swirl interaction of the flow is dependent on the piston bowl design, it is difficult to predict the flow regime that will occur at TDC. The toroidal design of the piston bowl creates a different flow pattern in the piston bowl at TDC. Work by Arcoumanis et al. (1983) considered the shape of the piston bowl wall for a flat sided piston bowl and a re-entrant (toroidal) design. This work showed that the swirl profile at TDC of the compression stroke is more dependent on the piston bowl geometry rather than on the intake swirl field. At the base of the piston it was assumed that solid body rotation continues throughout the compression stroke. Figure 2.13 shows the flowfields that were created when using a straight sided piston bowl and a re-entrant type piston bowl for the case of a swirl inlet.

![Flowfields](image)

(a) Straight sided  (b) Re-entrant

Figure 2.13 Sketches of flowfields at TDC for straight sided and re-entrant piston bowls (Arcoumanis et al. 1983)

The flow interaction of squish and swirl markedly change with the re-entrant design. In a straight sided piston with swirl (Figure 2.13 (a)), the charge air enters the bowl and flows down to the base of the bowl then inward and upward in a toroidal motion. In the re-entrant design (Figure 2.13 (b)), the charge air moves down to the base of the bowl and outwards into the undercut region of the bowl. The flow then splits, creating a recirculation area in the undercut region and a stream flowing inwards along the bowl base. The re-entrant design was observed to increase the swirl of the air charge at TDC. It was also found that turbulence intensity increased whilst cyclic fluctuations in the mean velocity were reduced.
Further investigation by Corcione et al. (1991) used the laser Doppler velocimetry (LDV) optical measurement technique to identify the varying emissions, turbulence and tangential velocities for a straight sided piston bowl and a re-entrant piston bowl. Both piston bowls had the same volume and squish area as can be seen in Figure 2.14.

![Figure 2.14 Piston bowl geometries (a) straight sided (b) re-entrant (Corcione et al. 1991)](image)

Two engines were used in this work, one that was used for emission testing the different piston bowls, the second was altered for optical access for motoring the engine to carry out LDV testing. Multiple LDV tests were carried out for different radii and the results averaged to present the data in velocity bandings across the piston bowl. The findings of this work showed how the re-entrant bowl increased the tangential velocity by up to 40% and the turbulence intensity by up to 25%. Figure 2.15 shows the velocity banding across the cross-sectional area of the respective piston bowls at 5° crank angle before TDC (BTDC).

<table>
<thead>
<tr>
<th>Tangential velocity</th>
<th>Turbulence intensity</th>
</tr>
</thead>
<tbody>
<tr>
<td>a) 0.0 - 5.0 m/s</td>
<td>a) 0.0 - 1.0 m/s</td>
</tr>
<tr>
<td>b) 5.0 - 9.0 m/s</td>
<td>b) 1.0 - 1.4 m/s</td>
</tr>
<tr>
<td>c) 9.0 - 11. m/s</td>
<td>c) 1.4 - 1.7 m/s</td>
</tr>
<tr>
<td>d) 11. - 13. m/s</td>
<td>d) 1.7 - 1.9 m/s</td>
</tr>
<tr>
<td>e) 13. - 18. m/s</td>
<td>e) 1.9 - 2.3 m/s</td>
</tr>
</tbody>
</table>

![Figure 2.15 Spatial distribution inside combustion bowls for tangential velocity and turbulence intensity at 1000 rpm (Corcione et al. 1991)](image)
Corcione et al. concluded that the re-entrant bowl aided the creation of higher velocities and turbulence intensity when compared to straight sided piston bowls. Engine smoke emission was reduced by nearly 50% by careful choice of the nozzle which injected fuel into the increased turbulence and tangential velocity regions.

2.4.4 Squish motion

During the final stages of the compression stroke, an additional flow affects the swirl profile of the intake charge. As the piston crown and cylinder head approach each other, an inward radial motion of the swirling charge occurs into the bowl of the piston, commonly known as squish, which is illustrated schematically in Figure 2.16.

The squish theory, which is only applicable to the outer region between the bowl periphery and cylinder, assumes that the flow is axisymmetric, the crevice volumes negligible and the air density is uniform with the rest of the combustion chamber. Heat transfer, compressibility effects, friction and leakage past the piston rings are also all neglected. The squish velocity for bowl-in-piston designs can be calculated as follows (Heywood 1988).
\[ v_{sq} = \frac{D_B}{4z_p} \left[ \left( \frac{B}{D_B} \right)^2 - 1 \right] \frac{V_B}{A_c z_p + V_B} S_p \]  

(2.3)

where \( V_B \) is the volume in the piston bowl

\( A_c \) is the cross-sectional area of the cylinder

\( z_p \) is the distance between the piston crown and the cylinder head

\( D_B \) is the diameter of bowl

\( B \) is the diameter of the cylinder bore

\( S_p \) is the instantaneous piston speed, calculated as

\[ S_p = \frac{\pi}{2} \sin \theta \left[ 1 + \frac{\cos \theta}{(s_{ratio}^2 \sin^2 \theta)^{1/2}} \right] \bar{S}_p \]  

(2.4)

where \( \theta \) is the crank angle in radians

\( \bar{S}_p \) is the mean piston speed, calculated as \( \bar{S}_p = 2s \cdot n \) (m.s\(^{-1}\))

where \( s \) is the stroke of the engine (m)

\( n \) is the crankshaft speed of the engine (rads.s\(^{-1}\))

\( s_{ratio} \) is the ratio of connecting rod length to crank radius (approx. 3.5).

The model results are illustrated in Figure 2.17 by normalising the squish velocity by the mean piston speed \( \bar{S}_p \) but varying the ratio of \( D_B/B \) and \( z_p \). It was shown that the maximum squish value occurred at approximately 10° BTDC (Horlock and Winterbone 1986). After TDC, the reverse effect occurs and the gas flows out of the bowl into the clearance volume. In a motored engine this effect is equal to \( v_{sq} \).
Figure 2.17 Theoretical squish velocity divided by mean piston speed for bowl-in-piston chambers, for different $D_B/B$ and $c/L$ (clearance height/stroke). $B/L = 0.914$, $V_B/V_d = 0.056$, connecting rod length/crank radius $= 3.76$ (Heywood 1988)

Gas inertia and friction effects were shown to be insignificant, but the effects of blow-by and heat transfer are significant. The decrement of squish velocity due to leakage ($\Delta v_L$) is proportional to the mean piston speed and a dimensionless leakage number, given by (Heywood 1988)

$$N_L = A_{E.L} \frac{\sqrt{\gamma R T_{WC}}}{N V_d}$$  \hspace{1cm} (2.5)

where
- $A_{E.L}$ is the effective leakage area
- $T_{WC}$ is the temperature of cylinder gases at inlet closing
- $N$ is the crankshaft speed
- $V_d$ is the displaced volume of the cylinder
- $\gamma$ is the gas specific heat capacity ratio
- $R$ is the gas constant.
Figure 2.18 Values of squish velocity decrement due to leakage $\Delta v_L$ and heat transfer $\Delta v_H$, normalised by the ideal squish velocity, as a function of crank angle (Heywood 1988)

In Figure 2.18 the leakage was modelled as a choked flow through the effective leakage area and it can be seen that $\Delta v_L / v_{sq}$ are minimal for normal gas leakage values. Additionally, values of the decrement of squish velocity due to heat transfer ($\Delta v_H / v_{sq}$) were derived using standard engine heat transfer correlations. In both of these graphs, the decrement due to squish velocity greatly increased after the maximum squish velocity ($\approx 10^\circ$ BTDC) had occurred and the piston approached TDC.

The effect of squish on the turbulence levels within the cylinder was explored by Arcoumanis et al. (1983). It was found that squish had a negligible effect on turbulence; however, the interaction of swirl with squish did have a significant effect upon the axial flow resulting in a counter-rotating vortex. Counteracting fields of centrifugal forces associated with swirl and squish caused this effect. Therefore, the balance between the two determined the extent of the inward penetrating jet.
2.4.5 Turbulence

The understanding of turbulence is important to the quantification of the flow processes present in the cylinder of an engine. This section is a basic introduction to engine turbulence and its effects.

Turbulence is always present within an operating engine and is known to be important for achieving high combustion rates. It has been an area of much study as the understanding of turbulent flowfields and processes are essential to the optimisation of combustion. Turbulent flows allow a much greater rate of mixing and transfer due to molecular diffusion. Turbulent diffusion is a result of local fluctuations within the flowfield and it increases the rate of momentum, heat transfer and mass transfer thus affecting combustion rates.

Examination and numerical simulation of turbulent flow is a complex task even for the simplest of geometries and can be subject to significant errors. An example of this is during an engine’s intake stroke when the incoming air has to pass through the opening intake valve, creating a jet-like flow. This flow then interacts with flows created by the piston motion as well as the cylinder head and wall (Turns 1996). Therefore, due to the irregularity of turbulent flows, statistical methods are employed to define the flowfield. These statistical methods can be divided into mean velocity, fluctuating velocity about the mean, as well as different length and time scales.

In a steady turbulent flow, the flow can be described as

\[ U(t) = \bar{U} + u(t) \]  \hspace{1cm} (2.6)

where
\begin{align*}
U(t) & \text{ is the instantaneous local fluid velocity} \\
\bar{U} & \text{ is the mean velocity for the steady turbulent flow over a time average of } U(t) \\
u(t) & \text{ is the fluctuating velocity component}
\end{align*}
In an engine cylinder the charge air motion can be further represented relative to the crank angle \((\theta)\) and cycle \((j)\) by

\[
U(\theta, j) = \bar{U}(\theta, j) + u(\theta, j).
\]  

(2.7)

This equation does not consider that the bulk flow varies significantly from cycle to cycle, as well as turbulent fluctuations about the specific cycle’s mean flow. These cyclic velocity fluctuations, \(\bar{U}_{CV}(\theta, j)\), are the difference between the mean velocity in a particular cycle and the ensemble average of mean velocity taken over many cycles. Hence

\[
\bar{U}_{CV}(\theta, j) = \bar{U}(\theta, j) - \bar{U}_{EA}(\theta)
\]  

(2.8)

where \(\bar{U}_{EA}(\theta)\) is the ensemble averaged velocity, i.e. the averaged values over a defined number of cycles at a specific crank angle. This can be represented as

\[
\bar{U}_{EA}(\theta) = \frac{1}{N_C} \sum_{j=1}^{N_C} U(\theta, j)
\]  

(2.9)

where \(N_C\) is the number of cycles for which data was taken. Substituting the cyclic variability into Equation (2.7) gives

\[
U(\theta, j) = \bar{U}_{EA}(\theta) + \bar{U}_{CV}(\theta, j) + u(\theta, j).
\]  

(2.10)

Figure 2.19 shows these different velocity component terms through a cycle using the equation above. It can be seen that the instantaneous velocity component was split into the ensemble averaged component, cycle mean velocity and a randomly fluctuating velocity about the mean velocity, defined in relation to individual cycle’s mean velocity.
These turbulent fluctuations about the individual cycle mean velocity are normally shown by their magnitude, known as the ‘turbulence intensity’ and represented by $u'$. This turbulence intensity is the root mean square (RMS) of the velocity fluctuation and can be calculated as

$$u' = \lim_{t \to \infty} \left( \frac{1}{\tau} \int_{t_0}^{t_0 + \tau} (U^2 - \bar{U}^2) dt \right)^{\frac{1}{2}}. \quad (2.11)$$

This turbulent fluctuation will be used in defining the flow in Chapter 7. The turbulence and cyclic variability can be separated through complex means of filtering. The following section now considers cyclic variability in engines.

### 2.4.6 Cyclic variability

It is commonly known that internal combustion engines are subject to variations in cylinder pressure between different cycles. This cycle-to-cycle variation is associated with the variations in fluid dynamics and combustion processes. With respect to engines, the combustion process is reliant on a suitable fuel air mixture being present in the cylinder at the appropriate point of ignition, and that the conditions of the in-cylinder flow prior to ignition are not repeatable. Some other causes of cyclic variation can be attributed to poor air-fuel mixing and low-frequency variations of the
in-cylinder turbulent flowfield. Enotiadis et al. (1990) investigated these cyclic flow variations in a motored engine using LDV (this measurement technique will be discussed in Chapter 3) in a transparent engine. Measurements were taken over a large number of cycles (700 – 1000) and the turbulence was ensemble averaged. The swirl velocity was measured about the cylinder axis and the datasets were ensemble averaged to give a smooth representation of swirl velocity within the engine, using high and low filters to remove spurious data. This filtering lead to underestimation of turbulence quantities in the engine.

Zhang et al. (1995) utilised PIV experiments to measure these cyclic variations in a CI DI engine. The optical engine used mirrors to obtain images of the flow from below the piston in the same way that Haste (2000) gained optical access to an SI engine, commonly referred to as a Bowditch piston. Zhang et al. used a simple flat bottomed bowl-in-piston combustion chamber, compression ratio of 13.5:1, 2-valve single cylinder engine that ran at a 500 rpm crankshaft speed. The cylinder head was interchangeable for a swirl and a non-swirl inlet to the engine. From the PIV data obtained, filtering was carried out and it was shown how the swirl centre changed from cycle to cycle but began from a similar pattern each time. The evaluation of flow fluctuations (turbulence intensity) was determined by subtracting the ensemble averaged velocity vectors from the cycle-resolved vectors. Towers and Towers (2001) investigated the cyclic variability with reference to a gasoline direct injection (GDI) engine. The engine was fuel injected and the time resolved PIV measurements were taken late in the compression stroke to obtain results for the mixing of fuel and air (100°- 30° BTDC). The camera used was only a 128 x 128 pixels, with a high-speed image rate (13.5 kHz); the engine was motored at 2000 rpm. The main area of interest to this paper was the processing of PIV data for cyclic variability. From the principle that cyclic variability could be distinguished from turbulence, it considered that flows in the region of 5 - 20 m.s⁻¹ were bulk flow motion in the engine and that the small eddies in the flow (eddies of radius < 2.5 mm) were representative of turbulence. As in other work, an ensemble average was calculated from a number cycles which is then subtracted from the velocities measured in a single cycle that had been low pass filtered and resolved. An average cyclic variability was calculated from the absolute cyclic variability data for a specific cycle.
St. Hill et al. (2000) studied cyclic variability using LDV techniques in dual intake ported SI engine. A Ford Zetec 1.8 litre four-cylinder head was mounted onto a Pyrex glass cylinder providing necessary optical access. As mentioned in Towers and Towers work, the ensemble average was determined over a number of cycles. The ensemble average was calculated as per Equation (2.9), and the turbulence level was considered in two parts as per Equation (2.10). The fluctuating velocity was filtered using high and low pass filters and related to the turbulence intensity. These results were compared with data that was obtained from the same cycle at the inlet jet, and normalised with mean piston speed. These results suggest that cyclic variation is over estimated by up to 100%. Furthermore, in order to obtain a true ensemble average, results had to be taken for 12000-18000 cycles to achieve 5% mean piston speed accuracy. Less than 12000 cycles could have an inaccuracy of mean velocity of up to 20% mean piston speed.

2.5 Summary

This chapter has introduced the fundamental areas of in-cylinder flows in CI DI engines. The importance of in-cylinder flows with respect to combustion has been identified along with the parameters that affect them. The following chapter will introduce measurement techniques that have been applied to in-cylinder flows. Particular emphasis of the PIV technique is given, which was the chosen method of measurement for this current study. Chapter 5 will introduce the quantification of swirl and give a detailed description of testing and calculation.
Chapter 3  Measurement Techniques

3.1  Introduction

The study of fluid flow in engine cylinders has been of significant interest to engineers in the past few decades. There have been various experimental techniques used to characterise the in-cylinder flow, turbulence and bulk motion in reciprocating IC engines. The most common four types of flow characterisation are:

i. Hot Wire Anemometry (HWA)
ii. Laser Doppler Anemometry/Velocimetry (LDA/LDV)
iii. Laser Induced Fluorescence (LIF)
iv. Particle Image Velocimetry (PIV).

All four techniques will be discussed in this chapter. Other techniques have also been developed aimed at quantifying the in-cylinder flows with different degrees of success. The continuing difficulty with making these measurements is that the measurement equipment should not affect the flow. For this reason, non-obtrusive measurement techniques, such as optical measurements, are favoured to obtrusive techniques that involve physical probes (i.e. hot wire anemometry).

With optical measurement techniques, it is necessary to introduce a method for recording the flowfield motion. This is typically achieved by introducing small flow tracing particles into the flow which infer the motion of the fluid, commonly referred to as ‘seeding’, and their motion recorded. The flow following capability of the seeding is dependent on the velocities experienced in the measured fluid and its properties with respect to those of the seeding particles. A further requirement of the seeding, necessary for most optical techniques, is the light scattering behaviour of the seeding medium. It is necessary that the seed disperses enough light so that it can be recorded on imaging equipment.
3.2 Hot Wire Anemometry (HWA)

The hot wire anemometry technique is considered to be a relatively old measurement method but is still used in some applications. It is an intrusive measurement technique that provides fluid flow data at a single point of interest. The principle of HWA is dependent on the rate of heat loss from a hot surface to a cooler gas stream and is directly affected by the velocity of this gas stream. Factors that affect this type of measurement include the properties of the fluid, the temperature differential between the gas stream and the hot surface and the nature of the flowfield itself. In most instances, the hot surface is a thin wire (of low thermal inertia) which is usually maintained at a constant temperature by use of varying electrical current. This electrical current is related, for a given gas stream temperature to the gas stream velocity normal to the wire axis. The problem with this type of measurement comes when quantifying the direction of flow, since the readings obtained are only dependent on the magnitude of the flow. HWA was used successfully in measuring turbulent flows, but a non-linear relationship exists between the electrical input and the magnitude of the gas velocity, consequentially careful calibration is required.

The HWA technique has been used fairly successfully in engine studies due to its following attributes:

i. Continuous signal
ii. Good frequency response
iii. High signal to noise ratio.

However, the disadvantages are significant, including

i. Obtrusive as it has to be placed in direct contact with the flow
ii. Unable to withstand combustion temperatures
iii. Calibration is required over the entire range of temperatures and pressures experienced in the engine
iv. Varying gas temperature within the engine, which continually alters the gas stream properties
v. Non-directional data; only magnitude of flow velocity
vi. Upper limit on turbulence intensity that can be measured (which is often high in IC engines).
The disadvantages associated with HWA have meant that data obtained using the technique can be subject to large errors when coupled with the many different parameters in an IC engine (Derham 1972). These errors are most prominent in the compression stroke and power stroke since these have the largest temperature gradients.

HWA, as a method for engine flow velocity measurement, was used extensively from the 1960's to 1970's, as there were few other suitable types of flow measuring techniques available. In more recent times this has changed with the onset of digital computers and optical measurement techniques. These are now discussed.

3.3 Laser Doppler Anemometry / Velocimetry (LDA/LDV)

LDV is an optical measurement technique that utilises at least two coherent laser beams to gain velocity data of a flow at a single point of interest (Stone 1999). The principle of its operation is that the two coherent laser beams are passed through a common lens and intersect in the flow at the focal point. This is shown in Figure 3.1.

![Figure 3.1 Optical arrangement for LDV showing the interference fringe pattern collection of the light scattered by the particles in the forward scatter mode, and a typical output from the photomultiplier (Stone 1999)](image-url)
The crossing of the beams creates an interference fringe pattern in the measuring volume, with the fringe lines running perpendicular to the axis of the lens. Seeding is introduced into the flow, which is assumed to follow faithfully. When seeding particles pass through the fringe pattern, light is scattered (known as a Doppler burst), the frequency of which is proportional to the particle velocity perpendicular to the fringes. A photomultiplier tube receives the scattered light and the measured frequency of the scattered light allows for direct determination of the particle’s velocity without the need for calibration. It is possible to have multiple systems so as to be able to measure multiple areas of interest or multi-component measurements at a single area of interest.

The advantages of this technique are:

i. Non-intrusive to flow which is not affected by the properties of the flow
ii. Linear measurement system and is independent of the flow conditions
iii. Directionally sensitive, providing a full velocity vector at the point of interest
iv. High temporal resolution.

The method, however, does suffer some drawbacks and these include:

i. Optical access to the flow is required in two perpendicular locations to obtain 3-D velocity vectors
ii. Intermittent nature of signal, due to seeding concentration in the flow
iii. Only a small region of flow can be measured at one time and therefore does not give a full field of flow measurement (poor spatial resolution).

This last disadvantage is the major drawback for this optical measurement technique. To enable a dataset of the in-cylinder flow, multiple measurements have to be taken at different points and at different times, allowing for only averaged data built up from different tests.

The majority of work using LDV in IC engines was in SI type engines due to the lower pressures within the engine, the slower air motion, and the flat piston tops, which meant measurements were easier to make. Market influence also favours SI engines, being of historically greater interest in the high production volume capacity.
automobile sector. As previously mentioned, as LDV is an optical technique, optical access is necessary in the combustion chamber of the engine. Bates (1988) for example, carried out research into optical access of the combustion chamber for a skip fired and motored SI engine arrangement with a compression ratio of 9:1. A number of transparent materials were considered for use within the optical engine including plastics, glasses and crystals. The major drawback encountered with plastics and glasses was their low thermal conductivities, which did not allow the material to be cooled significantly during operation. Another suggested material from this research was single crystal sapphire, which can be grown to sizes suitable for experimentation (diameters in excess of 250 mm). Single crystal sapphire also exhibit excellent optical qualities whilst maintaining strength comparable to steel. However, they are extremely expensive, costing tens of thousands of pounds each.

LDV was extensively used in the investigation of engine inlet ports due to it accurately measuring velocity vectors in a small area of interest. Tindal et al. (1988) investigated the inlet port effects between helical and directed ports in a transparent engine. In this research, an acrylic liner was employed (without a moving piston arrangement) and the working fluid was replaced with a mixture of oil, turpentine and tetraline. This gave the same refractive index as that of the transparent acrylic liner. LDV measurements were carried out with steady-flow through the inlet port and cylinder to show a number of velocity vectors through the system. Particular interest was paid to the valve exit region. These tests were carried out with varying lift of the inlet valve with the results presented as mean velocities and turbulence intensities. The intake structures of flow over different types of inlet port were observed, but this was of limited value in understanding overall bulk flow motion through compression (since piston motion was not considered), and the breakdown of turbulence (since the working fluid used was not air).

Furuno et al. (1990) used an LDV system to measure the effects of swirl inclination angle in a motored non-firing SI engine. The horizontal swirl motion was generated by the use of a helical inlet port and the inclination angle of the swirl was altered by the use of a shrouded inlet valve. This inclination was varied and flow tests were carried out to determine the approximate components of velocity. The research was primarily concerned with the effect of swirl inclination on turbulence and turbulence.
intensity. In work by Hong and Chen (1997) and Tabaczynski (1983), direct measurements of the integral length scales were taken and compared to calculated length scales. In order to instantaneously resolve these length scales in the engine, at least two spatially separated velocity measurements were required. Again, these results were restricted to a small region of interest (approximately 1mm x 1mm). A variation to this was researched by Dimopoulos and Boulouchos (1997), where the 3-D velocity measurements were taken by employing three beam pairs in the optical engine and two fibre optic probes. A number of locations were measured within the cylinder (ranging from 4.5 mm – 10.5 mm below the cylinder head flame face) at different speeds, inlet swirl (provided by a shroud valve) and different points in the engines cycle. The engine used a flat top piston. It was found that in high swirl conditions the tangential velocity profiles were close to solid body rotation.

Spicher et al. (1987) considered LDV experiments in a firing diesel engine. In this work, the parameters that were varied were the swirl ratio, the squish height and injection timing with the engine running at 1100 rpm. Optical fibre probes were introduced to the cylinder head, liner, and the piston. These were then used to detect the flame propagation within the cylinder, whilst also measuring the pressure traces within the engine. A 2-D LDV system was employed and measured the velocity vectors 15 mm above the bottom of the combustion bowl under motored conditions. Optical access was gained through a quartz window mounted in the cylinder head of the engine. The flow structures observed within the bowl at TDC were considered to be solid body rotation and the increase in swirl was seen to alter the ignition delay. It was also observed that the reduction in squish height lessened the ignition delay, retarded the flame spread into the squish area and showed faster flame propagation through the combustion bowl.

3.4 Laser Induced Fluorescence (LIF)

This optical technique provides molecular specific detection of chemical species in flows or flames. The ranges of areas where LIF can be employed in the measuring fluid are:
i. Species concentration
ii. Temperature
iii. Velocity
iv. Pressure
v. Density.

In LIF, a high energy pulsed laser with tuneable output wavelength is tuned to a specific fluorescence transition relative to the molecules of interest. The laser sheet is directed into the flow, where the molecules of interest absorb the energy from the laser, which artificially increases the energy level of the molecule. Once the molecules have been excited to this higher energy state, the laser sheet is removed and the additional energy introduced to the molecules is released as a fluorescent light. Charged coupled device (CCD) cameras are employed to record the images emitted from the fluorescence. These images are usually filtered to remove external light sources from combustion and processed via a computer. A basic arrangement of this technique is illustrated in Figure 3.2.

\begin{figure}[h]
\centering
\includegraphics[width=0.8\textwidth]{lif_optical_arrangement.png}
\caption{LIF optical arrangement}
\end{figure}

Work by Heinze and Baritaud (1992) utilised LIF in order to visualise the fuel distribution in a SI engine. Iso-octane fuel was employed in the fired optical engine. Since the wavelength required to excite iso-octane was so short, it was necessary to introduce dopants into the fuel, which would fluoresce. This main aim of this
research was concerned with making the LIF technique feasible for use in an SI engine. It was shown that a linear relationship existed between the amount of light intensity emitted and the fuel injected. Two different valve orientations were used to alter the flow. In one instance a tumble flow was observed whilst the other showed a swirling flow. Images were taken at the point of the mixing between the fuel and air occurred. In the swirl case there were large and small scale inhomogeneties of the fuel air, conversely with the tumble flow only small scale inhomogeneties were observed. Also shown are the different inhomogeneties due to cyclic variations within the engine. Further investigation into the in-cylinder fuel mixing in a SI engine was carried out by Akihama et al. (1999). This research was similarly primarily concerned with the fuel and introduction of suitable dopants to the mixture.

Further research investigating flames, by Lee et al. (1993), have shown how species of a particular type, in this case Nitric Oxide (NO) and the associated temperature of that particular species could be recorded. This type of research into flame temperature was applied to in-cylinder of a 2-stroke SI engine by Bauerle et al. (1994). The aim was to measure hot spots which created the conditions for knock in SI engines. Formaldehyde was mixed with the fuel and was used to obtain LIF results of areas of temperature-induced combustion in the engine.

Overall LIF is a very useful technique for concentration and temperature, but fully quantified results are difficult due to calibration challenges.

3.5 Particle Image Velocimetry (PIV)

PIV is an optical technique that can measure velocity vectors of a full field of flow typically in a two-dimensional (2-D) plane. There are a number of variations to the specific arrangements which will be discussed later. The technique essentially uses light sheets (produced typically by lasers) in a region of the flow, which were seeded with small particles. These particles are of such a size (typically 0.5 – 5 μm) which allows the flow of the fluid to be followed at an acceptable level of error. The scattered light from the particles in the light sheet is captured via a camera or other
image device over two frames with a known time separation. The two images are then correlated to obtain the displacement of the particles within a region of the flow. Each image is broken down into small interrogation regions and correlated with respect to each other to determine an average particle cluster displacement. Since the time interval between the two images is known, the respective velocities can be calculated for the particle clusters in the flow. The basic layout of a PIV experiment is illustrated in Figure 3.3.

![Figure 3.3](image)

Figure 3.3  Experimental arrangement for Particle Image Velocimetry in a wind tunnel (Raffel et al. 1998)

The major advantage of this technique over LDV is the ability to measure flow over extended regions and not just a single point of interest.

3.5.1 2-D PIV

Two dimensional PIV is the most common type of PIV utilised in flowfields. There is much research that has used 2-D PIV to study many types of flow, including IC engine flows (Haste 2000; Jarvis et al. 2005; Reeves et al. 1994; Hentschel 2002). The majority of this latter work was based around the charge motion of the intake air into the engine. In simple terms, 2-D PIV takes two successive images of the
particles in an illuminated 2-D planar slice of fluid flow. Each of the images is then divided into small interrogation regions and the clusters of particles in each region are correlated with the previous image. This correlation can be achieved using either 'cross-correlation' (two separate images with knowledge of the order of the frames) or 'auto-correlation' (both images are recorded to a single source).

The main difficulty when employing the PIV measurement technique is the equipment and seeding particle choice. Since small distances are being measured over short time separations, combined with optical limitations and flow complexities, it has often proven difficult to obtain good quality data. However, the PIV technique is an excellent tool for observing flowfield, and the pattern associated with it, provided the experiment is conducted correctly. The time separation between the two images is of importance to obtain good data (an estimation is essential so as to capture the particle movement within an interrogation region), hence it is necessary to have an estimate of the average flow within the fluid flow region. The basic set-up of PIV is shown in Figure 3.3. In the following sub-sections the major parts of the test equipment are discussed individually.

### 3.5.1.1 Laser light source

A laser is an appropriate light source to illuminate a planar sheet of particles when the image is recorded. The reason that a laser is used is that it can provide monochromatic light with high energy density in short pulses and can be manipulated by use of optics to create a good quality light sheet. In most arrangements, two lasers are used in conjunction for PIV to provide a double pulse laser sheet with a known time separation (Keane and Adrian 1990). The most commonly used laser (Hentschel et al. 2001; Nino et al. 1993; Reeves et al. 1994; Reeves et al. 1996; Reuss et al. 1989) is the Nd:YAG (Neodymium–Yttrium Aluminium Garnet), which is a solid state type laser with good mechanical and thermal properties and a high amplification. The pulse duration of laser is very short (in these instance 4 ns) which in effect freeze frames the particles in time.
### 3.5.1.2 Image recording

There are two main methods to record the image from PIV data, these are digitally through the use of a CCD (Charged Couple Device) camera (Hentschel et al. 2001; Marc et al. 1997; Willert and Gharib 1991), and photographic film (Nino et al. 1993; Reeves et al. 1994). The primary difference between these two recording methods is the spatial resolution, where CCD cameras offer images in the range of 512 x 512 to 2048 x 2048 pixel (picture element) quality, and typical Kodak Technical Pan photographic film which offers about 300 lines/mm, which equates approximately to 10500 x 7500 pixels (Kurada et al. 1993). A much higher resolution can be achieved with photographic film (into the region of several thousand lines/mm), however, the time taken for the light to expose the film makes it unpractical to use. The other major advantage, particularly with the rapid growth of computing power, is that digital means remove the steps of having to process photographic film and enables the datasets to be correlated in almost real time, or immediately after the experiment.

As the digital imaging technique has substantially less spatial resolution, the technique covers a smaller region of flow and builds a picture of the larger flow region by sub-sections. Recent experiments have trended towards digital imaging, due to the increase of digital CCD camera performance and decrease in cost.

### 3.5.1.3 Image analysis

Once the two images are recorded on either photographic film or computer hardware, the images are then analysed to determine the displacement of small particle clusters across the two images. These particle clusters are identified and interrogated with respect to the first and second images to determine the displacement of the particles. The images are sub-divided into interrogation regions in the order of 1 mm x 1 mm (dependent upon the particular type of flow). There are various ways of correlating these image regions based upon Fourier transforms of the particles to extract the mean velocity data for each sub-region. The main two types of correlation employed in PIV are auto-correlation and cross-correlation. In auto-correlation the interrogation region contains particles from the first and second light pulses and is correlated against itself.
In cross-correlation, the interrogation region is on two separate images for each interrogation region and the frames are labelled to allow the user to identify the first and second frame. These methods are illustrated in Figures 3.4 and 3.5.

![Diagram showing auto-correlation technique](image)

**Figure 3.4** Example of auto-correlation technique applied in PIV (Raffel et al. 1998)

![Diagram showing cross-correlation technique](image)

**Figure 3.5** Example of cross-correlation technique applied in PIV (Raffel et al. 1998)

Auto-correlation has two distinct disadvantages, which are:

i. A correlation peak exists at the origin of the spatial separation, which corresponds to a zero particle displacement formed by a particle image correlating with itself. This peak is the largest possible peak in the system but is not representative of the particle displacement. This leads to complexity in locating true particle pairs and identification of particle pairs with a separation less than twice the diameter of an average particle.

ii. A 180° directional ambiguity exists in this method of correlation, due to the way the correlation is formed it is not possible to identify which particle was identified first.
Cross-correlation does not suffer these problems due to the image being formed on two separate images and analysed individually. In DPIV (Digital Particle Image Velocimetry) each frame is labelled by the frame grabber to remove any ambiguity between the first and second images. The development of CCD cameras has allowed higher framing rates in the cross-correlation method, thus succeeding the auto-correlation method as the main technique of PIV.

In earlier auto-correlation work, a method used to overcome directional ambiguity was to use different wavelengths of light from the PIV lasers, hence giving different colours to each image recorded to the same medium (Haste 2000; Nino et al. 1993; Reeves et al. 1996). This method removed the directional ambiguity when using colour film.

3.5.2 Time resolved PIV

An advancement to the 2-D PIV method is the time resolved PIV. The principles of time resolved PIV are fundamentally the same as 2-D PIV in the way the images are illuminated, recorded and interrogated. However, the method takes many pairs of images in a short time period and reveals temporal development of the flow. The limitation with this system is the number of pairs of images that can be recorded in a short time period. The latest time resolved PIV systems available are capable of 1024 x 1024 pixels image resolution at a 1.5 kHz repetition rate (equivalent to running the engine at 1500 rpm and taking a PIV image pair each 6° crank angle), or by reducing the resolution of the camera to a 512 x 512 pixels can achieve 5 kHz can be achieved (equivalent to running the engine at 1500 rpm and taking a PIV image pair each 1.8° crank angle). With ever increasing camera speeds, this will become the most common type of PIV as it shows the development of a flow over time, rather than simple 2-D PIV which shows a snapshot of the flow.

Stolz et al. (1992) applied this time resolved technique to a square piston and cylinder (to remove curvature of optics). The optical CI engine was filmed during the induction and compression stroke. A drum camera was used to record the images using high sensitivity film with a resolution of 50 lines.mm\(^{-1}\). Low resolution film
was used due to the low power copper lasers that were employed for the experimentation. These images were processed and imaged using a CCD scanning camera, resulting in a 512 x 512 pixel resolution image, which was interrogated in regions of 5 x 5 mm. A technique of image shifting was applied through the use of the drum camera, effectively a known drum rotational speed was employed and a known image shift could be calculated from this, thus removing the directional ambiguity from the auto-correlation of the results. This system allowed up to 70 double-pulsed PIV images per stroke, with the engine running at either 200 or 400 rpm. The results showed good correlation with 3-D CFD models of the flow, showing more turbulent structures than predicted. However, there were spurious neighbouring vectors, which indicated that the spatial resolution was too low to resolve small-scale structures of the flow. The main disadvantage of this research was the geometry of the engine, but it proved a time resolved PIV movie could be applied to CI engine.

This work was furthered by Koehler et al. (1993) on the same square piston engine, using a higher resolution of film (100 line.mm⁻¹), producing more comprehensive results from the engine.

Lecordier and Trinite (1999) also investigated time resolved PIV. In their work they considered a digital system utilising CCD cameras with a maximum framing rate of 5 Hz for a 256 x 256 pixels resolution. Too increase this framing rate the resolution suffered, hence they used a drum camera and scanned the negatives for digital post processing. The major disadvantage of the drum camera system is the limited amount of time data that can be taken. However, the high framing rate (up to 12 kHz) is a major advantage over the digital system. In this study, flame propagation was investigated. This technique could be employed for the entire IC engine in-cylinder flowfield.

Current work at Loughborough (Jarvis et al. 2005) uses high speed time resolved PIV systems in an SI engine, successfully up to a frame rate of 5 kHz with a 512 x 512 pixels image. The system used was purchased after completion of testing on the CI engine in this present study and uses the latest (TSI Inc.) PIV equipment. This represents a considerable advancement on the PIV equipment used in the current work.
3.5.3 Endoscopic PIV

Optical endoscopes have been used to transfer the image from within the cylinder to an externally mounted camera. A light source is still required to illuminate the particles in the flow; therefore, the engine has still to be designed for some optical access into the cylinder. The advantage lies in that endoscopes, being a small unit, can be placed into production cylinder heads, reducing the cost of experimental engine, and a reduction of quartz glass parts within the engine. A problem encountered when endoscopes are used is the distortion of the images obtained. This is caused by the wide-angle lenses that are necessitated to create the required field of view within the engine (Gindele and Spicher 1998). The distortion present in the images results in PIV data showing incorrect velocity distributions. Gindele and Spicher (1998) have reported that such distortions can be mathematically corrected by further calibration of the optics. The method, at present, is still in development, largely owing to the limited optical access to the engine. A light sheet must be generated inside the engine, and this does not currently allow a large enough field of view of the flow.

3.5.4 Holographic PIV (HPIV)

Holographic PIV offers the advantage of being able to record the particle cluster displacement in three planes. In conventional 2-D PIV only the amplitude information is recorded and the phase information is lost, whereas in holographic PIV both amplitude and phase information are recorded to film, where upon the phase information is encoded in a reference beam in addition to the light scattered from the particles (Kurada et al. 1993). One of the difficulties encountered with this method is the time taken to record a stable interference pattern with particles that are moving in the flow, thus altering the object wave and interference pattern. As in 2-D PIV, the flow is seeded with particles, which are calculated to follow the flow. The first exposure records the particles 3-D position and the second exposure records the position the particles have moved to. The hologram is re-illuminated and the image of the particle reconstructed revealing both recorded images. Provided the first and
second exposure of the particles can be distinguished, the measurement of the magnitude and velocity of the particle (hence the flow) can be determined.

The processing of the holographic images can be performed via manually operated visual systems or automated environments. Digital image processing performs the majority of the processing required for HPIV. This type of processing involves digitising the volume of interest on a pixel-by-pixel basis with the intensity of each pixel being assigned a corresponding greyscale value. Post digitising, the two holograms are interrogated by two main methods:

i. **Tracking approach.** Consists of tracking the particles from one frame to the next, but is limited to when the displacement of the particle is greater than its own size between the images and the seeding density is low enough to prevent overlap.

ii. **Transformation approach.** This type of processing is split into two main categories, which analyse the multiple exposed images. The first type of processing utilises the Young’s interference fringe to measure accurately the local velocity vector. The second type employs Fourier filtering to determine the global distribution of velocity components.

HPIV was applied to in-cylinder flows in an IC engine (Konrath et al. 2001). A Suzuki DR750 motorbike 4-stroke single cylinder engine was motored at 1500 rpm. Optical access was gained by the use of a Perspex optical liner addition and an optical piston arrangement with mirror access from below the piston. The HPIV system used a full field of view of the optical area of the engine, with the ability to resolve spatial turbulence structures as small as 1 mm.

A novel type of holographic PIV technique was employed on the current optical CI engine in this research and this will be discussed in Chapter 7 with example data of holographic results.

The suitability of this measuring technique to the measurement of in-cylinder flows is currently being further developed, the main disadvantage being the time required to
obtain large datasets of the velocity vectors in the flow. The major advantage is a 3-D vector map that can spatially resolve the majority of the piston bowl.

3.6 Chosen measurement technique

From the measurement techniques that were presented in this chapter, 2-D DPIV was chosen for the measurement of in-cylinder flows in this research. This technique allowed a whole field measurement across the piston bowl and a large enough dataset to be recorded. These datasets could then be further analysed for ensemble averaged data and turbulence intensity. DPIV has emerged in recent years as the preferred measurement technique whereby near real time analysis of large datasets can be performed. The exact PIV set-up will be described in Chapter 6.

3.7 Seeding

The quality of PIV results obtained is directly dependent on the quality of the seeding employed in the system. The motion of the fluid is a translation of the particle movement in the flow, therefore, the knowledge of the particles following the flow is of great importance. The main requirements of the seeding material have to satisfy the following criterion:

i. Flow following with low temporal error (typically less than 1%)
ii. Does not alter the properties of the bulk fluid
iii. Seeding concentration sufficient for PIV measurement technique
iv. Does not readily agglomerate or damage the optics in the system
v. Scatter enough light to record the image to CCD camera.
3.7.1 Application of seeding to DI CI engine

Application of the PIV technique was applied to many different engine flows, the difficulty in the instance of a DI CI engine (bowl-in-piston) is that the optical access is restrictive, due to the design considerations made in order to maintain normal engine design parameters. With respect to the engine in this work, a compression ratio of 16.4:1 was used. Thus applying adiabatic compression results in

\[
\frac{P_2}{P_1} = \left( \frac{V_1}{V_2} \right)^\gamma \Rightarrow \frac{P_2}{101.325} = \left( \frac{16.4}{1} \right)^{1.4}
\]

(3.1)

\[
\frac{T_1}{T_2} = \left( \frac{V_2}{V_1} \right)^{(\gamma-1)} \Rightarrow \frac{298}{T_2} = \left( \frac{1}{16.4} \right)^{(1.4-1)}
\]

(3.2)

An illustrative pressure and temperature of the air at TDC during the compression stroke on a normally aspirated DI diesel engine is ≈ 51 bar and 912 K respectively. The optical access for the light sheet is through a quartz ring addition to the cylinder liner and a window in the cylinder head, which replaced the injector nozzle and one of the exhaust valves (discussed in Chapter 6).

Due to the high pressure and temperature involved in the standard arrangement of the engine, liquid seeding is not possible due to evaporation or combustion of the seed particles. The choice of seed particle for this current research is given in Chapter 6.

3.7.2 Flow following behaviour of seeding

The flow following behaviour of the seeding was an area of much study for the LDV technique and these calculations are identical for the seeding of flows for PIV studies. The force exerted on a particle is caused by the viscous drag from the surrounding fluid. The magnitude of force that is exerted on the particle from the fluid is the controlling factor in the flow following behaviour and determines the response of the particle to velocity fluctuations. A mathematical analysis of this was carried out by
Drain (1980) and is as follows. Using Stokes Law, the viscous drag on a spherical particle is

\[ F_p = m_p \frac{dv_p}{dt} = 6\pi \mu a (U - v_p) \quad (3.3) \]

where \( F_p \) is the viscous drag force on the particle (N)

\( U \) is the velocity of the fluid (m.s\(^{-1}\))

\( v_p \) is the particle velocity (m.s\(^{-1}\))

\( a \) is the particle radius (m)

\( \mu \) is the dynamic viscosity of the fluid (N.s.m\(^{-2}\))

\( m_p \) is the mass of the particle (kg).

Hence for a constant fluid velocity \( (U) \), the solution to Equation (3.3) is

\[ v_p(t) = U + (v_p(0) - U) e^{-t/\tau_p} \quad (3.4) \]

where \( v_p(t) \) is the particle velocity at time \( t \), and after a change in fluid velocity, the particle velocity approaches the new value of \( U \) with a time constant \( \tau_p \), given by

\[ \tau_p = \frac{m_p}{6\pi \mu a} = \frac{\rho_p a^2}{18\mu} \quad (3.5) \]

where \( \rho_p \) is the particle density.

To measure the frequency response of the particle relative to fluid velocity fluctuations, the fluid velocity variation is considered to be sinusoidal with amplitude \( U_0 \) and frequency \( f_u \). The equation of motion becomes

\[ \frac{dv_p}{dt} = \frac{(U + U_0 \cos(2\pi f_u t) - v_p)}{\tau_p} \quad (3.6) \]

where \( \overline{U} \) is the mean velocity of the fluid. Thus the steady-state equation of this becomes
where $\phi_i$ is the phase lag given by

$$\phi_i = 2\pi f_u \tau_p$$  \hspace{1cm} (3.8)

A common criterion for the response fidelity of the particle seeding for most applications is that the amplitude of variation of $v_p$ is equal to that of $U$ to within 1% of the maximum fluctuation. The condition for this is derived from Equations (3.5) and (3.7), as follows

$$a^2 < 0.1 \frac{\mu}{f_u \rho_p}$$  \hspace{1cm} (3.9)

this can be re-arranged so as to show the frequency for a given viscosity, particle radius and density by

$$f_u < 0.1 \frac{\mu}{a^2 \rho_p}.$$  \hspace{1cm} (3.10)

This analysis is a simplified method for the flow following of particles. The results of this simple calculation for flow following behaviour for the present study are presented later in Chapter 6 for the different sizes and types of seeding available for this project.

3.7.3 Vortical studies of particle seeding

This area was of interest with respect to bowl-in-piston combustion chambers, where a swirling flow in the bowl was used to aid in the mixing and combustion process. The swirl rate and swirl spin-up are of particular interest and the following helps to describe the behaviour of seeding material in such flows. In work by Maurice (1992) the primary interest was vortical flows for LDV experiments.
In a vortical flow, the seeding can be moved away in a radial direction from the origin of the vortex, increasing error in the measurements that can be taken by a PIV measurement system.

To quantify the potential flow, three dimensionless constants have been developed to describe the particle position at any point in the flow and to describe the equation of motion as follows

\[
\frac{r}{r_0} = f(Re_0, \beta, St) \tag{3.11}
\]

where \( r \) is the radial spatial co-ordinate and the subscript 0 refers to the initial condition, \( St \) is the Stokes Number, \( Re_0 \) is the Reynolds number of the flow and \( \beta \) is a ratio of the densities. These are calculated as follows

\[
Re_0 = \frac{\rho_p v_{r_0}}{\mu} \tag{3.12}
\]

\[
\beta = \frac{\rho_p}{\rho_p} \tag{3.13}
\]

\[
St = \frac{\rho_p v_p d^2}{18r_0 \mu} \tag{3.14}
\]

All of these parameters vary through the flow. It is therefore necessary to hold them constant by defining them in terms of total density and temperature. Figure 3.6 shows how the particle bias is affected by the Mach number of the flow in a compressible regime.
Figure 3.6  Particle trajectories within a potential vortex (Maurice 1992)

If we consider the flow in a CI DI engine and use suitable estimates to approximate this flow, Figure 3.6 can be used as a guide to the error that will be incurred from the particles in the fluid flow. The maximum Mach number experienced in the flow of the engine is less than 0.05 Mach. The ratio of fluid particle densities is, therefore, approximately the same, as is the Stokes Drag. The deviation from this model is the value for the Reynolds number, which in the instance of the CI DI engine at higher crankshaft speeds returns a value for the relative particle Reynolds number \(< 10\). Considering Figure 3.6, the error that will be incurred by the particle in the radial direction will be \(< 10\%\) for a particle travelling 90° of the vortex. As this work does not state the actual function used in this calculation, it can only be used as a guide.

3.7.4 Light scattering of seeding particles

Since light needs to be scattered by the seeding particles in the flow, an understanding of the light scattering behaviour of seed particles is necessary. The recorded particle image intensity (and hence the contrast of the PIV image recording) is directly proportional to the scattered light power. This scattered light power is dependent on two main factors, the laser power and the properties of the seed particle. Since the associated cost of increasing the laser power is very high, it is often more economical and effective to choose more appropriate seed particles (Raffel et al. 1998). Raffel et al. states that for the majority of cases, light scattered by small particles is a function
of the ratio of the refractive index of the particle to that of the surrounding medium, the particles size, shape, orientation, polarisation and the observation angle.

In the case of solid seed particles in air (which are assumed to be spherical) that are of diameters greater than the wavelength of the incident light, Mie's scattering theory can be applied. The principle of Mie theory is that when the incident light is randomly polarised, the light scattered by a small spherical particle consists of two incoherent, plane polarised components with mutually orthogonal planes of polarisation (Emrich 1981).

This distribution of light allows for greater forward scatter of light than backward scatter. This can be seen in Figure 3.7.

![Figure 3.7 Light scattering of 1 μm oil particle in air illuminated by laser of 532 nm (Raffel et al. 1998)](image)

Figure 3.7 shows the polar distribution of light for a spherical particle illuminated by a laser sheet in air. The intensity scales are logarithmic and the neighbouring circles increase by a factor of 100. The Mie scattering can be identified by the normalised diameter \( q \) defined as

\[
q = \frac{\pi d_p}{\lambda} \quad (3.15)
\]

where \( d_p \) is the particle diameter

\( \lambda \) is the wavelength of the laser light.
By increasing the value of $q$, the ratio of forward to backward scatter increases rapidly, hence it would be best to record the image in the forward scatter mode. Unfortunately this is impractical due to the limited depth of field. Therefore in the majority of PIV experiments the scattering is recorded at 90° to the light sheet. Emrich (1981) and Raffel et al. (1998) provide further details of Mie theory.

3.7.5 Introduction of seeding to charge air

In the seeding of gas flows, such as the case of an engine, the quality and feasibility of introducing the flow tracer particles is problematic. In the case of solid seed particles in a gas flow, the particles are difficult to distribute and often agglomerate at the area of measurement. It is therefore impractical to supply the particles a long time before the measurement area. Delivery has to be injected into the flow shortly before the test section. This injection of the seeding must not significantly affect the flow and has also to ensure homogeneous particle distribution in the flow. Commonly for solid seeding, the introduction of the seed material is achieved by either a fluidised bed or by air jets. The seeding delivery method used in this present study is discussed in Chapter 6.

3.8 Alternative optical methods

As an alternative to introducing seeding into the flow, it is possible to record the motion of the in-cylinder flow using the flame front of the combustion. This technique does not require illumination of the flow or the introduction of seeding since the light emitted from combustion allows recording of the motion of the flame front. This is much more difficult to observe in CI engines because the created combustion pressures are high and do not usually allow for optical access into the engine by quartz windows. One method of observing firing CI engine conditions is to redesign the engine, as carried out by Fujikawa et al. (1988). In this instance schlieren photography was used to observe the firing engine. To allow adequate optical access, the piston and cylinder were designed as a square cross-section with
one flat side being a quartz window. Another type of engine cylinder was also
developed in this work where the cylinder bore was of a distorted shape to take
account of the light path through the glass, this can be seen in Figure 3.8.

![Diagram](image)

Figure 3.8 Cross-Sectional view of the SVC cylinder and the path of light
rays incident to the cylinder (Fujikawa et al. 1988)

The material used for the cylinder was a transparent acrylic, which was not
appropriate for high temperature or pressure applications. Whilst optical recording of
the combustion in a CI engine allows for greater understanding of the burning regions,
it does not allow for quantitative knowledge of the charge air motion prior to the fuel
injection event.

3.9 Summary

This chapter has investigated the four main flowfield measurement techniques that are
available to be used with IC engines. Particular emphasis was given to the PIV
technique since this was the chosen technique employed in this project. The PIV
technique is still developing and an area of particular interest for future studies is
time-resolved PIV, this allows potential research into the processes involved in
altering the flow with a time history to each vector map. Holographic PIV was briefly
described. A novel type of holographic 3-D PIV was applied to the engine in this
current research and will be discussed in Chapter 7. The following chapter will
review work of others in the area of optical measurement techniques.
Chapter 4  

4.1 Introduction

This chapter reviews methods of optical measurement and in-cylinder flow studies that have been carried out on IC engines. Much of the recent work that has been carried out in this area is primarily involved with the development of modelling and the validation of models.

4.2 Literature Survey

It has long been realised that the interaction between air and fuel is an important factor in combustion (Ricardo 1930). As such the development of suitable equipment enabling the monitoring of in-cylinder flows was the major challenge. The basic principle of measuring a flow of air is to introduce a seed material to the air which will give a visual representation of the flow structure. This flow structure can then be recorded by an optical medium and interrogated for the movement of the flowfield. Experimentation of this principle, with respect to engines, was employed as early as 1939 (Lee 1939) whereby photographing of an in-cylinder flow with a known time separation was experimented and feathers were used as the seeding medium. Images were taken over a space of 30° crank angle. Shrouded inlet valves were used in this work to give different swirl configurations and the engine was motored at 1000 rpm, injection of fuel was also attempted but found to be too difficult to visualise.

Early research of the in-cylinder flow was carried out by Semenov (1958) where hot wire anemometry (HWA) was used in a motored IC engine (motored at 600-1200 rpm, compression ratio of 4:1 to 6:1). The hot wire anemometer was located 10 mm below the cylinder head flame face, but was moveable about the cylinder radius to measure different flows about the valves. Since the hot wire method measures the resistance change across the wire according to temperature, a statistical method of compensation for temperature and density fluctuations had to be introduced to show relevant turbulence data. These compensations are believed to have brought the
fluctuating turbulence to within 2% accuracy. Due to complexities of directional ambiguity using hot wire anemometry and the complex flow behaviour inside an IC engine, turbulence intensity \((u')\) was observed to be of 15 - 20% accuracy. Data was however obtained for the intake and compression strokes of the engine relating the instantaneous velocity to the crank position. A significant reduction of turbulence was observed immediately after inlet valve closure (IVC) and this lower turbulence intensity was observed for approximately two thirds of the compression stroke. The turbulence intensity was then seen to increase suddenly towards TDC due to the increased density of the charge (the density increased due to increased pressure in the cylinder). This work related the intake generated turbulence to compression and overall turbulence, as well as the engine parameters, volumetric efficiency \((\eta_{vol})\), engine speed \((N)\) and swirl ratio.

Tindal et al. (1974) further investigated the effects of swirl ratio on turbulence, but used a combination of hot wire anemometer probes within the cylinder. A masked valve was utilised for varying swirl ratio and spacers employed to allow varying compression ratios (6.7:1 to 10:1). The anemometer probes could be placed at different positions within the cylinder, as well as being moved through the cylinder by means of a cam running in phase with the engine. The hot wire anemometer was compensated for different temperatures within the engine and used a multi-wire orientation to determine the velocity vectors of the flow. The swirl in the engine was measured using the hot wire anemometer and considered to be a solid body rotation at the outer radii using the mean values of velocity from the anemometer. A bowl-in-piston arrangement was used, in which the probe was moved off-centre at TDC and penetrated into the bowl to take in-bowl recordings. These recordings are shown in Figure 4.1. The results are compared to a flat piston crown arrangement and show a marked increase in velocities at the TDC position.
Measurement of tangential rate of flow, 70% piston bowl radius, 0.25 inches below the top of the bowl. Bowl-in-piston crown, masked valve angle 90 degrees, speed 1000 rpm (Tindal et al. 1974)

Brandl et al. (1979) further researched the turbulent airflow in the bowl of a piston in a DI CI engine. Three different intake port designs with the same swirl number were used (two helical and one directed), with a compression ratio of 18:1 and varying engine speeds (1100 - 2400 rpm). Hot wire anemometry probes were mounted in the piston-bowl for flow velocity measurement. These tests concentrated on emissions and brake specific fuel consumption (bsfc) of the engine and the relevance to inlet geometries. It was found that the integral scale of turbulence at TDC was more dependent on bowl geometry than inlet port geometry. This was concluded to be due to the bowl dimensions determining the final eddy size. A trend towards higher integral scale turbulence was observed at the bowl centre for helical inlet geometries.
Figures 4.2 and 4.3 shows the results of Brandl et al. for swirl studies in a running engine. The results in Figure 4.2 show how the different inlets, although having the same steady-state swirl rig results, have a marked difference in swirl experienced in the engine. The tangential inlet was seen to not produce the same levels of swirl that were created by the helical ports. The swirl present in the upper part of the bowl for all three inlet configurations rotates faster than the lower part of the bowl up to TDC.
Thereafter, (for all the engine speeds tested) the lower part of the bowl was seen to rotate faster. Brandl et al. attributed this variation in the swirl to the conservation of angular momentum when transferring the air charge into the piston bowl. The result seen in the upper bowl is affected earlier by the forcing of the air charge into the bowl and hence the increase in tangential velocity, and later in the lower part of the bowl.

The swirl numbers in Figure 4.3 were determined from averaged results from 10° BTDC to TDC and are plotted against engine speed. The main difference observed with inner radii results is that the lower part of the bowl returns a higher speed than the upper part of the bowl. The difference between the bowls varies widely for engine speed, showing no direct correlation with swirl to engine speed. The different swirl ports were then tested on a firing engine for emissions and bsfc of the different inlet configurations. The smoke level was seen to be controlled by the velocity and turbulence in the centre part of the bowl. With lower engine speed the tangential velocity on the outer radius of the bowl was seen to be the dominating factor for the controlling the smoke level; this was considered to be due to the turbulence and velocity in the centre of bowl being too low to be the dominating effect. The bsfc was found to be controlled by the tangential velocity on the periphery of the bowl, with a general trend of bsfc decreasing with increasing tangential velocity. This reduced bsfc was accompanied with increased burning rate of the fuel at the beginning of combustion, suggesting that higher tangential velocities at the bowl periphery increased the amount of air-distributed fuel being burnt first. NOx was seen to rise with increased tangential velocities and burning rates, although significantly lower bsfc was achieved. Although all the ports reported the same swirl number on a steady-flow test rig, the helical ports showed higher in-cylinder flows (in terms of mean tangential velocity, turbulence intensity and micro-scale turbulence) for all engine speeds when compared with the tangential port. The overall conclusion of this work are that the helical inlet ports are superior to tangential ports in terms of bsfc and smoke over a wide engine speed range.

An alternative method used to visualize the induction swirl is that of ‘water analog’ as reported by Khalighi (1990). This type of experiment uses dimensionless analysis by replacing the intake charge air with a liquid at the same Reynolds number (Re). The water analog method allowed similar flow characteristics to be observed at lower
speeds (approximately 10% of equivalent engine speed). The rigs investigated could only consider the intake flowfield since liquids are almost incompressible. Water analogs are favourable due to the reduced velocities inside the engine to a suitable level to be able to record the flows experienced in the engine. These types of experiments are flawed in that they only consider the induction stroke. Whilst this provides some useful data, the actual mixing processes take place during the compression stroke, resulting in limited data. The results primarily addressed the tumble structure of the flow within the engine and employed PTV (Particle Tracking Velocimetry) to gain a picture of the intake process.

Similar studies were also carried out by Choi and Guezenec (1999) using 2-D PIV, 3-D PTV and LIF. In this work, DPIV was employed and cross-correlation was applied to the results of the two images, hence removing the need for image shifting techniques or directional ambiguity of the flow. The main focus of the PIV technique was to investigate the tumble flow effect during the induction stroke. Fifty complete intake strokes were taken at 0.25° of crank angle. These images were phase averaged at each crank angle to produce an average image at a specific crank angle. This average image was then subtracted from individual images to remove background lighting and to improve the image quality and contrast prior to correlation.

Work carried out by Yianneskis and Easson (1993) had a similar arrangement to that of a water analogue engine, but the working fluid was replaced with a mixture of turpentine and tetraline, yielding an identical refractive index to that of the acrylic material (at 21.5°C) employed for the test engine set-up. A novel system was applied to illuminate the flowfield. The laser beam was projected continuously onto a rotating mirror with several facets. When the mirror reached a certain point it reflected the laser beam into a parabolic mirror which illuminated the flowfield. This is illustrated in Figure 4.4. The time separation was dependent on the number of facets and the speed of rotation of the mirrored polygon, which could be controlled.
Figure 4.4 Rotating mirror system used to pulse the light sheet on the flowfield (Yianneskis and Easson 1993)

Images were taken of the flow in a similar fashion to other doubly exposed auto-correlation PIV. However, instead of then imaging the recorded photograph negative to interrogation regions, which are inspected and digitised by a CCD camera, the negative was enlarged to an A4 photographic print and scanned on a flat bed scanner. The flat bed scanner digitised the photograph at a resolution of 300 dots per inch (dpi) with each point being assigned a greyscale value between 0 – 255. Smaller areas were then interrogated on a computer and Fourier transforms applied to yield the auto-correlation peaks, the velocity vectors and the magnitude. Directional ambiguity was not considered in this study since the absolute direction of the bulk flow motion was known.

Intake generated turbulence was investigated near the intake valve and how variation of the delivery of the charge into the cylinder affected turbulence. Bicen et al. (1985), for example, investigated steady and unsteady air flow through the intake valve using LDV. In the experimental set-up, an acrylic (Perspex) cylinder was used up and downstream of the intake valve and varying tests performed with both steady and unsteady air feed and piston arrangements. Different valve geometries were considered and measurement of the discharge coefficient for each geometry was made with different valve lift and pressure drop conditions. Velocity measurements of the
intake flow were taken at the exit of the valve and downstream to create a flowfield. It was shown that the mean flow patterns remained the same during the induction process. The turbulence intensity velocities increased during the mid-late induction due to the development of the flow to fully turbulent. These results were compared to steady-flow tests and showed that at the exit of the inlet valve, the radial and axial velocity distributions were very similar, however, the turbulence intensity in the unsteady tests was significantly higher. This showed a good correlation between steady-flow test rigs and unsteady motored engine flows at the exit plane of the inlet valve in a low speed engine.

Corcione et al. (1991) investigated, using LDV, the difference between re-entrant and straight sided piston bowls with emphasis on emissions and in-cylinder flow. Results were taken at 15° BTDC on compression stroke at an engine speed of 1000 rpm at different depths from the cylinder head flame face. It was found that the re-entrant type of bowl toward the end of compression increased the swirl by up to 50% at 15° BTDC and up to 25% at 5° BTDC, with areas of greater tangential velocity and turbulence intensity inside the re-entrant type of bowl. These studies included mixture analysis and emissions analysis. It was also found that by spraying the fuel into the areas of greater turbulence in the re-entrant bowl, better fuel-air mixing took place and reduced the smoke emission whilst maintaining bmep and bsfc.

Arcoumanis and Whitelaw (1991) made investigations into in-cylinder flows in motored engines for SI and CI engines with different bowl configurations and shrouded valves (helical and directed inlets), using the LDV method. Steady inlet conditions were measured for varying engine speeds (500 to 3250 rpm) and used as boundary conditions for a multidimensional model that was being computed for the engine. Measurements of the flow were taken for varying inlet geometries and it was found that in most cases the swirl memory remained throughout the time from IVC to TDC compression. Similar turbulence velocities in the range of 0.5 - 0.7 mean piston velocity were shown. However, for the high swirl case it was found that the turbulence velocity levels increased by approximately 15% when compared to low swirl turbulent velocity. They also studied the effect of mixing in a DI Diesel engine arrangement, whereby a heavy gas (R12) was used in place of fuel and injected through the inlet port. The air charge was seeded with hollow micro balloons and
their velocity was measured using a laser Rayleigh scattering method. The impingement of the injection jet and the interaction with the in-cylinder flow was recorded by photographic means. Ensemble averaged axial and swirl velocities were obtained at TDC, using different injection nozzles (single and multiple hole). The main area of interest was the injection impingement region. It was found that with multiple nozzle injectors, the fuel jets had a faster decay rate, reduced penetration and enhanced mixing, creating a near homogeneous charge. It was also reported that the centrally located injector aided, through the fuel spray, stabilisation of the swirl centre of the air charge motion.

Bopp et al. (1986) investigated the relationship between engine speed and swirl velocities at TDC using a disc type combustion chamber with an optical window. LDV was used for taking velocity measurements. Comparisons of the turbulence at TDC with and without swirl were also examined. It was shown that the turbulence intensity at TDC markedly increased with engine speed, showing an exponential relationship. This was observed both with and without induction swirl. These results were normalised against the average piston speed and it was shown that without inlet-induced swirl, there was approximately 10 – 15% increase in turbulence intensity throughout the engine speed range. The results also showed in both instances of swirl and non-swirl that during compression, the turbulence was generally non-isotropic but tended to isotropic at TDC. Corcione and Valentino (1994) further investigated the in-cylinder air turbulence and the dependence of engine speed using an optical access engine and LDV measuring technique with a re-entrant type piston-bowl. Two locations of measurement were taken, at a 5 mm depth from the bottom of the cylinder head flame face. A linear dependence of ensemble tangential velocity was observed between the same crank angle positions over engine speeds of 1000 to 3000 rpm, see Figure 4.5.
The reduction of the swirl above 3000 rpm with the 24 mm diameter measurement (closer to the piston bowl lip) was attributed to the combined effects of squish, swirl and the chamber wall friction. It was also shown that squish interacted towards the end of the compression stroke to slow down the swirling flow and produced high shear regions that increased the turbulence intensity at both locations. The results showed a strong correlation between engine speed and swirl speed, deviating at the 3000 rpm test point and the outer test point, where a decrease in the swirl level was observed.

PIV measurements were carried out in the bowl of a piston by Sweetland and Reitz (1994). In this research, a motored single cylinder diesel engine with optical access from a window in place of an exhaust valve and the fuel injector in the cylinder head was employed. A conical mirror was located on the centre of the piston, which created the laser sheet within the cylinder. The running gear in the top end of the engine was operated without oil, but used grease instead. This removed problematic oil splash near the window in the cylinder head. Turbulence information of the flow (from using an ensemble averaging technique applied to the PIV data) was taken over a number of cycles and compared to KIVA-3 CFD prediction code. The mirror mounted on the piston altered the piston geometry (by approximately 4 mm in height) but was not believed to have significantly interrupted the flow around the bowl.
Problems arose in this research due to engine vibration. It was found that the movement of the cylinder head relative to the laser beam could block the path of the laser. As a result the smallest beam diameter that could carry sufficient energy to create the light sheet in the bowl was used. Data in the bowl was acquired at two points, 15° and 25° BTDC of the compression stroke at an engine speed of 750 rpm. Within the PIV results obtained, many out of plane flows were found and hence reduced the number of particle pairs matched. These out of plane flows were believed to be due to the toroidal flow pattern set-up in the bowl, as shown in the schematic of Figure 4.6, which was previously observed by Arcoumanis et al. (1983).

![Toroidal flow in piston bowl](image)

**Figure 4.6** Location of PIV measurement region and qualitative description of engine flowfield (Sweetland and Reitz 1994)

Bulk motion was difficult to determine in these experiments because of the small field of view into the engine. However, from the PIV data the turbulence intensity, turbulence dissipation rate and integral length scale were derived from methods that had been developed from PIV measurements of simple turbulent jet flows. There was degree of success with the comparison of predicted KIVA 3 code and the PIV measured data, showing under prediction of velocity magnitudes by the KIVA 3 code. Integral length scales and turbulence intensity did show a high correlation between the instantaneous results of PIV and the averaged results of KIVA 3 code. Example PIV data obtained from this engine is shown in Figure 4.7.
In Figure 4.7 the diverging lines represent the laser light sheet and the circle indicates the location of the exhaust valve window of diameter 32 mm. It was reported to be difficult to obtain a dataset, and as the vector map shows only a small segment of the flow, it is difficult to extrapolate data for the whole bowl.

Miles et al. (2001) carried out more up to date analysis of turbulent structures in a motored, modelled and fired HSDI (high speed direct injection) CI engine with a re-entrant type bowl. The engine had optical access from below the piston as well as from the side through the use of an extended piston incorporating a quartz piston top. Air supplied to the engine could be varied in temperature and pressure and held constant for experimentation by use of a plenum. LDV was employed for the optical measurements at two locations (inside and outside the bowl) and measured the tangential and radial velocities of the flow. The quartz piston measurements were limited to measuring tangential velocity due to the complication of the curved bowl surface observed from beneath. Zirconium Oxide seeding was used and was introduced into the flow via a fluidised bed seed generator. The engine was motored at idling speed and skip fired experiments were carried out whilst moving the measurement point, changing the investigation to suitable positions to measure
velocities at different crank angles in different cycles (compression and expansion). The measured data from this work showed a good correlation with computer predicted results based on the same inlet conditions for the compression and expansion strokes. However, the turbulent intensity did deviate from model prediction, showing that the model results deviated from experimental data due to spatial distribution of turbulence.

Much of the early PIV work carried out used wet-film methods and auto-correlation techniques, however, with the improved digital CCD camera technology and lowered cost of digital systems, much of the latter work employs DPIV using tow frame cross-correlation as the image analysis method.

Work by Reuss et al. (1989) carried out PIV in a 12 x 32 mm area inside a two-stroke SI engine. Suitable optics was placed to allow access for a laser light sheet inside the engine and additional optics fitted into the cylinder head so that images of the flow could be recorded. The engine was motored at 600 rpm and images recorded at 78° BTDC on the compression stroke and 12° ATDC on the power stroke. The laser was double pulsed with a separation of 20 – 40 μs, resulting in displacements of the paired particles in the region of 0.2 mm. The images were recorded to photographic film (400 lines/mm) with a laser sheet thickness of 0.3 mm. Each negative contained two images of the particle with an image separation. The seeding was achieved by introducing a secondary dry airflow saturated with Titanium tetrachloride (TiCl₄), which was mixed with the humidified main air to form particles of Titanium dioxide (TiO₂) with a primary particle size of < 1 μm. This seeding provided approximately 100 particle image pairs in a 1 mm area of 0.3 mm thick. The interrogation was carried out by illuminating (using a Helium Neon (HeNe) laser) 0.9 mm square areas within the photograph onto a 256 x 256 CCD camera, digitising the image and performing an auto-correlation on a computer. The results suffered 180° directional ambiguity, however, this was considered to be of minor importance since a swirling flow was being investigated and the general direction of the flow was known. Further processing involved manually removing ‘bad’ vectors (vectors that differed from the neighbouring vectors within a certain criteria) resulting in approximately 5% removal of vectors. Interpolation was then performed to replace the vectors that had been removed by manual means; these vectors were based on an estimate of the
neighbouring vectors. Smoothing was further applied to remove noise from the image using axi-symmetric Gaussian kernel criteria. In some of the tests large areas of data had no vectors at all. This was attributed to out of plane flows during the compression process.

Similar work by Schipperijin and Sawyer (1990) investigated the flow at the exit of the inlet valve and the associated jet entrainment using PIV technique. This work used an inlet simulator on a bench test. Two images of the flow were recorded to the film each showing the particle with a 40 μs separation. The region of interest in this study was 20 mm x 30 mm and the interrogation region applied was 1 mm square. Image shifting was used to remove the directional ambiguity of the particles. This was achieved by shifting the image between the first and second laser pulse a known distance which is greater than the particle displacement. The flow was seeded using 0.3 to 0.7 μm Aluminium oxide (AlO₂) particles introduced a suitable distance from the point of measurement. As was employed by Reuss et al. (1989), the image was captured by a CCD camera (256 x 256 pixels) in 1.4 mm square interrogation regions, digitised and examined using a computer. A Fast Fourier Transform (FFT) was applied to the raw data to remove the high and low frequency data (outside of an acceptable criterion). The array was then binarised so that the bright spots (areas where particle displaced light) had a value of unity and areas with no light had a value of zero. The central auto-correlation peak was masked out in this binarised form of the image and written to a data file on the computer. The process was repeated and a complete velocity vector field of the original photograph reconstructed.

Lee and Farrell (1992) investigated the exit flow through an inlet valve. A standard CI engine cylinder head (Cummins NH250) was fitted to an acrylic liner so as to carry out bench flow testing. In this testing the valves were actuated and PIV image pairs were captured for varying cam positions. Two Nd:YAG lasers were employed for the light source of the PIV experiments, each laser had an individual power source and was tuned to fire the same energy pulses with a time separation that could be set between 5 – 100 μs. The laser sheet was 0.5mm thick and TiO₂ (∼1μm) was used for seeding the flow. The images were doubly exposed and the image processing was carried out by using a collimated laser beam passed through the doubly exposed film. The particle images created a diffraction pattern on the projected image, producing a
Young's fringe pattern, from which the displacement and directional information could be determined, with a 180° directional ambiguity. The data was analysed optically. Instead of using a FFT to remove high and low data, an optical Fourier transform was used to obtain the auto-correlation of the particle images. The Young's fringe was captured by a CCD camera and played back on a liquid crystal television (LCTV). This image on the LCTV screen was then Fourier transformed optically through an additional collimated laser, a polarizer, a transform lens and a microscope objective. This final image was a 2-D auto-correlation of the particle images in the interrogation region of the film. Lee and Farrell's optical system is illustrated in Figure 4.8.

![Figure 4.8 Optical image processing system (Lee and Farrell 1992)](image)

It was found that in certain regions of the flow that no detectable peaks were observed, indicating that the velocity was below a minimum discernable limit for this configuration or that the velocity gradients in the interrogation region had smeared the correlations and that no detectable peaks were identified.

Reeves et al. (1994 and 1996) employed a PIV system to interrogate the in-cylinder flows of a production geometry SI engine. This work concentrated on creating a suitable PIV system to monitor IC engine flows, with the consideration of the difficulties that are encountered. A double pulsed Nd:YAG laser was used with time separations ranging from 50 – 400 μs for the flow illumination. The flow was
illuminated in both the horizontal and vertical plane. The camera used 35 mm film at a resolution of 300 lines/mm\(^1\) and the seeding employed was silicone oil (diameter \(0.5 - 2\ \mu m\)). This work introduced the use of corrective optics to remove astigmatism suffered from viewing vertical planes through a thick glass cylinder. A setting up stage was also discussed on how to ideally focus the camera lens and this was achieved using a CCD microscope that gave a highly magnified view of the region of interest and allowed it to be displayed on a monitor. Using this, the focus and seeding density of the images could be assessed prior to PIV experimentation. Processing of the data was carried out by a commercial optical and digital auto-correlation system. Interrogation regions were overlapped and post processing, editing and interpolation of vector maps was carried out. Optical access into the cylinder was achieved by having a thick glass cylinder liner. For vertical sheets in the flow, a 45° mirror was mounted at the bottom of the cylinder and beamed through a glass-topped piston. One of the important aspects of this research was the discussion of the limitations of PIV with respect to IC engines at that time.

As was briefly discussed in Section 3.5.1.3, Nino et al. (1993) used a 2-colour PIV system to measure IC engine flows. This work used a bowl-in-piston configuration to generate large swirling bulk motion in the engine. Optical access was gained through the use of a quartz ring addition at the top of the cylinder liner for the laser sheet and a quartz disc mounted in the cylinder head to image through. A dye laser was employed to alter the illumination light so as to distinguish which was the first and second photograph imaged thus removing the 180° directional ambiguity. The processing of the information used the cross-correlation method and highlighted the advantages of this technique when compared with auto-correlation.

Work carried out by Reuss et al. (1995) attempted to bridge the gap between computer simulation of in-cylinder flows and experimental results obtained from PIV. In this study an optical test engine (which was used in Reuss et al. 1989) was seeded and the particles were double exposed onto photographic film. An image shifting technique was employed to create a finite displacement between the exposures, thus removing directional ambiguity and zero displacement particles. A CCD camera was employed to process the data from the photograph; the photograph was measured in
1.25 mm square regions at each node of a rectangular grid with 2 mm spacing. It was reported that with high seeding density and this analysis method, vectors were yielded at 90 - 95% of the grid nodes in each photograph. A dataset of 20 - 30 processed PIV vector fields were collated for a given crank angle and averaged to create the ensemble mean velocity to compare with predicted computer simulation results. From this ensemble averaged data, individual PIV recordings were analysed to determine the ensemble fluctuation velocities, including indistinguishable contributions from both turbulence and cyclic variability.

With the improvement of CCD camera technology, DPIV has become the foremost and most widely used form of PIV using cross-correlation analysis. Work by Marc et al. (1997) used a DPIV system in a square piston geometry SI engine. Tumbling motion of the flow was the main aspect in this research. Oil droplets were employed for the seeding, these were imaged to a CCD camera (768 x 512 pixels) and cross-correlated by computer. The interrogation regions of the cross-correlation were 32 x 32 pixels with a 16 pixel overlap. A 2-D Gaussian curve fit of the cross-correlation peak was used to achieve sub-pixel accuracy. Difficulties were encountered with bright regions in the flow close to black painted walls and with low seeding densities. Thresholds were imposed in the post processing stage, based on the signal to noise ratio and velocity vector amplitude. This removed approximately 0.5% of 'bad' vectors and no interpolation was necessary. A dataset of 30 - 50 PIV processed recordings were taken to give a 95% and a statistical estimation of 0.3 m.s⁻¹ of absolute error on the mean velocity.

Endoscopic optics were used by Gindele and Spicher (1998) to record DPIV images in an optical SI engine. The main part of their reported research was the investigation of using endoscopic optics and two different ways of correcting the distorted images. The image magnification varies with its distance from the optical axis due to the lenses and aperture (known as the iris) used in endoscopes. Another problem that arose from the use of endoscopes was the strong illumination required to overcome the small aperture and number of optical elements the signal has to pass through. However, this is feasible with current lasers. Two methods were suggested for the correction, the first is applied to the raw data of the digital images where a calibration image was used to cross-correlate the distorted image at a certain point, thus
superimposing an image of correct calibration. The second method corrected the vector plot at the post processing stage, provided cross-correlation was possible with the distorted raw image. Vectors were assumed to be shortened due to the optics and the position to have been rotated. The known distance from the optical axis corrected this and the vectors were normalised with the vectors on the optical axis where distortion was zero. The second method was preferable due to the correction of the vector maps being carried out on a computer in close to real time, whereas the first method was constrained by time intensive rectification of the digital images.

Work by Hentschel et al. (2001) researched the two phase air-fuel flowfield existent in the bowl of a piston in a DI CI engine. A double pulsed Nd:YAG laser was employed with a light sheet thickness of 1 mm. Seeding employed was oil droplets in the range of 1 – 2 μm and a double image CCD camera with resolution 1280 x 1024 pixels was used to record the image. The field of view covered the entire bowl of the piston yielding a velocity map of the swirl spin-up during the compression stroke. A short time separation between the pulses was used (4 μs) and the engine was motored at 2000 rpm. Optical access was gained through a quartz ring between the top of the liner and the cylinder head, and the camera imaged, via mirrors, from the bottom of the engine through a glass observation window in the piston, in a similar arrangement to Reeves et al. (1994 and 1996). The laser sheet was 7 mm below the flame face of the cylinder head and measurements were considered at this point. A swirl flow velocity map across the bowl was recorded for different crank angle positions across the diameter of the bowl. The processing of the data involved high pass filtering of the raw data, an FFT and a 2nd order correlation. Post processing involved removing ‘bad’ vectors by a criterion set by previous experimentation but no smoothing or interpolation filling of data points was used. The results showed a good correlation of the swirl spin-up mechanism expected to be observed in this experiment. Fuel injection was also investigated in this engine and its interaction with the swirl. The work was being used to validate CFD codes. Whilst this is very pertinent to the present project, the paper did not give any details of the design of the bowl (which is most likely to be a flat bottomed piston bowl to allow for the optics) and the compression ratio was not reported.
The early work of Hentschel et al. (2001) was furthered (Hentschel et al. 2002) for PIV experimentation and expanded the testing to LIF investigation of the fuel air mixing in the liquid and vapour stage to reveal the fuel distribution in the cylinder after injection. High speed filming (8000 frame.s\(^{-1}\)) of the flame propagation was also carried out, using the light emitted from the flame. The PIV results showed that when using two inlet valves, which produced a weaker swirl, the centre of the swirl became offset from the geometric centre of the cylinder by approximately 8 mm. In the single inlet testing, which produced higher swirl, the swirl was seen to be centred on the geometric centre of the cylinder. The swirl velocity for the two inlet valve scenario was approximately half of the swirl velocity of the single inlet valve scenario. An attempt to record PIV with injection was found to be unsuccessful due to the light scattered by the injected fuel being greater than the light distributed by the oil seeding.

Choi et al. (2003) carried out an in-cylinder flowfield analysis using PIV and compared the results to CFD code. The engine used a simple flat bottomed bowl, with optical access from below the piston. This allowed measurements to be taken in both the horizontal and vertical planes within the cylinder. A single engine speed was used of 600 rpm, since the engine suffered too great a vibration level above this speed. Titanium oxide particles were used, fed by a nebuliser chamber to the engine. The particles were reported to be < 1 mm, which is a very large seeding for PIV, LDV or similar techniques. Measurements were taken on the compression stroke at crank angle positions 220\(^\circ\), 250\(^\circ\), 280\(^\circ\), 310\(^\circ\) and 340\(^\circ\) ATDC. The 340\(^\circ\) measurement was omitted from the vertical measurements due to the piston obscuring the view. The engine maintained a 17.7:1 compression ratio, which is of interest for this type of engine arrangement, where in previous studies only low compression ratios could be achieved with a large glass liner. The results of this work were compared with the CFD model of the engine with some degree of success. Results from the CFD code were seen to under estimate the flow when compared to the results from the PIV analysis. The main conclusions drawn were that the swirl flow in the flat bottomed piston bowl did not show distinct swirl until the squish effect appeared after 310\(^\circ\) ATDC. The vorticity and turbulence intensity increased after 310\(^\circ\) ATDC, resulting in lower velocity magnitude of the swirling flow.
In recent work by Deslandes et al. (2003) an optically accessible DI CI engine was used with two different piston designs, a flat topped piston and a flat bottomed bowl-in-piston. PIV measurements were taken over the whole piston area at different points in the stroke and at different locations. The engine used in this research was based on production geometry (except for the piston bowl) and achieved a compression ratio maximum of 18.1:1. Optical access was in a similar arrangement to a Bowditch piston (i.e. Reeves et al. 1994), where the image was taken from underneath the piston. This set-up did not allow for complex bowl shapes (i.e. pip centred bowl) as this would distort the image too greatly from below the piston. However, this method did allow for the visualisation of squish flows which were also researched in this work. Experimental PIV results were taken at six different planes and between three and nine vertical locations in the flow during the compression stroke, creating a 3-D picture of the average flow being experienced in the cylinder or piston bowl. Some 300 results were taken at each measurement plane and averaged to create the 3-D image and the centring of the flow during the compression stroke. The conclusion was that the charge air was much better centred with a piston bowl when compared to that of a flat piston. Swirl spin-up was also considered and found to be close to the theory proposed by Heywood (1988) for zero-dimensional model swirl spin-up, but deviated from the theory at TDC. The other main area of interest in this paper was the squish phenomenon, as previously discussed in Chapter 2. The squish flow was over estimated by standard calculations (Heywood, 1988) and found to be approximately 3 to 4 times weaker than theory predicted, suggesting little overall effect to the strong swirling flow experienced in the bowl of the piston.

This investigation was furthered in Deslandes et al. (2004) whose work was based around the development of the turbulence from intake to the exhaust phase using Proper Orthogonal Decomposition (POD) to aid in the modelling of the flow in the engine. The use of the POD mathematical model was an attempt to separate the cyclic variation (large scale of varying swirling motion) from the small scales of turbulence. This paper discussed the deviation of the flow from the presumed solid body rotation, where it was observed that at BDC the flow was non representative of solid body rotation and that it persisted through the compression stroke giving an unsymmetrical flow at TDC. It also mentioned that dry powder seeding was used but does not give specific details. PIV error measurement was estimated at 0.1 pixels,
this related to absolute error of instantaneous velocity less than 0.5 m.s\(^{-1}\). Results showed how, with a bowl-in-piston, there was better centring of the swirl and less cyclic variation when compared with a flat piston, whilst the swirl intensity and turbulence kinetic energy dispersion increased with a bowl-in-piston due to the amplification of the swirl speed. Of particular interest in this study, were the non-symmetrical flow properties of the charge air periphery of the bowl, this displayed non-symmetrical squish phenomenon towards the end of compression and large deviation from solid body rotation of the charge air. The ability to measure the squish phenomena led to a clearer understanding of the air charge motion. However, since a flat bottomed bowl was used, these results should be used with care when comparing to an ‘omega’ shaped piston bowl.

Further studies of in-cylinder flow using PIV measurements and POD analysis are reported by Grafiteaux (2001) where the PIV studies of a swirling flow were taken and analysed using POD algorithms to try and identify cyclic variability in unsteady swirling flows. This work showed how analysis of PIV results, which are in effect an interval in time, can be represented to show the large scale fluctuations in the engine cylinder. This work is a mathematical representation of the flow which is currently not yet thought to be used in CFD modelling techniques.

Similar studies using statistical methods were employed by Reuss (2000) where PIV (using film methods) was carried out on a two-valve SI engine. The aim of the research was to study the difference of cyclic variations between a highly directed swirl flow and non directed flow. Results of individual cycles were compared to the ensemble mean for similarities of the flow structure. Large scale cyclic variations were considered using a Probability Density Function (PDF) to describe the behaviour of individual cycles compared to the ensemble mean. Since this work used film methods for PIV auto-correlation, only a low number of samples were taken for each data point.
4.3 Summary

This chapter has presented some of the major findings from previous research in the area of in-cylinder flows utilising a variety of measurement techniques. Much of the data is applicable only to the specific engine that was used in the research, hence it is difficult to gain a holistic overview of the in-cylinder flows. However, it does emphasise the growing use of the PIV technique for measuring in-cylinder flows. Of particular importance were the methods used for gaining data via optical measurements and the design of the engines to allow for this optical access. The next chapter will present the current industrial method for quantifying swirl measurement and will cite the results of this testing for the optical cylinder head used in this work. This will be followed by a kinetic energy model used to determine the spin-up swirl values of the engine.
Chapter 5  Swirl Measurement and Modelling

5.1  Introduction

As previously stated, swirl is defined as the organised rotation of the air charge about the cylinder axis. The measurement and modelling of swirl is complex due to the many different parameters that need to be calculated in a constantly changing environment.

Current practise in industry is to use steady-flow test rigs, based on the work of Tippelmann for example, to measure swirl generated by the cylinder head ports. The measured swirl parameter is the swirl ratio, which is a dimensionless number that is defined by dividing the swirl speed by the engine speed (Equation (1.1)). It is then usually assumed that a linear relationship with the engine crankshaft speed exists. Hence the swirl ratio is maintained through different engine speeds.

The importance of these steady-flow test rig results lies in them being used as the basis for starting condition data for spin-up modelling. The majority of spin-up models consider the measured results from steady-flow test rigs to be the condition of the flow at BDC or IVC, which highlights the importance of these results. Since the results for cylinder head swirl testing is a dimensionless number, the true relationship between engine speed and swirl needs to be validated at TDC with the measured flow from this work. Whilst CFD is becoming more commonly employed for calculating the fluid movement in engines, these results have to be validated before their results can be used confidently in fully predictive engine design.

This chapter introduces the method of steady-flow testing the cylinder head and how these results are measured and calculated. A spin-up model, previously developed by Croston and Garner (2001) based on work by Borgnakke et al. (1981) is then described. The results from this modelling are used for comparison with the PIV measured data in Chapter 7.
5.2 Swirl measurement by standard steady-flow techniques

There are several methods used in the measurement and prediction of in-cylinder swirl. Industry typically uses swirl data as a major design tool for achieving the desired flow within the cylinder. Since many research documents about engines quote figures for the steady-flow swirl ratio, it is helpful to consider how the swirl number is defined. In general, the cylinder head of the engine is fitted to a rig where air is passed at a constant rate through the inlet port and valve into a dummy cylinder. One traditional method of measuring the swirl is to position a light paddle wheel within the dummy cylinder, which has a diameter approximating the bore of the cylinder. This paddle is centred on the centreline of the bore and situated a distance of about 1 - 1.5 times the bore diameter down the dummy cylinder.

The rotation rate of this paddle is used as an indication of the swirl. Since the paddle wheel design, location and the air flow rate affect the result, this type of measurement was superseded by the impulse swirl meter, proposed by Tippelmann (1977), as shown in Figure 5.1. In this method, the swirling flow exerts a torque on the honeycomb flow straightener allowing a swirl measure to be ascertained.

![Figure 5.1 Schematic of Tippelmann steady-flow testing rig (Tippelmann 1977)](image-url)

Figure 5.1 Schematic of Tippelmann steady-flow testing rig (Tippelmann 1977)
It is assumed that the changing velocity pattern of the flow can be neglected due to the equalising process in which only the sum of angular momentum of the intake charge are steady. This can be mathematically represented as

\[ T = \dot{i} \]  \hspace{1cm} (5.1)

where \( T \) is the driving torque
\( \dot{i} \) is the angular momentum flux.

Tippelmann's theory assumes the flow can be divided into two velocities, the axial \( (C_{ax}) \) and the circumferential \( (C_u) \). This is illustrated in Figure 5.2.

\[ \text{Figure 5.2 Two components of swirl flow} \]

The circumferential velocity is responsible for the angular momentum and is expressed as

\[ d\dot{l} = r C_u(r, \varphi)dm \]  \hspace{1cm} (5.2)

where \( r \) is the radius
\( \varphi \) is the segment angle of interest
\( \dot{m} \) is the mass flow rate of the air across the cylinder head.

\( C_u \) is calculated from
\[ C_w = \omega r \]  

(5.3)

where \( \omega \) is the angular velocity of the swirl flow.

The change of mass flow rate across the cylinder head is calculated as

\[ \dot{m} = C_{ax}(r, \varphi) \rho r \, dr \, d\varphi. \]  

(5.4)

Combining Equations (5.2) and (5.4) yields the following equation

\[ T = \dot{\dot{\int}} = \int \int \rho C_u(r, \varphi) C_{ax}(r, \varphi) r^2 \, dr \, d\varphi \]  

(5.5)

The following assumptions are then used

\[ C_w = \omega r \]

\[ C_{ax} = \text{constant} \]

i.e. the flow is assumed to be essentially 'solid body rotation'. Substituting into Equation (5.5) yields

\[ T = \hat{\dot{\int}} = \omega \rho C_{ax} \frac{R^4}{4} 2\pi. \]  

(5.6)

\( C_{ax} \) can then be defined as the flow rate across the cylinder head divided by the cross-sectional area of the cylinder, given as

\[ C_{ax} = \frac{V}{\pi R^2} \]  

(5.7)

and can be substituted into Equation (5.6), giving
\[ T = \dot{I} = \omega \rho V \frac{R^2}{2} \]  

(5.8)

and rearranging this to find swirl (\( \omega \)) of the flow by

\[ \omega = \frac{2T}{\rho VR^2} = \frac{2T}{mR^2} \text{ (rad.s}^{-1}) \].

(5.9)

Tippelmann further simplified this equation by showing that the angular momentum (\( C_v \)) is proportional to the second power of the volumetric flow rate (\( \dot{V} \)), the air density (\( \rho \)) and a geometrical parameter (\( D^* \)). These are determined from the properties of the steady-flow rig for angular momentum as a function of flow rate, hence

\[ T = \dot{I} = D^* \dot{V}^2 \rho \]  

(5.10)

and from this a dimensionless swirl parameter can be obtained from

\[ D = D^* \frac{R_{sy}}{\dot{V}^2 \rho} \].

(5.11)

The effect of the volumetric flow rate was studied by Tippelmann and it was found that the swirl parameter (\( D \)) can be treated as a constant, provided that the density of the air is assumed to be an average across the inlet valve. This analysis showed that a simple blower fan without accurate control could be used as the air supply because the swirl parameter was treated as a constant and the density averaged across the cylinder.

When considering a cylinder head with two swirl producing ports, the swirl number is a factor of 2 times smaller than a single port head of the same geometry. This is due to the volumetric flow rate passing through each valve. If a cylinder head is considered with two inlet ports (as is the case with this engine) but only one valve is opened so that the complete volumetric flow (\( \dot{V} \)) passes through one valve, then the swirl number is determined by

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If the second valve is opened, only half of the volumetric flow has to be inducted through each of the valves. Therefore, the angular momentum flux for each port becomes

\[ D_i = \frac{T_i R_{cyl}}{V^2 \rho} \]  

(5.12)

and the swirl number becomes

\[ T_1' = T_2' = \frac{D_1 \left( \frac{V}{2} \right)^2 \rho}{R_{cyl}} = \frac{T_1}{4} \]  

(5.13)

and can be further simplified as to represent the widely used 'rule of thumb' for combined inlet ports as

\[ D_{i,2} = \frac{\left( \frac{T_1}{2} \right) d_{cyl}}{V^2 \rho} = \frac{D_1}{2} = \frac{D_2}{2} \]  

(5.14)

In real testing terms, the swirl number for the two port inlet is normally a few percent higher than this theoretical amount, indicating that the two swirling flows can have a positive influence on the total swirl. A full mathematical breakdown of how this swirl was derived is given in Appendix A1.

The 'rule of thumb' in Equation (5.15) for combined inlet swirl ratio is of particular interest to this research project. From the experimentation part of this project the two different types of swirling flow can be visualised. Many modern CI engines employ combination of inlet ports due to experiential results providing preferential emissions when compared to high swirl single ports, as well as increased volumetric efficiencies at higher engine speeds (Benajes et al. 2004; Shimada et al. 1986). It is thought that
in the two inlet port scenario, there is greater turbulence introduced into the air charge, giving a better mixing between the injected fuel and the swirling air charge due to interaction of the generated swirl fields.

These steady-flow rigs do not account for moving pistons and valve gear, which vary the intake air charge greatly. Although a relationship exists between the intake swirl measurement and the processes that occur during compression, it is not yet fully understood since the swirl is greatly altered during the compression stroke and the flow is complex and 3-dimensional.

5.3 Results from steady-flow testing of optical cylinder head

Testing of the optical engine cylinder head was carried out at Perkins Engines to determine the AVL swirl number (on a Tippelmann rig) of the different port arrangements. This test comprised of fitting the cylinder head to an impulse swirl measurement rig employing a honeycomb flow straightening device. A pressure drop was created across the cylinder head and measurements recorded for each valve lift position (in 1 mm intervals). Figure 5.3 shows the optical engine cylinder head fitted to the steady-flow swirl rig. The intake manifold was fitted to the optical cylinder head since it influences the charge air motion on entry to the cylinder.

![Figure 5.3 Optical cylinder head attached to impulse swirl measurement rig](image-url)
Three tests were carried out on the head to determine the static swirl measurement of the head for the configurations of directed inlet, helical inlet and combination of both inlet ports. The mean AVL swirl number and the mean valve port coefficient were determined and the results are tabulated in Table 5.1 below. A mathematical explanation of these is given in the following section.

<table>
<thead>
<tr>
<th>Inlet Configuration</th>
<th>Valve Port Mean Flow Coefficient</th>
<th>Mean AVL Swirl Number (normalised by engine speed)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Directed Inlet</td>
<td>0.3298</td>
<td>3.078</td>
</tr>
<tr>
<td>Helical Inlet</td>
<td>0.3289</td>
<td>3.376</td>
</tr>
<tr>
<td>Combination Inlet</td>
<td>0.2722</td>
<td>1.512</td>
</tr>
</tbody>
</table>

Table 5.1 Impulse steady-flow test rig results for optical engine cylinder head

5.3.1 Dimensionless swirl number

The swirl number is a dimensionless unit determined from the swirl of the air normalised by a nominal engine crankshaft speed derived from a related air flow rate

\[
Swirl Number = \frac{n_d}{n}
\]  (5.16)

where \( n_d \) is the swirl speed (rpm)

\( n \) is the fictitious engine speed gained from equating the mean piston speed with the axial component of velocity (\( C_{ax} \)) which is shown in Figure 5.2.

The axial component of velocity was defined in Equation (5.7) as

\[
C_{ax} = \frac{V}{\pi R^2}
\]
where $V$ is the volume of flow across the cylinder head

$R$ is the radius of the cylinder.

This is then combined with the calculation for mean piston velocity, calculated as

$$C_m = \bar{S}_p = 2sn$$  \hspace{1cm} (5.17)

where $\bar{S}_p$ is the mean piston speed

$s$ is the stroke of the engine in question

$n$ is the crankshaft speed of the engine.

This can be rearranged to determine $n$ as follows

$$n = \frac{C_m}{2s} \text{ (rps)} \text{ or } n = \frac{C_m \pi}{s} \text{ (rad.s}^{-1}).$$  \hspace{1cm} (5.18)

By equating mean piston speed (from Equation (5.18)) with mean axial velocity (from Equation (5.7)) the nominal engine speed ($n$) can be calculated, with the assumption that the positive direction is upwards, the mean piston speed is negative when the axial velocity is positive, hence

$$n = -\frac{V \pi}{V_{cyl}} \text{ (rad.s}^{-1})$$  \hspace{1cm} (5.19)

where $V_{cyl}$ is the swept cylinder volume.

This can then be combined with Equation (5.7) to give a dimensionless unit of swirl by

$$Swirl \ Number = \frac{n_d}{n} = \omega \frac{V_{cyl}}{V \pi} = -\frac{2T V_{cyl}}{n R^2 V \pi}.$$  \hspace{1cm} (5.20)
5.3.2 Mean AVL swirl number calculation

The AVL mean swirl number (Tippelmann 1977) is a method of describing the entire movement of the cylinder charge in a singular number and is based on the dimensionless swirl number described previously with the following assumptions:

- the sum of moments of momentum during the intake process is equal to the moment of momentum of the entire cylinder load
- each moment of momentum can be characterised by the swirl ratio \( \left( \frac{n_d}{n} \right) \) for one particular valve lift
- instantaneous piston speed is taken into account, but the compressibility of the air is neglected.

Therefore, the equation for AVL mean swirl number is given by

\[
\left( \frac{n_d}{n} \right)_{\text{mean}} = \frac{1}{\pi} \int_{\text{BTDC}(\varphi=0)}^{\text{TDC}(\varphi=\alpha)} \frac{n_d}{n} \left( \frac{C_{\text{inst.}}}{C_m} \right)^2 d\alpha \quad (5.21)
\]

where \( \alpha \) is the crank angle (radians)

and \( \frac{C_{\text{inst.}}}{C_m} \) is the instantaneous speed to mean speed ratio and is calculated as follows

\[
\frac{C_{\text{inst.}}}{C_m} = \frac{\pi}{2} \sin \alpha \cdot \sqrt{1 + \frac{\cos \alpha}{\sqrt{s^2 - \sin^2 \alpha}}} \quad (5.22)
\]

Hence the swirl number for each valve lift value can be calculated by combining Equations (5.21) and (5.22) to give

\[
\text{AVL Swirl Number} = \left( \frac{C_{\text{inst.}}}{C_m} \right)^2 \left( \frac{n_d}{n} \right) \quad (5.23)
\]
This allows a swirl number to be obtained for each valve lift condition. However, since cam profiles widely differ according to engine, application and emission controls. The rig manufacturer (AVL) created a non-dimensional cam to compare different cylinder heads. This standard AVL cam profile is described by the following equation

\[ \frac{h_r}{d_v} = -3 \times 10^{-5} (\alpha - 105)^2 + 0.275 \]  \hspace{1cm} (5.24)

where \( h_r \) is the absolute valve lift, \( d_v \) is the inner seat diameter of the valve.

This allows for a standard set of values for non-dimensional valve lift to be obtained. To convert actual valve lifts to these ideal \( \frac{h_r}{d_v} \) ratios, it is necessary to divide them by the inner seat diameter and perform a linear interpolation to calculate the swirl ratio at these specific valve lift points.

The coefficient of the valve port was calculated from the valve diameter and coefficient of discharge of a standard nozzle. The calculations necessary to determine these results are given in Appendix A1. This gives the exact equations used to determine these swirl rates, as well as the specific value for each data point.

Figure 5.4 is a plot showing the three different inlet arrangements (AVL standardised equations for valve lift versus inlet diameter), their respective normalised swirl velocities and the coefficients of discharge (based on mass flow rate) across the inlet valves.
Figure 5.4 AVL and port discharge coefficient for different inlet configurations
Figure 5.4 showed the helical and directed ports produced very similar steady-flow swirl levels until the peak valve lift was reached, where the directed inlet produced slightly more swirl than the helical inlet. The air mass flow rate remained similar for both the helical and directed inlet conditions due to having the same valve area. The combined inlet allowed a higher mass flow rate across the head due to increased valve open area (i.e. two valves open).

This measured swirl and flow data will be used later as the assumed swirl in the engine at BDC for the swirl spin-up model that seeks to predict TDC in-bowl mean swirl, described in the next section.

5.4 Swirl spin-up modelling

Swirl spin-up theories reported to-date have been based on the conservation of momentum calculations. These mathematical models work on the assumption that the flow represents a solid body rotation and cannot account for changes in depth of the bowl or the piston pip.

In many cases, swirl spin-up models are based on data that is gained from testing cylinder heads on steady-flow test rigs. This rate of swirl is then considered as the basis for the swirl spin-up during the compression stroke.

Computational spin-up models can be divided into two main categories, these are:

i. zero-dimensional models

ii. multi-dimensional models.

These two types of modelling are now discussed separately.
5.4.1 Zero-dimensional modelling

The zero-dimensional modelling (also known as ‘phenomenological’ modelling) approach is to divide the cylinder into a number of regions and apply a group of ordinary differential equations for mass, chemical and energy balances with the only variable being time. In order for this type of model to work, a number of assumptions are made which can limit the accuracy of the model. Various models have been previously reported (Derham (1972); Horlock and Winterbone (1986)) that show reasonable agreement with experimental data. Figure 5.5 presents some results of experimental data to zero-dimensional modelling results compared to experimental data obtained by Dent and Derham (1974).

![Figure 5.5](image)

Figure 5.5 Experimental swirl results compared with modelled results (Dent and Derham 1974)

Figure 5.5 showed a good correlation between the zero-dimensional model that incorporated the affects of friction (Derham 1972) with the experimental results. These experimental results were obtained using the hot wire anemometry method with a specially modified piston. If friction was not accounted for in this model, the swirl was over estimated. Models that neglect friction effects are based on the principle of solid body rotation and cannot represent different swirl velocity profiles across the piston bowl.
Although zero-dimensional models are limited in overall accuracy, they can still prove to be a useful tool for the macroscopic processes happening in the cylinder. For example, the results can be compared easily with experimental data and do not require powerful computation.

5.4.2 Multi-dimensional modelling

Multi-dimensional models seek to predict the entire flow region, considering at many points the full temporal and spatial variations of the in-cylinder flow. There are three main parts to this modelling approach reported by Horlock and Winterbone (1986);

i. Partial-differential conservation equations of physics governing the spatial and temporal variation of velocity, including pressure, temperature and viscosity.

ii. Additional equations representing the mathematical models of key sub-processes occurring within the fluid, including turbulent motion, heat transfer, chemical reaction and blow-by.

iii. Computer-based numerical procedures for solving the equations for the circumstances of interest (as the equations are too complex to be solved by analytic solution).

This type of modelling is designed to predict the effects that changes in the input parameters will have on the flow regime within the cylinder at points during the cycle. Multi-dimensional CFD models have become more popular with the rapid advancement of computing power now capable of modelling complex flows in practical time-scales. There are limitations to this modelling in that prediction of the movement of turbulence fields that accommodate for losses such as piston blow-by and cyclic variations cannot be computed. Conventional CFD results of this type of modelling are presented in discrete regions of the cylinder. The results are based on averaged data due to the additional complexity and uncertainty of calculating cycle-to-cycle variations that occur within an engine.
5.5 Zero-dimensional models based on optical research engine geometry

Swirl is generated during the induction stroke (as discussed in Section 2.4) but is significantly modified during the compression stroke. In the instance of bowl-in-piston combustion chamber designs, the tangential velocity of the swirl during the induction stroke is substantially increased during compression by forcing the air into a bowl-in-piston design. This is due to conservation of angular momentum and increases the mixing rates between the air and the injected fuel. The moment of momentum conservation changes with time, and can be described mathematically by the expression

\[
\frac{d\Gamma_c}{dt} = J_i - T_f \tag{5.25}
\]

where \(\Gamma_c\) is the angular momentum of charge
\(J_i\) is the flux of angular momentum into the cylinder
\(T_f\) is the torque due to wall friction.

\(J_i\) is calculated for a point in the flow jet inlet by

\[
J_i = \int_{A_v} \rho v_0 \mathbf{v} \cdot dA_v \tag{5.26}
\]

where \(dA_v\) is an element of the valve open area as shown in the Figure 5.6.

![Figure 5.6 Definition of symbols in equation for angular momentum flux into the cylinder (Heywood 1988)](image)
The angular momentum entering the cylinder during induction can be calculated as

$$\Gamma_{c,i} = \int_{v_{ci}}^{v_{ci}} \int_{A} \rho v_{\theta} r dA dt.$$  \hspace{1cm} (5.27)

This equation for angular momentum does not consider losses due to wall friction which will continue through the compression process. The wall friction of the boundary layers can be modelled using the analogy of friction on flow over a flat plate and estimated for the cylinder wall, cylinder head, piston crown and bowl-in-piston combustion chamber.

The alteration of this swirl during the compression stroke, where the induction charge is forced into the bowl of the piston, can be considered from before BDC of the induction stroke. Considering the flow to be of solid body rotation within the cylinder ($\omega_{s,i}$) with a mass ($m_c$), the initial angular momentum ($\Gamma_{c,i}$) (neglecting the effects of friction) can be related as follows

$$\Gamma_{c,i} = I_c \omega_{s,i}$$  \hspace{1cm} (5.28)

where $I_c$ is the moment of inertia of the charge and can be calculated for a bowl-in-piston by the expression from Heywood (1988)

$$I_c = \frac{m_c B^2}{8} \left[ \frac{(z_p/h_b) + (D_b/B)^4}{(z_p/h_b) + (D_b/B)^2} \right]$$  \hspace{1cm} (5.29)

where $D_b$ and $h_b$ are the diameter and depth of the bowl respectively, and $z_p$ is the distance between the cylinder head and the piston crown. Hence, at TDC, $z_p \approx 0$, therefore the moment of inertia at TDC reduces to

$$I_c \approx \frac{m_c D_b^2}{8}.$$  \hspace{1cm} (5.30)
This approximation can be applied to the end of the induction stroke so that

\[ I_c \approx \frac{m_p B^2}{8}. \]  

Therefore, neglecting friction effects, the solid body rotation would increase by 
\[ \approx \left( \frac{B}{D_b} \right)^2, \] a factor of approximately 4 depending on bowl design. If friction effects are considered this factor falls to approximately 2 to 3 (Heywood, 1988).

At BDC the tangential flow within the bowl velocity increases with increasing radius, apart from close to the cylinder walls where friction affects at the boundary layer to reduce the flow velocity. This flow structure is accelerated ('spun-up') during compression, as discussed previously, due to the effect of angular momentum conservation. Heywood (1988) provides a relationship between the bowl, the engine swept volume and mass of charge air in the bowl, which can be related as follows

\[ \frac{dm_b}{dt} = m_c \left( \frac{V_b}{V} \right) \left( \frac{V_b}{V} \right) S_p. \]  

The assumption that the flow is solid body rotation at the end of induction is an approximation. The flow is typically close to solid body rotation but departs from this with increasing engine speed and different piston configurations. The lip of the piston plays a part in altering the flow.

A further basic zero-dimensional swirl spin-up model is proposed by Horlock and Winterbone (1986). These basic modelling methods did not account for any losses within the cylinder. It is worth noting that the surface area to volume ratio of the cylinder rapidly increases towards TDC and, therefore, the friction effects will have a much greater affect at TDC. These mathematical models are really only for guidance since the friction effects cannot be considered to be negligible. However, this type of mathematical modelling does highlight the different factors affecting the swirl in the engine. The model uses the major dimensions of the cylinder and engine which have
been normalised against a characteristic length, the cylinder bore \((B)\). The basic swirl magnitude can be calculated from

\[
\frac{\omega}{\omega_{nc}} = \frac{z_p^* + d'^* H^*}{z_p^* + d'^2 H^*}
\]

(5.33)

where \(\omega\) is the instantaneous swirl velocity \((\text{rads}^{-1})\)

\(\omega_{nc}\) is the swirl velocity at inlet valve closure \((\text{rad.s}^{-1})\)

\(z_p^*\) is the instantaneous cylinder length \((\text{m})\)

\(z_p^*\) is the length to bore ratio, given by \(z_p / B\)

\(d'^*\) is the piston bowl to bore ratio, given by \(d'^* = D_b / B\)

\(H^*\) is the height of bowl to bore ratio, given by \(H^* = h_b / B\).

The instantaneous cylinder length can be expressed in terms of crank angle \((\alpha)\) by

\[
z_p = G + \frac{s}{2}(1 + \cos \alpha)
\]

(5.34)

where \(G\) is the bump clearance between cylinder head flame face and piston at TDC \((\text{m})\) and \(s\) is the stroke length \((\text{m})\).

By applying this basic mathematical modelling to the engine geometry, the following graph in Figure 5.7 shows the swirl spin-up ratio from BDC compared between the two different piston geometries. This is compared with the piston bowl volume as a ratio of cylinder volume.
It can be seen in Figure 5.7 that the major effect on swirl spin-up ratio is the piston geometry on the piston. As stated previously, Heywood (1988) described an alternative mathematical method for modelling the swirl spin-up from BDC based on the angular momentum at BDC. Again, the main assumption is that the flow is based on solid body rotation. The initial angular momentum is then calculated for BDC as

\[ \Gamma_{c,d} = I_c \omega_{rev} \]  

(5.35)

where \( \Gamma_{c,d} \) is the initial angular momentum

\[ I_c = \frac{m_c B^2}{8} \left[ \frac{(z_p / h_B) + d_{a,4}^2}{(z_p / h_B) + d_{a,2}^2} \right]. \]  

(5.36)

Results obtained using Equation (5.36) are presented in Figure 5.8, where it is shown that the Heywood (1988) method for spin-up calculates a slightly lower value than the Horlock and Winterbone (1986) method. This was attributed to the normalising of the
instantaneous cylinder length (distance from piston crown to cylinder head flame face, $z_p$), whereby, in the Horlock and Winterbone (1986) method, it is normalised by the bore, and in the Heywood (1988) method normalised by the piston bowl height.

![Diagram](image)

**Figure 5.8** Comparison of Heywood (1988) method swirl spin-up ratio and volume ratio

Ideally for spin-up conditions, the reduction of the bowl diameter increases the rotational speed of the flow. However, the bowl depth has to increase to maintain the compression ratio. Therefore, bowls that are of smaller diameter have to be deeper, which is undesirable when considering the fluid process of high penetration fuel injection. Both of the above calculations have been made using the assumption that the bowl is a simple flat bottomed geometry (i.e. not re-entrant or with a central 'pip') due to the complexity of calculating varying depth bowls.
5.6 More advanced swirl spin-up modelling

A progression from the basic swirl modelling presented in the previous section is the momentum model. This incorporates many more factors neglected in the basic modelling method. The additional factors during the compression process are:

i. air state within the cylinder
ii. instantaneous piston speed
iii. kinetic energy of the swirl flow
iv. friction against surfaces
v. turbulence
vi. effect of squish flows
vii. velocity profiles across the bowl.

This type of model was adopted in previous work by Croston and Garner (2001) where a model was developed based upon work by Borgnakke et al. (1981). The model works on the principle that the swirling flow at BDC was the set conditions for the compression phase, but also considers the decay of swirl due to the friction effects arising from the cylinder walls, turbulence and internal fluid shear. Amplification of the swirl during the compression phase is addressed using the theory of conservation of angular momentum and calculations made for each crank angle from BDC to TDC. Angular momentum of the swirl is calculated as

\[ \Omega = I_{cylinder} \omega \]  \hspace{1cm} (5.37)

where \( \Omega \) is the angular momentum of the charge air, \( I_{cylinder} \) is the moment of inertia of the swirl trapped in the cylinder and \( \omega \) is the angular velocity of the swirling flow, based on the assumption of solid body rotation. In real terms, as previously mentioned, the \( \Omega \) decreases by approximately 30% (Heywood 1988) during the compression phase due to friction, turbulence effects and blowby.
From this angular momentum term for the in-cylinder air flow, the rate of change of the angular momentum at any instant during the compression phase is equal to the sum of the torque force acting on it at that time, and is expressed as

\[
d[\mathbf{i}_{\text{cylinder}} \omega] = -T_s \, dt
\]  \hspace{1cm} (5.38)

where \( T_s \) is the torque exerted by the shear forces acting over the cylinder walls, piston faces, cylinder faces and between the fluid layers.

A brief description of these additional factors that account for this resisting torque are now discussed and the actual model utilised for spin-up prediction, written in Matlab programming language, is presented in Appendix A2.

5.6.1 Air State within the Cylinder

The air state is of importance to the swirl spin-up model due to the direct effect that density has on the friction losses. The temperature and pressure of the air during the compression process can be described by the polytropic relationship

\[
\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1}
\]  \hspace{1cm} (5.39)

and

\[
\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\gamma}
\]  \hspace{1cm} (5.40)

where \( T \) is the absolute temperature of the fluid (K)

\( V \) is the volume that the charge is in (m\(^3\))

\( P \) is the pressure of the fluid (Pa)

\( \gamma \) is the specific heat ratio \((C_p / C_v)\).

The isentropic constant, \( \gamma \), for air is normally 1.4, however, to compensate for heat transfer and losses, \( \gamma \) can be assigned a value of 1.35 (Heywood 1988). The subscripts (1 and 2) are representative of the next and previous step in increments of
1° crank angle from BDC to TDC, used to give an accurate calculation of the air at any point in the cycle. Air density is calculated from the standard ideal gas equation, as follows

\[ PV = m R_{\text{gas}} T \]  

which rearranges for density as

\[ \frac{m}{V} = \rho = \frac{P}{R_{\text{gas}} T} \]  

where \( m \) is the mass of air in the cylinder (kg)

\( R_{\text{gas}} \) is the gas molar constant for air (0.287 kJ.kg\(^{-1}\).K\(^{-1}\)).

5.6.2 Instantaneous Piston Speed

The losses incurred on the swirl are directly related to the time taken for each stroke, as the turbulence, friction and squish affect the swirl momentum on a time basis. The time steps need to be on a sufficiently small scale to accurately determine these losses. The instantaneous piston speed was calculated by Heywood as

\[ S_p = \frac{\pi}{2} \sin \theta \left[ 1 + \frac{\cos \alpha}{(s_{\text{ratio}}^2 - \sin^2 \alpha)^{1/2}} \right] \bar{S}_p \]  

where \( \alpha \) is the crank angle (radians)

\( \bar{S}_p \) is the mean piston speed (calculated in Equation (5.17)) (m.s\(^{-1}\))

\( s_{\text{ratio}} \) is the ratio of connecting rod length to crank radius which is 3.504 in the case of the optical engine used in the present study.

This speed is then considered at each crank angle position to be an average, as it is estimated over such a short time frame. Results for this calculation are shown in Figure 5.9. The graph shows the piston speed accelerates to its maximum value at approximately 285°, before decelerating to TDC.
5.6.3 Kinetic energy of the swirl flow

The kinetic energy of the swirling flow can be calculated at BDC by applying the kinetic energy equation for a rotating mass. By considering the volume of fluid in the engine to be a solid with changing density, the kinetic energy of the flow is calculated for each degree crank angle. Hence

$$KE_{\text{rotational}} = \frac{I_{\text{cylinder}} \omega_{\text{rc}}^2}{2}$$  \hspace{1cm} (5.44)

where $KE_{\text{rotational}}$ is the kinetic energy (J)

$I_{\text{cylinder}}$ is the moment of inertia (N.m.s$^{-2}$)

$\omega_{\text{rc}}$ is the angular velocity of the charge at BDC (rad.s$^{-1}$).

The angular momentum of the charge ($\omega_{\text{rc}}$) is calculated from the non-dimensional swirl number that was tested for that port, and multiplied by engine speed to determine the swirl at BDC (rad.s$^{-1}$).
If the moment of inertia \((I_{\text{cylinder}})\) is considered for simple flat bottom geometry bowl, as previously stated, the two areas of the flow can be divided, as shown in Figure 5.10. To calculate the moment of inertia, the following calculation is applied (Ugural 1993)

\[
I_{\text{cylinder}} = \frac{m R_{\text{cyl}}^2}{2} \quad (5.45)
\]

where \(m\) is the mass of the body \((\text{kg})\)

\(R_{\text{cyl}}\) is the radius of the body \((\text{m})\).

In the case of a flat bottomed piston bowl, both cylinders can be considered together by the following relationship

\[
I_{\text{cylinder}} = \frac{\rho \pi \left[ (R_{\text{cyl}}^2 z_p) + (r_b^2 h_b) \right]}{2} \quad (5.46)
\]

where \(\rho\) is the air density, calculated in Equation (5.42) \((\text{kg.m}^{-3})\)

\(R_{\text{cyl}}\) is the radius of the cylinder bore \((\text{m})\)

\(z_p\) is instantaneous cylinder length, calculated in Equation (5.34) \((\text{m})\)

\(r_b\) is the radius of the piston bowl \((\text{m})\)

\(h_b\) is the height of the piston bowl \((\text{m})\).

Equation (5.46) accounts for the changing density of the air and the spin-up in the bowl as the piston approaches TDC.

### 5.6.4 Friction against surfaces

Borgnakke et al. (1981) suggested that the cylinder and bowl be considered as two separate volumes in relation to the friction affects of the air motion. The distinction of these two regions is shown in Figure 5.10.
This method of separating the cylinder into two distinct parts allowed for measuring the mass flow between the volumes providing a more accurate prediction of swirl amplification and squish flow. Hence the friction (considered as resisting torques) can be expressed as follows:

\[ T_{s1} = 2\pi z_p (R_{cyl}^2 \tau_w(r) - r_b^2 \tau_{air}(r_b)) + 4\pi \int_0^R r^2 \tau_w(r) \, dr \]  
\[ T_{s2} = 2\pi r_b^2 (h_b \tau_w(r_b) + z_p \tau_{air}(r_b)) + 4\pi \int_0^R r^2 \tau_w(r) \, dr \]  

where \( T_{s1} \) and \( T_{s2} \) are the restraining torques due to friction in volumes 1 and 2 (N.m)

- \( z_p \) is the cylinder length from cylinder head to the piston crown (m)
- \( h_b \) is the height of the piston bowl (simple flat bottom geometry) (m)
- \( R_{cyl} \) is the cylinder radius, \( B/2 \) (m)
- \( r_b \) is the radius of the piston bowl (m)
- \( r \) is the discrete radius of the cylinder (m)
- \( \tau_w \) is the shear stress at fluid-solid interfaces (kg.m\(^2\).s\(^{-2}\))
- \( \tau_{air} \) is the shear stress between adjacent fluid layers (kg.m\(^2\).s\(^{-2}\)).
The first term in Equations (5.47) and (5.48) calculates the shear stresses along the cylinder walls and fluid layers. The second term is the shear stress over the piston and cylinder head. The integral term in each equation represents the increase in velocity with radius, and therefore increased shear stress. These shear stresses can be calculated as

\[ \tau_w = C_1 \frac{1}{2} \rho V_o(r)^2 \text{Re}^{-C_2} \]  

(5.49)

where \( \rho \) is the fluid density (kg.m\(^{-3} \))
\( V_o(r) \) is the tangential velocity of the fluid (m.s\(^{-1} \))
\( \text{Re} \) is the Reynolds Number.

The Reynolds number is considered as a flat plate that uses the empirical constants \( C_1 \) and \( C_2 \) (0.055 and 0.2 respectively (Munson et al. 1994)) that allow for the difference between flat plate geometry and cylindrical geometry. It is given by

\[ \text{Re} = \frac{\rho V_o(r)r}{\mu} \]  

(5.50)

where \( \mu \) is the dynamic viscosity of the fluid (N.s.m\(^{-2} \)).

The value for \( \mu \) is temperature dependent only and can be approximated for air at each crank angle by the Sutherland relationship (Munson et al. 1994) for air, such that

\[ \mu = \frac{C_{\text{sutherland}} T^{1.5}}{T + S_{\text{sutherland}}} \]  

(5.51)

where \( T \) is the absolute temperature calculated at each crank angle (K).
\( C_{\text{sutherland}} \) and \( S_{\text{sutherland}} \) are empirical constants which can be determined for the temperature range of interest, in this instance between 25°C and 500°C.

The constants for this range are: \( C_{\text{sutherland}} = 1.52 \times 10^{-6} \) and \( S_{\text{sutherland}} = 124.856 \).
The shear stress between the adjacent fluid layers is omitted from this model due to the complexity of defining the turbulent eddy diffusivity term and its dependence on turbulent length scales. However, it is important to have an understanding of these shear stress calculations for adjacent fluid layers, as the eddy diffusivity term will be repeated, and again omitted, from the squish terms. Hence, the shear stress between adjacent fluid layers is calculated by

\[ \tau_{sr} = \rho \nu_t \left( \frac{\partial V_\theta}{\partial r} \frac{V_\theta}{r} \right) \]  

where \( \nu_t \) is the turbulent viscosity which is calculated by

\[ \nu_t = \frac{C_p k^2}{\varepsilon} \]  

where \( k \) is the turbulence kinetic energy (m².s⁻²)

\( \varepsilon \) is the dissipation rate of turbulence kinetic energy (m².s⁻³).

\( C_p \) is a constant of 0.09 based on work of Launder and Spalding (1972) as cited by Borgnakke et al. (1981).

Borgnakke et al. (1981) based this eddy diffusivity as an average decay term integrated over the combustion chamber volume and takes into account the inlet turbulence created at the valve during intake by using the inlet velocity and the geometric details of the open valve area. This decay term is specific to the design of inlet port geometry and is considered to be outside the scope of this work.

5.6.5 Turbulence

When considering turbulence in this model, the terms have to be simplified. Equation (2.10) used previously is a simplified form

\[ U(\theta, j) = \tilde{U}_{ex}(\theta) + \tilde{U}_{cv}(\theta, j) + u(\theta, j) \]  

When modelling, it is difficult to account for individually fluctuating components with the chaotic nature of the flow. Borgnakke et al. (1981) therefore suggested
modelling the turbulent losses using the viscosity of the fluid. To do this a $k-\varepsilon$ model was adopted which links the kinetic energy ($k$) and dissipation rate ($\varepsilon$) of the turbulent flowfield, and is expressed as

$$\frac{dk}{dt} = P_k + D_k - \rho\varepsilon \tag{5.54}$$

where $P_k$ is the volumetric production of turbulence and $D_k$ is the gradient diffusion of the turbulence.

This can be integrated over the combustion volume chamber to provide spatially averaged turbulence predictions. Terms $P_k$ and $D_k$ represent the transportation of kinetic energy over the system boundaries, such as through the intake or exhaust valves, or losses through the piston rings, etc. The dissipation rate $\varepsilon$ is related to the integral length by

$$\varepsilon = \frac{C_D k^{3/2}}{l_I} \tag{5.55}$$

where $C_D$ is the discharge coefficient (a constant of 0.09 (Heywood 1988))

$$l_I$$ is an integral length scale that is approximated as $l_I = 0.22 L_v$

(function of valve lift (Heywood 1988)).

Tabaczynski (1990) developed this work further and showed that Equation (5.54) can be approximated as

$$\frac{dk}{dt} = \frac{2k}{3} \frac{d\rho}{dt} - \varepsilon \tag{5.56}$$

where the first term represents the strain produced by the piston motion (which is directly related to density) and $\varepsilon$ is the only term for dissipation. Hence, by substituting expressions for $\varepsilon$ and other known factors, the following equation is obtained
\[
\frac{dk}{dt} = \frac{n}{45} k \ln(CR) - \frac{120k^2}{(s^2 \xi^2 n)} \tag{5.57}
\]

where $\xi = 1$ for $s < B$; $\xi = \frac{B}{s}$ for $B < s$

$k$ is the initial turbulence kinetic energy

$n$ is the crankshaft speed (rpm)

$s$ is the stroke (m)

$B$ is the bore (m)

$r_c$ is the Compression Ratio.

The value for turbulence kinetic energy ($k$) is considered to be approximately 30% of the total kinetic energy at BDC (Borgnakke et al. 1981; Heywood 1988; Tabaczynski 1990).

The Tabaczynski turbulent energy model also incorporated production terms for swirl and squish during the compression phase, based on the principle that the viscous forces between fluid-fluid and fluid-solid interfaces was a source of turbulence production. The squish effect as the piston approaches TDC becomes greater, as the radial squish flow is perpendicular to the tangential swirl flow. This is assumed to result in turbulent production in the air charge. Tabaczynski (1990) went onto calculate these production terms for the squish and swirl turbulent production per unit mass as

\[
P_{\text{squish}} = (0.1v_{\text{squish}})^2 \tag{5.58}
\]

and

\[
P_{\text{swirl}} = v_t \left( \frac{\partial V_s}{\partial r} - \frac{V_t}{r} \right). \tag{5.59}
\]

Equation (5.59) is almost identical to the calculation for shear stress between adjacent layers that was given in Equation (5.52), and as it is based on the turbulent eddy diffusivity term. As previously discussed, this was omitted from the present model. Equation (5.58) is to be used in the model and can be added to Equation (5.57) to give a total turbulence kinetic energy at each crank angle.
The production of turbulence kinetic energy is represented in Figure 5.11 for a mathematical modelled engine running at 800, 1200 and 1600 rpm utilising a deep bowl geometry, with combined inlet port operation, giving a swirl ratio of 1.512. The initial turbulence kinetic energy is calculated from the assumption that it is 30% of the total swirl kinetic energy at BDC. The turbulence kinetic energy can be directly related to the turbulence intensity (Equation 2.11) by the following relationship, assuming isotropic turbulence (Turns 1996).

\[ k = \frac{3}{2} u^2 m_c \]  (5.60)

The turbulence intensity is plotted with the turbulent kinetic energy in Figure 5.11.

![Figure 5.11 Production of turbulence kinetic energy and turbulence intensity during compression stroke](image)

In Figure 5.11 it is shown that the kinetic energy production (and hence turbulence intensity) is greater than the dissipation rate, resulting in an increase of the total kinetic energy. This increase was due to the additional energy imparted to the air charge from the piston motion during compression phase. With increased initial
kinetic energies, the production of turbulence kinetic energy was seen to reduce until equilibrium between the kinetic energy and dissipation was achieved. The equilibrium value for this geometry engine was seen to occur at approximately 3.4 J, which is an equivalent kinetic energy term of the rotating air charge at 21000 rpm at BDC. This rotating speed was much greater than the swirl generated in this engine, where a maximum of 7000 rpm at BDC could be expected. This indicated that the turbulence kinetic energy will increase throughout the stroke, as shown in Figure 5.11. Tabaczynski (1990) suggested that the turbulence is a characteristic of the flow, which implies that the turbulent energy field alters according to the initial conditions.

5.6.6 Squish Flows

Squish flows were introduced in Chapter 2 and Equation (2.3) used to calculate the radial inward motion of the charge as it approaches TDC as

\[
v_{\text{squish}} = \frac{D_B}{4z_p} \left[ \left( \frac{B}{D_B} \right)^2 - 1 \right] \frac{V_B}{A_z} \frac{z_p}{z_p + V_B} S_p \quad (2.3)
\]

where \( V_B \) is the volume in the piston bowl
\( A_z \) is the cross-sectional area of the cylinder
\( z_p \) is the distance between the piston crown and the cylinder head
\( D_B \) is the diameter of bowl
\( B \) is the diameter of the cylinder bore
\( S_p \) is the instantaneous piston speed.

The factor in this equation which had the main controlling effect on the squish velocity was the bump clearance (the distance between the piston crown at TDC and the cylinder head flame face). The affect of altering the bump clearance with respect to the squish velocity is presented in Figure 5.12 where other engine parameters were maintained. For this calculation, the assumed engine speed is 800 rpm, using a deep bowl piston.
The standard bump clearance of the optical engine used in this work is 0.7-0.8 mm, which showed a peak squish velocity of 17 m.s\(^{-1}\) at 7° BTDC. This confirmed Heywood (1988) and Horlock and Winterbone’s (1986) prediction that maximum squish velocities occur approximately 10° before TDC. It is also reported that although the squish turbulence had an effect on the swirling flow, the lower part of the flow in the piston remained similar to a solid body rotation.

5.6.7 Velocity profiles across the bowl

In the zero-dimensional modelling techniques, one of the main assumptions of air flow behaviour was that it approximated solid body rotation. In work presented by Borgnakke et al. (1981), three different velocity profiles were presented to account for the deviation of the flow from solid body rotation. These were incorporated into the momentum model. These three velocity profiles are shown in Figure 5.13.
The velocity profiles are based upon previous measured results which showed deviation from the solid body rotation assumption. These velocity profiles are governed by quadratic and a cubic equation which change according to the total angular momentum of the flow. This resulted in a developing profile which represented the change of mass distribution in the cylinder. All the velocity profiles showed a close agreement with solid body rotation and none had a no-slip condition at the wall of the piston bowl. These profiles are based on the assumption that the velocity followed a quadratic velocity profile up to a radius where the velocity was at a maximum (termed as $R_m$) and from $R_m$ to the cylinder wall, the velocity profile was described by a cubic profile. The three different profiles will now be presented.

5.6.7.1 Velocity Profile 1

Velocity profile 1 is a negative quadratic according to the result of

$$V = ar^2 + br \quad \text{for } 0 \leq r \leq R_m$$  \hspace{1cm} (5.61)

with the local maximum occurring at $R_m$.

The cubic part of the profile is given by

$$V = cr^3 + dr^2 + er$$  \hspace{1cm} (5.62)
for the radius from $R_m < r < R$.

The boundary conditions for the velocity profile are assumed as

i. $\frac{dV}{dr} = 0$ at $r=R_m$

ii. $V = V_{\text{max}}$ at $r=R_m$

iii. $V = 0$ at $r = 0$.

From these boundary conditions, the coefficients $a$ to $e$ can be calculated resulting in the following equations for the quadratic and cubic profile respectively

$$V = \left( -\frac{V_{\text{max}}}{R_m^2} \right) r^2 + \left( \frac{2V_{\text{max}}}{R_m} \right) r \quad \text{for condition} \quad 0 < r < R_m \quad (5.63)$$

and the constants for the cubic are as follows

$$c = \frac{d(2R_m - R)}{(R^2 - 3R_m^2)}$$

$$d = \frac{V_{\text{max}}}{\chi(-2R_m^2 - R_m^2)} \quad \text{where} \quad \chi = \frac{2R_m - R}{R^2 - 3R_m^2}$$

$$e = -R_m(3cR_m + 2d).$$

These coefficients are calculated for the velocity profile between $R_m < r < R$.

5.6.7.2 Velocity Profile 2

Velocity profile 2 is a positive quadratic according to the result of

$$V = ar^2 \quad \text{for} \quad 0 \leq r \leq R_m \quad (5.64)$$

with the local maximum occurring at $R_m$.
The cubic part of the profile is given by

\[ V = c r^3 + d r^2 + e r \]  \hspace{1cm} (5.62)

for the radius from \( R_m < r < R \).

The boundary conditions for the velocity profile are assumed as

i. \( V = V_{\text{max}} \) at \( r = R_m \)

ii. \( V_{\text{max}} = a R_m^2 \)

\[ \therefore a = \frac{V_{\text{max}}}{R_m^2}. \]

From these boundary conditions, the coefficients \( a \) to \( e \) can be calculated resulting in the following equations for the quadratic, and the same cubic part as calculated in Velocity Profile 1:

\[ V = \frac{V_{\text{max}}}{R_m^2} r^2 \]  \hspace{1cm} for condition \( 0 < r < R_m \).

5.6.7.3 Velocity Profile 3

Velocity profile 3 is a negative quadratic according to the result of

\[ V = a r^2 + b r \]  \hspace{1cm} for \( 0 \leq r \leq R_m \]  \hspace{1cm} (5.65)

with the local maximum occurring at \( R \).

The cubic part of the profile is given by

\[ V = c r^3 + d r^2 + e r \]  \hspace{1cm} (5.62)

for the radius from \( R_m < r < R \).
The boundary conditions for the velocity profile are assumed as

i. \( \frac{dV}{dr} = 0 \) at \( r = R \)

ii. \( V = V_{\text{max}} \) at \( r = R_m \)

iii. \( V = 0 \) at \( r = 0 \)

iv. \( \frac{dV}{dr} = 2ar + b = 0 \) when \( r = R \)

\[ \therefore b = -2aR \]

and at \( V_{\text{max}} = aR_m^2 - 2aRR_m = a(R_m^2 - 2RR_m) \)

\[ \therefore a = \frac{V_{\text{max}}}{R_m^2 - 2RR_m}. \]

From these boundary conditions, the coefficients \( a \) to \( e \) can be calculated resulting in the following equation for the quadratic equation

\[
V = \left( \frac{V_{\text{max}}}{R_m^2 - 2RR_m} r^2 \right) - \left( \frac{2RV_{\text{max}}}{R_m^2 - 2RR_m} r \right)
\]

for condition \( 0 < r < R. \)
5.7 Summary

This chapter has introduced the concept of modelling the in-cylinder flow that is experienced in the engine during the compression stroke. A detailed discussion of the steady-flow test rigs and how the rate of swirl for a particular cylinder head is calculated was presented. Steady-flow test rigs are still considered to be the industry standard for defining the swirl level in an engine and are still the basis from which many modelling techniques are built. The most basic zero-dimensional modelling techniques were shown and the results for spin-up given. A more comprehensive momentum model was introduced which accounted for many factors that the zero-dimensional models omit. The various calculations for this momentum model have been discussed and have been programmed into Matlab programming language (this Matlab program is detailed in Appendix A2). Each of the 18 engine test point conditions used in this project were modelled. The results of this momentum modelling technique will be compared with PIV data that were measured in the engine at TDC and compared to the measured PIV results in Chapter 7.

The following chapter will consider the experimental set-up of the engine in order to obtain PIV measurements in a motored CI engine. The alterations that have been made to the engine to allow for optical access into the piston bowl will be discussed in detail.
Chapter 6 Experimental Set-up

6.1 Introduction

Examples of the use of the PIV measurement technique with relation to in-cylinder flows were discussed in previous chapters. With respect to CI engine it was noted that the main problem in gaining information of in-cylinder flows are the modifications required allowing for good optical access into the engine cylinder. Whilst square cylinders and pistons, lower compression ratios, mirrors in the cylinder and flat top pistons have all been employed, they are limited since they obstruct the flowfield or do not accurately represent the geometry found in production engines. Also, due to the introduction of optics within an engine, much of the research was carried out using lower than standard compression ratios and only considered lower engine speeds.

The aim of this research project was to investigate the in-cylinder flows at TDC in a standard geometry engine with typical engine operating speeds (800, 1200 and 1600 rpm), different bowl configurations (shallow and deep bowl geometries) and various inlet configurations (directed swirl, helical swirl and a combination of both). In this chapter the optical engine design will be described.

Details of the PIV system will be discussed and an explanation of how it was applied to this work to enable measurement of the in-cylinder flows. Setting up of the PIV system will also be described and how the timing for the firing of the PIV system was achieved.

Further to this, the choice of the seeding medium is investigated as are the different seed materials that were suitable for the visualisation of the in-cylinder flows. It is also shown how the seed was introduced into the air flow
6.2 Optical engine

A number of components on the engine have been altered to allow optical access to the engine. The main block of the engine was a Perkins Trailblazer research single cylinder engine, with a bowl-in-piston combustion chamber. The basic specification is given in Table 6.1.

<table>
<thead>
<tr>
<th>Swept Volume</th>
<th>0.997 litres</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compression Ratio</td>
<td>16.4:1</td>
</tr>
<tr>
<td>Stroke</td>
<td>127 mm</td>
</tr>
<tr>
<td>Bore</td>
<td>100 mm</td>
</tr>
<tr>
<td>Max. Engine Speed</td>
<td>2000 rpm</td>
</tr>
<tr>
<td>Bowl Volume</td>
<td>45.72 cm³</td>
</tr>
</tbody>
</table>

Table 6.1 Perkins Trailblazer engine details

In Figure 6.1 the layout of the engine can be seen:

![Image of engine layout](image)

Figure 6.1 Optical engine layout (set-up here for high speed flow visualisation)
Optical access was provided in the cylinder head, the piston and the cylinder bore. For clarity, each individual component modification for optical access will be discussed separately. Appendix A4 details the upfit and testing procedure employed prior to PIV experiments to ensure integrity of the optical engine.

### 6.2.1 Optical Cylinder Head

Figures 6.2 a and b are photographs of the optical cylinder head. One of the exhaust valves was removed and the injector port used to provide maximum window area into the bowl of the piston and realisation of the geometric centre of the piston pip. The top of the cylinder head was simply cut out to allow for maximum viewable area of the window on the pressure face of the cylinder head. The window encompasses the view field from the centre of the piston bowl (and surrounding on a 20 mm diameter) to the edge of the bowl in the piston. Whilst this window did not allow for total imaging of the flow, it allowed a full 60° segment of the flow, from which the rotating flow within the piston bowl could be inferred.

The window used in the cylinder head is shown in Figure 6.3a. This comprised of a quartz window set into an alloy housing using epoxy resin which was then sealed into the cylinder head using an o-ring seal between the window and the machined cylinder head. The face of the window is set so it is flush with the cylinder head flame face and was held in place using four mounting bolts. The window had a minor
magnification effect on the image area that it was recording. However, the magnification effect was taken into account when creating a calibration image of the flowfield (the segment that the laser would illuminate). Figure 6.3b shows the layout of the cylinder head window when viewing down through the cam carrier into the engine, with dimensions showing the approximate viewable area.

Figure 6.3a Cylinder head window    Figure 6.3b Layout of the cylinder head window view into the piston bowl

The cylinder head had two inlet ports to the engine, a helical and a directed swirl inlet. Inlet configurations that were used in this work are helical only swirl, directed only swirl and a combination of both. The data for these swirl regimes for the different inlet configurations can be seen in Chapter 5, where the directed and helical inlets were hand fettled to return very similar swirl values for steady-flow swirl rig testing. This ensured that an accurate comparison of the effect of different swirl configurations could be made. The activation and deactivation of each valve to allow for different inlet configurations was achieved by installing, or not, the rocker arm to that particular valve during assembly of the engine.
6.2.2 Cylinder access

Access to the cylinder bore was required to form a flat laser sheet within the combustion bowl. The engine was kept standard with regards to the bottom end. The optical access was provided by a 25 mm thick optical quartz ring (see Figure 6.4). It was not possible to take undistorted images through this ring due to the thickness of the glassware. The 25 mm thickness was necessary due to the 16.4:1 compression ratio within the engine.

The quartz ring acted as an optically accessible spacer upon the cylinder. The original blank piston (no bowl machined, tapped holes and recess for optical cassette) was mounted to the small end in the engine and ‘cassettes’ were added for optically accessible piston bowls and to increase the height of the piston by 25 mm so that the overall compression of the engine was maintained. There is a relatively small increase in crevice volume (approximately 3 cc increase) which lowered compression ratio from a typical 17.1:1 to 16.4:1. This was accounted for and the exact compression ratio was calculated.
6.2.3 Piston cassettes

Since the region of interest was in the combustion bowl of the piston, it was necessary to provide optical access for the laser sheet through the piston bowl wall. As previously mentioned, a cassette arrangement was bolted to the top of a blank piston. This enabled the compression ratio to be defined by the design of the piston cassette. In this instance the engine was kept to standard specification.

Two types of optical piston cassette were manufactured for the engine. The importance of different piston bowls was discussed in Chapter 2, and the different effect on bulk swirl and squish effect will be considered during experimentation. Images of the two piston bowls can be seen below in Figure 6.5 a and b.

![Shallow bowl-in-piston with optical access](image1)

![Deep bowl-in-piston with optical access](image2)

Figure 6.5a  Shallow bowl-in-piston with optical access  
Figure 6.5b  Deep bowl-in-piston with optical access

These piston cassettes are divided into three main parts (as shown in Figure 6.6) are the CAD exploded views of the pistons.
The piston bowl was manufactured from the same grade aluminium as the standard piston, the glass was manufactured from quartz silicone glass and the piston top was manufactured from titanium plate, since the strength required at the lid is to hold the glass and piston top to the piston bowl during operation. Volume of the bowl was 45.72 cc in both instances. Crevice volume increased due to the piston cassettes. However, as one of the exhaust valves had been removed, the bump clearance for one of the exhaust valves was not machined into the piston top in order to reduce the crevice volume. This resulted in a compression ratio of 16.4:1 in the final set-up. The piston cassettes were anodised to a matt black finish so as to reduce the light flare from stray light when illuminating the flow with the laser. Stray and reflected light can falsify results since it illuminated particles that are out of plane, or illuminated areas of the flow brighter than the displaced light from the particles. The engineering drawings of the piston bowls, glassware and piston tops can be found in Appendix A3.
Figure 6.7 Optical arrangement of engine

Figure 6.7 illustrates the plane of interest where PIV measurements were taken at a position 1 mm above the piston pip. This plane was chosen as it allowed full illumination of the interrogation area (not obstructed by the piston pip) at TDC, whilst not reflecting any laser flare off the piston pip.

6.2.4 Cam carrier

In the original single cylinder engine, the camshafts ran inside the alloy cam carrier on a bed of lubricating oil from the oil pump. Tappet clearances were maintained by hydraulic pressure from the pressurised lubricating oil. Lubricating oil that leaked by the tappets and camshafts drained down onto the top side of the cylinder head, where it created a bed of oil which lubricated the valves, rocker arms, valve springs and then drained back to the oil return. Since optical access was required, it was necessary to remove this bath of oil from the valve gear to avoid the possibility of leakage onto the top side of the cylinder head quartz window. A new cam carrier was designed and manufactured, which incorporated needle roller bearings (run on grease) as the main bearing for the camshafts. The tappets were mechanically shimmed to the correct clearance after the cam carrier was fitted to the cylinder head and the rocker gear fitted. Rollers on the rocker arms were lubricated with grease prior to running. Since
the engine was only motored for short time periods and was a non-firing engine, the need for lubrication was minimal when compared to a normal running engine. This redesign allowed much greater optical access to the cylinder head quartz window for a camera vertically mounted over the engine. Figure 6.8 is a view from above the cam carrier, with valve gear and cylinder head in place, to show the view into the combustion chamber.

The cam from the redundant exhaust valve was ground back to shaft diameter so there was no interference of the image from the camshaft. Further alterations to this design were to improve the method of timing the valve gear. The original method of timing the engine involved pinning the camshafts into the cam carrier, the piston would be set to TDC and the gear wheels on the end of the camshafts were free to turn. The cam belt was then tensioned and the gears pulled into the correct position. The gears would then be torqued on the end of the camshaft, thus relying on friction drive between the gears and camshafts. Since the pinning was no longer possible due to the needle roller bearings, an alternative method was found by using a timing plate. Figure 6.9 shows in schematic form the exploded view of the cam carrier.
It was necessary to machine flats onto the camshafts (at a precise location relating to TDC). A plate was fitted between the two camshafts of the correct size to stop rotary movement during timing set-up. This also prevented mistakes of incorrect timing during the rebuild of the engine.

6.2.5 Additional alterations

The piston cooling jet (oil spray below the piston) was removed and blanked to reduce oil carry over into the combustion chamber. The oil sump was separate to the engine and a 3-phase motor drove the oil pump. The oil pressure was controlled by a relief valve after the oil pump which returned to tank. The engine was driven with a 3-phase motor with a variable speed controller connected by a belt drive to the flywheel side of the engine. All these items were bolted rigid to a common bed.

6.3 Timing box

It was necessary to accurately control the point in the cycle that the PIV measurements were taken. Therefore the engine was fitted with an incremental shaft encoder on the crankshaft and a Hall effect sensor on the camshaft. The Hall effect
sensor measured a half metal disc to determine when the engine was on the compression and power strokes. This crankshaft encoder generated 1800 pulses per revolution, which was fed to the timing box to give a resolution of 0.2° crank angle. The encoder zero point was manually set to TDC by use of a dial test indicator and the encoder reader manually adjusted to suit. The function of this timing box was to generate an electrical pulse signal to the PIV synchroniser at TDC on the compression stroke, the timing box allowed for many more features of measurement than simply pulse generation. Timing of the pulse could be altered by defining the angle of interest that the trigger was fired. The layout of the timing box is shown in Figure 6.10.

![Timing box with external trigger outputs](image)

**Figure 6.10** Timing box with external trigger outputs

### 6.4 PIV Arrangement

This section describes the optical measurement equipment used in the experimentation on this engine, which is a commercial DPIV set-up compiled by TSI Inc. The PIV system consisted of an Nd:YAG laser, pulse synchroniser unit, CCD camera and frame grabbers controlled by a computer, in conjunction with software (TSI Insight 3), to control, image and process the data. The individual components of this PIV set-up are now described.
6.4.1 Laser

A 'New Wave' Research Mini Nd:YAG flash-lamp pumped laser was employed comprising of a dual-laser with integrated beam optics to supply the illumination light to the region where flow measurements were taken. The laser was capable of firing up to 100 mJ light energy pulses in 4 ns pulse duration, which is powerful enough for the majority of PIV applications. The laser produced coherent light at a wavelength of 532 nm operating on 10 Hz repetition frequency. The pulse separation was not limited since it had two independent lasers, allowing the pulse separation to be user controlled without losing power between laser pulses. The power of each laser pulse was computer controlled by the TSI Insight software, via the ‘Q-Switch delay’. It was possible to control the laser power by the ‘Flash-Lamp Energy Control’, but it was preferable to run the flash-lamps at full power to achieve the best beam quality. The Q-Switch delay controlled when the laser pulse was released, by reducing the Q-Switch delay the beam will not have achieved full power before being released. Whilst this method does not allow the user an exact knowledge of the power of the laser pulse, it was an accurate way of creating the correct intensity light sheet by inspection of the image created.

The laser releases the pulse of light as a 3 mm diameter beam and sheet forming optics were mounted to the front of the laser to create a flat sheet of light at the area of interest. A cylindrical lens was used to control lightsheet width divergence angle and a spherical lens to control the lightsheet thickness. At the measurement region, if the correct lens configuration was chosen, the lightsheet gave the thinnest part of the lightsheet, commonly known as the 'waist'. This was controlled by the focal length of the spherical lens and either side of this waist the beam would diverge. Previous experiments have shown that for good PIV results that the light sheet at the measurement region should be less than 1 mm thick.

6.4.2 Digital camera

The DPIV camera used was a TSI PIVCAM 10-30, a 1000 x 1016 pixel camera with the two frame capture mode. The camera employed a 'frame straddle' technique
enabling the two images with short time separation to be recorded independently. Frame straddle technique is best shown on a timeline (presented in Figure 6.11). The synchroniser created a pulse delay with the camera, enabling the first laser pulse to expose the end of the first camera frame. The second laser pulse then exposed the second image to the frame grabber at the start of the second frame, allowing for very small time separations between the pulses. The images captured on the CCD chip were recorded by frame grabber cards in the PC and saved to either the random access memory (RAM) or to a Hard Disk Drive (HDD). The time taken to write to the HDD is substantially longer than writing to RAM, hence the PIV system automatically skipped a shot if the frame grabber was still writing to disk. Whilst saving to RAM was the quickest method, it was limited by the amount of RAM available in the PC, in this instance a maximum of 200 image pairs. The pulse separation achievable with this PIV system was down to 0.3 µs. However, in the instance of this CI engine, the lowest time separation used was 5 µs.

Figure 6.11 Schematic of timing diagram for Nd:YAG laser and digital camera for PIV data acquisition in triggered twin frame straddled mode (Jarvis 2003)
6.4.3 Synchroniser

The synchroniser used in this PIV set-up was a TSI Model 610034. This synchroniser is the PIV system’s timing and control module, its purpose being to link the computer, frame grabber, camera, laser and external trigger. The synchroniser enabled the system to be completely computer controlled via a serial interface to the PC. The external engine timing box provided an external trigger to the synchroniser at the correct crank angle position that the PIV images were to be taken for measurement of the flow at TDC. The synchroniser was controlled through the TSI Insight software.

6.5 PIV Image Evaluation

Once images for the area of flow were recorded, it was necessary to extract the velocity data held within these image pairs. This process is commonly called the image evaluation technique and utilised the digital cross-correlation method. The advancement of CCD camera technology coupled with the processing speed of computers and storage capabilities, has resulted in the fully digital PIV evaluation technique becoming the most common technique. The cross-correlation method is considered to be superior to the auto-correlation technique for the following reasons (Keane 1994):

i. The size and location of correlation peak is larger
ii. The seeded air flow concentration is lower
iii. An increased dynamic range of the velocity measurements
iv. The velocity gradient bias is minimal.

As described previously in this experimental set-up, a digital camera was employed to capture two separate images of the flow, thus allowing the use of the cross-correlation method of PIV. Cross-correlation was performed on the exposed image pairs, which are normally obtained from double frame PIV recording. The first image is recorded by illuminating the particles in the flow at time \( t \), and subsequently the second image recorded at a short known time interval later, at time \( t + \delta t \). The average velocity of the DPIV image is then computed by extracting small areas (interrogation regions)
from the image and analysing them individually. This was accomplished with the use of a discrete cross-correlation routine, which is given by

$$R_{II}(x, y) = \sum_{i=-k}^{k} \sum_{j=-l}^{l} I(i, j) I_1(i + x, j + y)$$  \hspace{1cm} (6.1)

where $R_{II}(x, y)$ is the cross-correlation of the intensity distribution $I$ and $I_1$.

$I(i, j)$ is the initial interrogation window light intensity distribution at time $t$.

$I_1(i + x, j + y)$ is the second interrogation window light intensity distribution at time $t + \delta t$.

Keane and Adrian (1992), Keane (1994) and Raffel et al. (1998) have highlighted that instead of computing $R_{II}(x, y)$ directly in the spatial domain, a Fast Fourier Transform (FFT) can be performed reducing computation time. An example of this was shown in Figure 3.5 and will be repeated here for reference.

Figure 3.5  Digital cross-correlation method using a Fourier transform  
(Raffel et al. 1998)

When the digital method of image recording is employed, the particles (illuminated in the flow) are recorded to the CCD camera exhibit pixellation. As each pixel is a square element on the CCD chip, the greyscale can be seen. An example of the pixellation of an image is given in Figure 6.12 from an image recorded in the engine.

Further reading of the cross-correlation technique, the peak search and power spectrum analysis can be found in Raffel et al. (1998).
6.5.1 Diffraction limited imaging of particles

When a particle is imaged through a spherical lens with a circular aperture, the image that is imprinted onto the CCD chip or film does not appear as a point in the image plane but forms a Fraunhofer diffraction pattern, as shown in Figure 6.13. This diffraction phenomenon is due to the source of the light and the screen on which the particle image is observed, which is effectively infinite distance from the aperture. The image of the particle on the recording media shows a bright central spot, known as the Poisson spot, surrounded by fainter concentric circles known as Airy rings.

This circular pattern is known as the Airy disk pattern and the intensity distribution of this can be quantified using a Fourier Transform of the aperture’s transmissivity distribution. With regard to the scaling theorem, it can be shown that large aperture
diameters correspond with small Airy disks and small aperture diameters with large Airy disks (Raffel et al. 1998).

The Airy disk pattern can be shown mathematically by the Intensity Distribution, \( I(r) \), to be the square of the 1st order Bessel function, \( J_1(r) \), given as (Goodman 1996)

\[
I(r) \propto \left( \frac{J_1(r)}{r} \right)^2. \tag{6.2}
\]

This Bessel function can be more easily approximated as a Gaussian assumption to a reasonable accuracy as can be seen in Figure 6.14.

![Figure 6.14 Bessel Function compared to Fourier Transform (Anandarajah 2005)](image)

As previously discussed in Section 3.6, the importance of the seed particles following the flow faithfully yields an accurate PIV result. With respect to imaging the particle, the particle image diameter, \( d_i \), is not only dependent on the particle size, but also the properties of the imaging optics, with the image of the particle that is recorded \( (d_i) \) being larger than the diameter of the particle \( (d_p) \). This can be mathematically described as

\[
d_i = \sqrt{d_g^2 + d_d^2} \tag{6.3}
\]

where \( d_d \) is the minimum image diameter
\( d_g \) is the geometrical image diameter.
\[ d_i = \sqrt{\left(\frac{M d_p}{M} \right)^2 + \left(2.44(M + 1)^2 f^* \lambda\right)^2} \]  
(6.4)

where the optical configuration (shown in Figure 6.15) consists of an imaging lens with f-number \( f^* \), which is the ratio of the focal length \( f \) to aperture diameter \( D_o \). The tracer particles with a known diameter \( d_p \) in the object plane are illuminated with a light source of a known wavelength \( \lambda \), and the image magnification is denoted by \( M \), which is defined as

\[ M = \frac{z_0}{Z_0} \]  
(6.5)

where \( z_0 \) is the distance between the image plane and the lens

\( Z_0 \) is the distance between the lens and the object plane.

Figure 6.15  Optical configuration of PIV set-up (Anandarajah 2005)

The variables that affect the particle image diameter are shown to be more dependent on the imaging of the lens than the physical particle diameter. In the following graphs
Figures 6.16 and 6.17) the influence of the particle physical diameter compared to the magnification is shown.

Figure 6.16 Variation of particle image diameter, $d_i$, with particle diameter, $d_p$, with changes in object format, $M_0$ (Anandarajah 2005)

Figure 6.17 Variation of image pixel size with particle diameter $d_p$ (Anandarajah 2005)
Grant (1997) described that $d_d$ would only be obtained when imaging small particles in the order of a few $\mu$m and/or at small magnification $M$. Conversely for larger $d_p$ and/or larger $M$, the influence of $d_g$ is more dominant. This effect is best illustrated using the simulated 2D and 3D expanded Gaussian intensity profile as shown in Figure 6.18.

Figure 6.18 Simulated 2D and 3D expanded intensity profile images showing that increased particle diameter ($d_p$) has minimal affect on image diameter ($d_i$) (Anandarajah 2005)
It is evident from these simulated images that the imaged size does not increase linearly with the particle diameter. The difference in the image size between the 3 μm and 10 μm particle diameter show little significant change in image size. This is due to the minimum image diameter, \((d_d)\) being dominant when \(d_p\) is small. In the instance where \(d_p\) is increased from 20 μm to 30 μm the influencing factor is \(d_p\) in increasing the image size \(d_i\). This confirms the previous figures (Figure 6.16 and 6.17) showing the affect of particle size on image size. The diffraction limited image size and the seeding used will be discussed later in this chapter.

6.6 DPIV measurement accuracy

The overall accuracy of the PIV system is a challenge to absolutely quantify due to the way the measurements are taken. In this current research there are two main sources of error in the measured PIV results, these are:

i. Inferred motion of particles to actual motion of air charge

ii. PIV error due to stretch/strain across interrogation regions as a result of velocity gradients.

There are further relatively minor (in this instance) sources of error due to optical deformations, background noise of the image, agglomeration of seed particles and centre fitting of the particle image. The central fitting peak of the TSI Insight software states that it was approximately 0.1 pixel accurate for particle image diameters in the region of 3 pixels. The two main sources of error stated above are now discussed and quantified. A current and complete analysis of PIV error can be found in Anandarajah (2005).

6.6.1 Inferred motion of particles

Chapter 3 presented how the motion of a particle in a fluctuating flow would respond to a change in the magnitude of the flow. Whilst this formula is of importance when
considering the turbulence in the flow, it is important to remember the nature of the flow being studied. Maurice (1992) gave a further example of how seed motion would rotate and deviate from the air flow in an aircraft wing eddy, showing how increasing particle size tracked the tangential flow with increasing error. In this work, we are considering the flow as a swirling motion in the bowl of the piston. Since the particle density (0.9 μm Zirconium Oxide, density 5760 kg.m⁻³) is much greater than the charge air density (air density at TDC is approximately 19.74 kg.m⁻³), the centrifugal force acting on the particle will be greater than the air. Therefore, it is necessary to quantify this difference in particle motion in the radial direction compared to the air charge motion.

Durst et al. (1981) described a method for analysing the behaviour of seed particles in a vortex flow. By considering the forces acting upon the flow, the equation of drift motion of the particle can be reduced to a simple drift motion. This is as follows

\[
\frac{V_R}{V_\theta} = \frac{m_p V_\theta}{6 \pi \alpha \mu \mu_{air} r}
\]

where

- \( V_R \) is the radial velocity of the particle (m.s⁻¹)
- \( V_\theta \) is the tangential velocity of the particle (m.s⁻¹)
- \( m_p \) is the mass of the particle (kg)
- \( \alpha \) is the particle radius (m)
- \( \mu_{air} \) is the dynamic viscosity of the air at TDC (N.s.m⁻²)
- \( r \) is the radial spatial co-ordinate (m).

The tangential velocity can either be derived from the approximated angular momentum spin-up, as explained in Chapter 5, or by simply using the measured data taken from the engine. For this purpose, the highest swirl rate is used to consider the radial motion of the flow, presented in the deep bowl piston, helical inlet at 1600 rpm condition. The result of this drift velocity can be seen in Figure 6.19.
Figure 6.19  Radial drift velocities compared to tangential velocities

Figure 6.19 shows the radial component due to the calculated drift velocity. The values of the drift are very small and correspond to approximately 0.5-0.7% of the tangential velocity. This is sufficiently small to consider the particles to faithfully follow the air charge motion.

6.6.2 Velocity gradient across interrogation regions

The most recent and thorough work into error analysis was carried out by Anandarajah (2005) who investigated the error of FFT cross-correlated PIV vector maps. Prior to this work, one of the main methods of quantifying error was the Lawson equation (Haste 2000; Jarvis 2003) which considered the strain across interrogation regions leading to error. Anandarajah used computer generated images in 32 x 32 pixel regions to determine the absolute error in the DPIV measurement technique. The author went onto suggest a normalised method of correlating data using a Normalisation by Signal Strength (NSS) correlation technique, which normalised the signal strength of the particle image prior to the Gaussian fitting of the peak for the interrogation region. A DPIV model was created which generated sequences of PIV image pairs with a one-dimensional displacement gradient across the 32 x 32 pixel interrogation region. Electronic noise was accounted for in this
model to ensure accuracy to measured data. The generated images plus noise were modelled in terms of particle image diameter in pixels for 1 pixel and 4 pixel mean displacement. With increased mean pixel displacement, the accuracy increases due to the greater displacement. The results of this model are given in Figure 6.20.

![RMS error comparisons](image.png)

**Figure 6.20** RMS error comparisons between theoretical images plus noise and experimental images (32 x 32 pixels) (Anandarajah 2005)

To use this measure of RMS error to the data acquired, it was necessary to consider the flow recorded in a one-dimensional plane and hence the flow was considered in terms of \( \frac{dU}{dy} \) and \( \frac{dV}{dx} \) which was achieved using the TecPlot analysis software. This revealed the area of greatest strain across interrogation regions. Once the area of maximum strain across interrogation regions was found, the information at that region was probed by revealing the velocities in the \( U \) or \( V \) direction. Since the interrogation regions overlapped by 50%, the velocity magnitudes of the interrogation regions around the central maximum strain were considered. These velocities were then
interpreted in terms of particle image diameter in pixels to determine the displacement gradient on the x-axis in Figure 6.20. The mean displacement was then calculated in terms of pixels. These results are then compared directly with Figure 6.20 for a direct measurement of error.

After applying this method of error measurement to the results obtained, it was found that in the areas close to the cylinder head window edge and areas of brightspots in the data (showing artificially low data points), large errors had occurred. Gradient displacements in the region 0.5 particle image diameters with a mean of 1 pixel displacement, resulted in RMS errors of approximately 10-11%. For the majority of the vector maps, however, it was found that the highest strain resulted in gradient of particle image diameter < 0.1 particle image diameter, and a mean displacement in the region of 4 – 5 pixels. This resulted in RMS errors in the region of less than 2% for the majority of the vector map and only deviating in areas where there were brightspots or those close to an image edge.

6.7 Preliminary Testing

Initial testing was carried out on the engine with the DPIV equipment to identify difficulties involved in obtaining good PIV data. A static test was set-up and a flow of air was blown across the bowl of the piston. This was achieved by removing the valve gear from the cylinder head to allow free flow of air and seeding into the engine. Seeding was delivered using a TSI jet atomizer that seeded the incoming air with olive oil droplets of approximately 1 μm diameter. A slight positive pressure was applied to the inlet side and a negative pressure on the exhaust manifold by an extraction fan to ensure a flow. The TSI DPIV system was set-up using the necessary optics to create a laser sheet which was introduced into the engine through the cylinder glass and the piston cassette set-up, whilst the piston was held stationary at TDC. The cylinder head was bolted in place and the camera mounted directly above the cylinder head focussing through the cylinder head window. Good PIV results of this slow flow across the cylinder were obtained for this stationary configuration.
This data was used for understanding and exploring the set-up of the PIV system in a motored CI engine and what could be achieved with the equipment available.

Routine measurements of flows in the engine, however, took considerable fine tuning of the PIV equipment and the seeding system. One of the main challenges was to ensure that the laser sheet was formed correctly (flat and at correct height) in the required image plane when the piston was at TDC. Further problems encountered in testing of the engine were glassware breakage under test conditions (cylinder head window and the cylinder glass), correct sealing of the cylinder head to the cylinder glass and seeding density. Once the full measured dataset were obtained, the experimental investigation began to look at various parameters of the in-cylinder flow and how they could be interpreted qualitatively. The main areas were:

i. Computation of the normalised dynamic swirl ratio, which is defined as the swirling motion normalised by engine speed. This was of particular interest when considering the different inlet geometries, piston design and how the swirl spin-up is affected during the compression stroke.

ii. Ensemble averaged flow data within the bowl based on datasets comprising in excess of 1000 vector maps at each engine test point.

iii. The velocity profile across the piston bowl for the different speeds, inlet configurations and piston bowls.

iv. Comparison of measured data with the modelled data proposed in Chapter 5.

6.8 Experimental Procedure

6.8.1 Introduction

The general set-up of the engine and the arrangement of the optical access were shown previously. The details of the exact experimental set-up will now be given. The PIV and engine system required the following areas to be addressed:

i. Choice of seeding particles

ii. Introduction of seeding to the inlet charge
iii. Diffraction limited imaging spot size calculation
iv. Laser sheet alignment (relative to two laser set-up)
v. Alignment of the laser sheet to image plane in engine
vi. Setting up PIV camera
vii. Focussing and alignment of the camera to correct image plane
viii. Collation of raw images
ix. Cross-correlation of raw images to create vector maps
x. Validation of velocity vector maps

The PIV experimental method is reliant on optimal settings to ensure good datasets. How these optimal settings were achieved and made repeatable will be now discussed.

6.8.2 Choice of seeding particles

The most common type of solid seeding employed by previous researchers using PIV was Titanium Oxide (TiO₂). Other common types of seeding used in PIV measurements include Silicone Oxide (SiO₂) and Zirconium Oxide (ZrO₂), both of which display good light scatter properties and are produced sufficiently small enough to follow the flow faithfully.

Simple tests were carried out on the different possible seeding materials. Seed material was introduced using a Malvern seeder, directing the seed into the engine, the flow was imaged using high speed flow visualisation (copper vapour laser and Kodak high speed camera operating at 9 kHz). Various sizes of Titanium Oxide and Zirconium Oxide were imaged and found to scatter light sufficiently well for 2-D PIV.

From Equation (3.10), the response fidelity criterion of the particle is that the amplitude of variation of \( v_p \) is equal to that of \( u \) to within 1% of the maximum fluctuation. This is derived from
\[ f_s < 0.1 \frac{\mu}{a^2 \rho_p} \]  

(3.10)

where \( \mu \) is the dynamic viscosity of the fluid (N.s.m\(^{-2} \))

\( a \) is the particle radius (m)

\( \rho_p \) is the density of the particle (kg.m\(^{-3} \)).

The time response for a particle can then be derived from Equation (3.5) as

\[ \tau_p = \frac{m_p}{6 \pi \mu a} = \frac{2 \rho_p a^2}{9 \mu} \]  

(3.5)

which yields the time constant \( \tau_p \) for the particle to approach fluid flow velocity \( u \) after a velocity change.

These fundamental equations were applied to different types of seeding that were readily available for this project and can be seen in Table 6.2. The dynamic viscosity for the air was determined in Chapter 5 by applying the Sutherland equation and was calculated to be \( 40 \times 10^6 \text{ N.s.m}^{-2} \) at TDC (910 K, 51 bar).

<table>
<thead>
<tr>
<th>Seeding Material</th>
<th>Particle diameter ( \phi ) (( \mu )m)</th>
<th>Particle radius ( a ) (( \mu )m)</th>
<th>Particle density ( \rho_p ) (kg.m(^{-3} ))</th>
<th>Frequency response ( f_s ) (Hz)</th>
<th>Time constant ( \tau_p ) (( \mu )s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicon Oil</td>
<td>1.4</td>
<td>0.7</td>
<td>965</td>
<td>8882</td>
<td>2.5</td>
</tr>
<tr>
<td>Expance 551DE20d60</td>
<td>20</td>
<td>10</td>
<td>60</td>
<td>700</td>
<td>31.7</td>
</tr>
<tr>
<td>Expance 551DE40d42</td>
<td>40</td>
<td>20</td>
<td>42</td>
<td>250</td>
<td>88.9</td>
</tr>
<tr>
<td>Zirconium Oxide ( \text{ZrO}_2 )</td>
<td>0.9</td>
<td>0.45</td>
<td>5760</td>
<td>3600</td>
<td>6.2</td>
</tr>
<tr>
<td>Zirconium Oxide ( \text{ZrO}_2 )</td>
<td>2</td>
<td>1</td>
<td>5760</td>
<td>729</td>
<td>30.5</td>
</tr>
<tr>
<td>Zirconium Oxide ( \text{ZrO}_2 )</td>
<td>5</td>
<td>2.5</td>
<td>5750</td>
<td>117</td>
<td>190.1</td>
</tr>
<tr>
<td>Titanium Dioxide ( \text{TiO}_2 )</td>
<td>1</td>
<td>0.5</td>
<td>4200</td>
<td>4000</td>
<td>5.6</td>
</tr>
<tr>
<td>Titanium Dioxide ( \text{TiO}_2 )</td>
<td>5</td>
<td>2.5</td>
<td>4200</td>
<td>160</td>
<td>138.9</td>
</tr>
<tr>
<td>Silicon Carbide</td>
<td>1.5</td>
<td>0.75</td>
<td>3200</td>
<td>2333</td>
<td>9.5</td>
</tr>
<tr>
<td>Nylon</td>
<td>4</td>
<td>2</td>
<td>1140</td>
<td>921</td>
<td>24.1</td>
</tr>
<tr>
<td>Polystyrene Latex</td>
<td>0.54</td>
<td>0.27</td>
<td>1050</td>
<td>54870</td>
<td>0.4</td>
</tr>
<tr>
<td>Metallic Coated ( \text{TiO}_2 )</td>
<td>9</td>
<td>4.5</td>
<td>2600</td>
<td>80</td>
<td>278.6</td>
</tr>
<tr>
<td>Hollow Glass Spheres (HGS)</td>
<td>10</td>
<td>5</td>
<td>1100</td>
<td>153</td>
<td>145.5</td>
</tr>
<tr>
<td>Metallic Coated HGS</td>
<td>14</td>
<td>7</td>
<td>1650</td>
<td>52</td>
<td>427.8</td>
</tr>
<tr>
<td>Water Droplets</td>
<td>1</td>
<td>0.5</td>
<td>1000</td>
<td>16800</td>
<td>1.3</td>
</tr>
</tbody>
</table>

Table 6.2  Flow following behaviour of seed particles in compressed air
The $f_u$ predicts the flow fluctuation below which the particles will have less than 1% error of the fluctuation amplitude. It can be seen that the main controlling factor in determining $f_u$ is the particle diameter. The requirement of the seed, with respect to a CI engine, needs to survive the compression phase where temperature and pressure is approximately 912 K and 51 bar respectively based on adiabatic compression. This reduced the possible seed particles to a few possibilities as water evaporates and oil combusts. The Expancel, Polystyrene Latex and Nylon were all polymer based materials which degraded and compressed under this pressure and temperature. The glass micro spheres were found to be damaging to the engine and too low a flow following capability when compared with the alternatives. This resulted in the choice of seeding between Zirconium Oxide, Titanium Dioxide and Silicone Carbide. Titanium Dioxide is the most commonly used of the solid seed materials, however, on testing with the Malvern seeder available in this study, it was found that it easily agglomerated when exposed to air, leading to blockage and poor quality of seeding. Silicone Carbide was not readily available and there were concerns over the compatibility of using this material inside an engine with regards to wear. Zirconium Oxide was found to readily be broken down by the seeder unit and provided a constant flow of seeded air to the engine inlet plenum chamber. The 0.9 µm Zirconium Oxide particles also displayed good flow following behaviour in the above table, with particle following up to 3.6 kHz and a time constant of 6.2 µs. The 2 µm Zirconium Oxide was found to display substantially less flow following behaviour, but with reference to Tables 6.3 and 6.4, without showing a significant increase in diffraction limited image size.

From these investigations the 0.9 µm Zirconium Oxide was the most suitable choice for seeding for the 2-D PIV experiments in this project.

6.8.3 Introduction of seed to inlet charge

The seed was introduced into the inlet plenum chamber of the engine via the use of a Malvern 6306 Particle Seeder. This equipment is normally used in conjunction with a Malvern Particle Sizer, however, it was successfully used in this project to introduce
solid seeding into the air flow of the inlet plenum. The basic principle of this seeder used a vibrating table to control the flow of the seed material, which flowed over ball bearings aiding in the breaking up of agglomerated particles and introducing the particles into a pressurised airflow by the venturi effect. Figure 6.21 presents a schematic of the system and Figure 6.22 shows a photograph of the actual unit.

![Schematic of Malvern particle seeder](image1)

Figure 6.21  Schematic of Malvern particle seeder

![Photograph of Malvern particle seeder](image2)

Figure 6.22  Photograph of Malvern particle seeder

Whilst this seeder was efficient at breaking the seed down from agglomerated lumps, it was difficult to maintain a constant supply of the seeded flow. A combination of controlling the air pressure and the vibration of the table had to be employed to regulate the flow. The outlet pipe of the seeder was placed into the inlet plenum chamber where it seeded the incoming charge air to the engine.
6.8.4 Alignment of laser sheets

Within the New Wave Laser, two independent flash lamp pumped Nd:YAG lasers run, each of which could be fired independently, thus allowing for a timing difference to sub μs between the laser pulses. Within the laser head, there are optics that altered the beam by the use of a second harmonic generator and dichroic mirrors to ensure a beam of 532 nm wavelength. These optics could become misaligned during transportation of the laser head and hence the laser pulses had to be accurately aligned to ensure that the sheet of light illuminated the same plane on the first and second pulse. The layout of the laser head is shown in Figure 6.23.

![Figure 6.23 Layout of New Wave laser internal mirrors (TSI 2000)](image)

To accurately align the laser sheets, the beams are pulsed at low power without any sheet forming optics over a distance of approximately 10 m and aligned to each other (i.e. ensuring the beam is illuminating the same spot). Once these beams were aligned, the laser was placed into the experimental set-up in order to fine tune the laser sheets. Sheet forming optics was fitted to the laser head and the mirrors put in place. The beam was then approximately aligned to illuminate the correct plane within the engine, which was 1 mm above the piston pip at TDC. Mirrors were then
used to beam the laser pulses over a distance of 1m from behind the engine into the piston bowl, as shown in Figure 6.24.

![Figure 6.24 Laser sheet path into piston bowl](image)

Once the laser sheet was approximately aligned in the flow region, a backboard with a stencilled line was used to horizontally align the sheet flat and to the correct height. The piston was set to TDC and a plate shaped to the contour of the piston bowl was placed onto the piston, the flat top of this insert was exactly 1 mm above the piston pip. The beam was then manoeuvred so that it skimmed the surface of this insert. A final stage for the alignment of the laser to the engine was to test the two laser pulses for illumination of the same plane. This was achieved by placing a silicone block with suspended seed particles into the laser sheet area. The lasers were pulsed and the images acquired by the camera were compared. Since the particles were suspended in the silicone, there was no movement of the particles between images. The camera should therefore see identical particle images for each laser pulse. The two images were then cross-correlated to ensure that the camera was imaging the same particles, and hence no displacement vectors, between the laser pulses. An example of a pair of particle images for the silicone block with suspended particles is shown in Figure 6.25.
Figure 6.25 Pair of particle interrogation regions of silicone block with suspended particles (64 x 64 pixels interrogation region size)

This pair of images was then cross-correlated to determine if there was a movement of particles due to the camera viewing different particles between the laser sheets. The result of this cross-correlation can be seen in Figure 6.26. The vector scaling was increased to indicate vectors (scaled up by 50).

Figure 6.26 Cross-correlated vector map of silicone block with suspended particles in TSI Insight 3 software (vector scaling = 500)
The cross-correlation of the image pairs does not allow for a zero value, hence a value of velocity vector will be calculated, which in this case are very small vectors. These vectors are sufficiently small to be of no consequence to the velocity vectors that were measured. To quantify this velocity map, it was interpreted as a velocity vector map, with a scale in m.s\(^{-1}\), shown in Figure 6.27.

![Velocity vector map of silicone block with suspended particles](image)

Figure 6.27  Velocity vector map of silicone block with suspended particles

From the velocity vector map for the static silicone block, the cross-correlation performed on the two image files shows the magnitude of these velocities. The vectors are the result of correlating the background noise and minor errors in the alignment of the laser sheets, which allowed for illumination intensities of different particles between the sheets. This error is very small with the mean U velocity component being 0.0095 m.s\(^{-1}\) with a standard deviation of 0.035 and the mean V velocity component being 0.005 m.s\(^{-1}\) with a standard deviation of 0.034 (calculated from Insight software).
6.8.5 Focussing of the camera

The focussing of the camera to the correct image plane was achieved by the use of the silicone block used in the alignment of the laser sheets. When the two laser sheets were aligned to the correct image plane, the camera was focussed to the plane of particles that were illuminated within the silicone block. Previously in Section 6.5.1, the case of diffraction limited imaging of small particles was discussed. In order to achieve accurate results, each particle should cover in the region of 2–3 pixels (TSI 2000). This allowed a Gaussian fit to find the geometric centre of the particle with an error of less than 0.1 pixels (Raffel et al. 1998). The size of the particle images onto the CCD chip was controlled by the f-number (f#), which is defined as the ratio between focal length and the aperture diameter of the lens. In practical terms the correct f# for the lens can be determined from the following expression

\[ d_{\text{diff}} = 2.44 f\#(M + 1)\lambda \]  \hspace{1cm} (6.7)

where \( d_{\text{diff}} \) is the diffraction limited image size

\( f\# \) is the ratio between focal length and aperture diameter

\( M \) is the magnification factor from Equation (5.5) where \( M = \frac{z_0}{Z_0} \)

\( z_0 \) is the distance between the image plane and the lens

\( Z_0 \) is the distance between the lens and the object plane

\( \lambda \) is the wavelength of the light being used to illuminate the particles.

From this, the minimum diffraction limited image can be calculated and hence the actual image. Neglecting lens aberrations, the total image size can be calculated for a finite size particle from

\[ d_r = \sqrt{(M d_p)^2 + d_{\text{diff}}^2} \] \hspace{1cm} (6.8)

In the case of DPIV, this image size was then converted to pixel image size, so it gave a description of the image size in terms of pixels. The image size in pixels is calculated as follows
\[
d_{\text{pixel}} = \frac{d_r}{M.R_{\text{pixel}}} \tag{6.9}
\]

where \( R_{\text{pixel}} \) is the ratio of image size to pixel size.

In the instance of the DPIV camera used, the number of pixels in each direction is 1000 (hence a 1 megapixel camera) and the CCD chip size is 9 mm. The total image size varies according to whether the experiment used the shallow or deep bowl piston, and was measured using a reference grid and reference image, an example of this is shown in Figure 6.28.

![Reference image and reference grid for PIV calibration](image)

Figure 6.28 Reference image and reference grid for PIV calibration

The reference image was inspected using a suitable drawing package (for example Paint Shop Pro 8) and the grid measured in terms of pixels. This measurement gave the user exact knowledge of the image size and allowed for calibration of the PIV images to measure the velocity in absolute distance rather than pixels.

Since the image size was fixed by the radius of the piston bowl, the CCD chip was a fixed size in the camera, the particles were limited to 1 - 5 \( \mu \text{m} \) for sufficient flow following capability and the laser sheet wavelength was also fixed. The only factor in changing the particle image size on the chip was the f\# of the camera. Reducing the
aperture size created a larger image of the particles onto the CCD chip. Tables 6.3 and 6.4 show the results of the calculation in Equations (6.7) – (6.9) for different size particles and f#’s, for deep and shallow piston bowl respectively.

<table>
<thead>
<tr>
<th>Particle size (μm)</th>
<th>f# 4</th>
<th>f# 8</th>
<th>f# 11</th>
<th>f# 16</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.76</td>
<td>1.53</td>
<td>2.1</td>
<td>3.06</td>
</tr>
<tr>
<td>2</td>
<td>0.77</td>
<td>1.53</td>
<td>2.1</td>
<td>3.06</td>
</tr>
<tr>
<td>5</td>
<td>0.78</td>
<td>1.54</td>
<td>2.11</td>
<td>3.06</td>
</tr>
<tr>
<td>10</td>
<td>0.84</td>
<td>1.57</td>
<td>2.13</td>
<td>3.08</td>
</tr>
</tbody>
</table>

Table 6.3  Deep bowl diffraction limited image results (pixel size)

<table>
<thead>
<tr>
<th>Particle size (μm)</th>
<th>f# 4</th>
<th>f# 8</th>
<th>f# 11</th>
<th>f# 16</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.73</td>
<td>1.47</td>
<td>2.02</td>
<td>2.94</td>
</tr>
<tr>
<td>2</td>
<td>0.74</td>
<td>1.47</td>
<td>2.02</td>
<td>2.94</td>
</tr>
<tr>
<td>5</td>
<td>0.75</td>
<td>1.48</td>
<td>2.02</td>
<td>2.94</td>
</tr>
<tr>
<td>10</td>
<td>0.79</td>
<td>1.5</td>
<td>2.04</td>
<td>2.96</td>
</tr>
</tbody>
</table>

Table 6.4  Shallow bowl diffraction limited image results (pixel size)

From Tables 6.3 and 6.4, the camera was set to f# 16 to create a large enough image size on the CCD to determine the geometric centre of the particles.

Once the camera lens was focussed to the image plane illuminated by the lasers and the f# was correctly set, the lens was fixed into place to prevent the camera from vibrating out of focus during running. The camera was mounted on a slide table attached horizontally to the roof cage of the engine housing. This allowed the camera to be moved away when stripping, cleaning and rebuilding the engine without having to refocus the lens to the correct image plane.
6.9 Data Acquisition

In the previous section, the set-up of the DPIV equipment was described. The following section explains how all the sub-processes were brought together to record PIV data from the motored engine.

6.9.1 Motor controller

The engine was motored using a 3-phase 7 kW motor coupled to the engine by a belt drive on a 1:1 ratio. A Control Techniques speed controller was used for varying motor speed. The motor speed could be controlled by either an analogue speed dial, or by inputting into the digital control panel the desired engine speed in rpm. By use of the digital control panel, the speed could be fixed and maintained without further input from the user.

6.9.2 Oil pump

The Perkins Trailblazer engine was a dry sump type engine, which meant that the oil held for lubricating the engine was in a tank separate to the engine. Since the cam carrier and cylinder head were redesigned to run on grease, the oil requirement was only to feed the main engine block. The oil pump was a 3-phase motored rotary type pump which could deliver oil pressure and flow rate beyond the requirement of the engine. After the pump, an oil filter was mounted which removed any foreign bodies from the oil likely to have been introduced from the seeding of the engine, as well as wear of metallic parts in the engine. A relief valve was fitted after the filter to control the supply pressure to the engine, the gauge for the pressure being mounted on the side of the engine. Oil pressure was set to 10 bar to ensure full lubrication of the bottom end of the engine.
6.9.3 Computer and TSI Insight 3 software

Software used for the capturing and processing of data was Insight 3, produced by TSI Inc. The PC was a dual processor Pentium 3 fitted with a frame grabbing card to capture the PIV images and to store them to memory or HDD. Timing between the frames was controlled by the Insight software so that PIV could be captured with a suitably small time frame. Laser power was controlled by altering the Q-switch delay of the lasers. Sensitivity of the CCD chip in the camera was variable and was increased to the maximum sensitivity to enable the lowest power laser beam to illuminate the flow. A number of running tests for each data point were carried out to ensure that the correct timing, illumination strength and correct seeding density were achieved.

6.9.4 Data recording

When recording data, a set of 400 pairs of images were taken, each at TDC, and written directly to the HDD of the PC. During the image recording mode, the data was written to the HDD and not analysed in anyway, but was imaged on the screen so that the user could visualise the quality of the data being recorded. For faster operation of image recording, it was possible to record the images to the computers Random Access Memory (RAM) and write to disk after the experiment. The timing diagram of the two-frame cross-correlation was shown in Figure 6.11 providing visualisation of this method.

6.9.5 Post processing recorded data

Once the images were recorded to the HDD, they could be analysed to create the velocity vector files. The images were imported into the Insight 3 software and the parameters for the inspection of the two images set (velocity calibration, interrogation region size, signal to noise ratio, etc.). A polygon mask was created around the image where information was recorded (as the entire image did not contain particle information) to reduce computation of the entire field where no information had been
recorded. During the analysis set-up a calibration value for the velocity was set by inputting the actual image size in mm so that velocity vector data could be calculated in SI units (m.s\(^{-1}\)). Post processing was set to calculate the velocity of each interrogation region. Each interrogation region being 32 x 32 pixels in size with a 50% overlap. The correlation engine employed was a Fast Fourier Transform (FFT) with a Gaussian peak search algorithm. The computer then correlated the images to create a vector map for each pair of recorded PIV images.

Cross-correlation method of processing PIV can regularly produce in excess of 95% correct velocity vectors (TSI 2000), however, spurious results are also created. It was necessary to ensure that each experiment that was processed regularly showed in excess of 90% valid vectors in the correlated region. These spurious vectors were removed after the processing by further validating the vector fields using the procedure now discussed.

6.9.6 Validation of vector maps

A small number of spurious vectors were created in each vector map, typically fewer than 5%. These resulted from a number of factors:

i. High or low seeding density (i.e. number of particles per interrogation region)
ii. High background noise
iii. Laser flare
iv. Out of plane flows
v. Incorrect correlation of peak due to poor signal to noise ratio
vi. Fouling of optical window
vii. Irregular local seed density.

These spurious vectors were removed by a strict validation method that was built into the Insight 3 software. A macro was recorded and the validation method applied to each vector map. The following details the different types of validation that were available for data post processing:
i. **Standard deviation** used the vector field global mean to calculate the standard deviation. The standard deviation could be reduced by the user to limit the values of the vectors.

ii. **Global range** set the maximum and minimum range of the values in the U and V directions. This was a basic filtering method for removing large spurious vectors.

iii. **Local median filter** compared each vector to the median value of the neighbouring vectors. The neighbourhood size was input by the user in the range of 3 x 3 to 9 x 9, whereby small neighbourhood sizes were used for large velocity gradients, and large neighbourhood sizes for small velocity gradients or velocity maps with areas of missing data.

iv. **Local mean filter** computed the ensemble average of the U and V components of the neighbouring vectors. As above, the neighbourhood size was set by the user in relation to the velocity maps. A value for the deviation from the mean U and V components was input.

v. **Smooth filter** replaced each velocity vector with a weighted average of the velocity vector and the neighbouring vectors. It also filled the areas of no data or blanked vectors with interpolated values. As above, a neighbourhood size was defined and a Gaussian distribution set the relative weighting of the neighbouring vectors. Much care must be exercised with this filter as it altered the velocity vector map greatly and can significantly change the results. This filter could be used for primary filtering of high frequency velocity variations from vector maps.

Validation methods could significantly alter the data that was recorded and correlated. In this work, two specific filters were applied to the velocity vector maps; a range filter and a median filter. Importantly, the vectors that were removed from the velocity vector maps were not interpolated or filled as a result the raw data was little changed. Table 6.5 details the range and median filters that were applied to velocity vector maps obtained, as follows:
<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>Range Filter</th>
<th>Median Filter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>U Velocity (m.s(^{-1}))</td>
<td>V Velocity (m.s(^{-1}))</td>
</tr>
<tr>
<td>Deep Bowl</td>
<td>Max.</td>
<td>Min.</td>
</tr>
<tr>
<td>800</td>
<td>15</td>
<td>-10</td>
</tr>
<tr>
<td>1200</td>
<td>25</td>
<td>-15</td>
</tr>
<tr>
<td>1600</td>
<td>35</td>
<td>-20</td>
</tr>
<tr>
<td>Shallow Bowl</td>
<td></td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>15</td>
<td>-10</td>
</tr>
<tr>
<td>1200</td>
<td>25</td>
<td>-15</td>
</tr>
<tr>
<td>1600</td>
<td>35</td>
<td>-20</td>
</tr>
</tbody>
</table>

Table 6.5 Validation values applied to correlated results

The values above were calculated from the maximum vectors and removed only spurious vectors. The majority of the vector maps maintained 90-100% of the original vectors from these validations. This resulted in a small number of spurious vectors in the flow that statistically had little impact on the average flow (averaged over 400 vector maps).
6.10  Summary

This chapter has presented the arrangement of the engine, the DPIV system used and the experimental process used to acquire data from the engine. The DPIV equipment that was used has been described in detail and the method of correlating the image pairs to deduce the velocity vector maps, showing the affect of particle size on the recorded image, which is significantly greater than the original geometric size of the particle. Measurement accuracy was introduced and an estimation of the accuracy of the PIV system that was employed given. The chosen seed material was shown and the reasons for this choice of seeding given. The seeding unit was presented, showing how seed was introduced to the engine inlet manifold. Data that was recorded from the engine was then cross-correlated using the TSI Insight 3 software. The method of validating the vector maps with minimal alteration was described and the values for validation given. The following chapter will present the raw data acquired from the engine in terms of an averaged flow map and an instantaneous vector map for each of the 18 test points in this current research. This data is then further analysed using a Matlab program to gain a deeper understanding of the flow regime. The results are compared to modelled data from Chapter 5. The results are then analysed for the affects of changing engine speed, piston bowl and inlet configuration to the velocity profiles in the piston bowl.
Chapter 7  PIV Results

7.1 Introduction

This chapter presents the results of the validated vector maps for the full 18 engine test points. For each test point some 1000 to 1500 different PIV maps were recorded. The data that is presented has been validated according to the criteria in Table 6.5. For the raw data results, the ensemble average of a sample of 400 consecutive vector maps and an example instantaneous vector map are presented in Appendix A5. The velocity vectors in these maps are all plotted in m.s$^{-1}$.

Using the raw data recorded and presented in Appendix A5, a Matlab analysis program was used to further process the data to gain a deeper understanding of the flow structures that are present in the piston bowl. The Matlab program interpreted the results into a graphical form that could be directly compared to the mathematical model that was introduced in Chapter 5. Data was then processed in terms of normalised dynamic swirl ratio at TDC, which allowed for direct comparison of the affects of engine speed and inlet port configurations.

A discussion of the data will follow each section which will build the knowledge of the in-cylinder flows for the 18 engine test points.

A sample of the 3-D Holographic PIV results will also be presented which shows the 3-D fully resolved technique in the piston bowl. This data will be used for illustration purposes.
7.2 Raw data results for 2-D PIV

The 18 engine test points are shown in Table 7.1 which details the test point configuration. Figures for each test point are referred to for the instantaneous and averaged results which are presented in Appendix A5.

<table>
<thead>
<tr>
<th>Engine speed (rpm)</th>
<th>Deep bowl piston</th>
<th>Shallow bowl piston</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Combined inlet</td>
<td>Directed inlet</td>
</tr>
<tr>
<td>800</td>
<td>Figures A5.1 and A5.2</td>
<td>Figures A5.3 and A5.4</td>
</tr>
<tr>
<td>1200</td>
<td>Figures A5.7 and A5.8</td>
<td>Figures A5.9 and A5.10</td>
</tr>
<tr>
<td>1600</td>
<td>Figures A5.13 and A5.14</td>
<td>Figures A5.15 and A5.16</td>
</tr>
</tbody>
</table>

Table 7.1 Full 18 engine test point matrix

The raw data results presented indicate the quality of the PIV and the average velocity vector map for each engine test point. However, these results are difficult to interpret due to the centre not being defined. Figure 7.1 shows an example of an ensemble averaged result from the engine and the approximate centre of the piston bowl for illustrative purposes. Figure 7.2 is an example of an instantaneous PIV result from the same dataset that make up Figure 7.1. In the following section, when the results are further processed, the exact centre of the bowl is clearly shown.

Referring to Appendix A5 (Figures A5.1 to A5.36) a short discussion of the raw data is given. The vectors are presented using colour identification bars (the scale of which changes for each set of engine speeds) which represent the velocity vectors in m.s$^{-1}$. 

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Figure 7.1  Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, combined inlet, 800 rpm crankshaft speed

Figure 7.2  Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, combined inlet, 800 rpm crankshaft speed
7.3 Discussion of raw data results

From these vector maps, it was seen that the directed and helical ports exhibit a higher velocity flowfield magnitude for the same engine speeds, when compared with the combined inlet. The deep bowl results displayed a higher velocity magnitude than the shallow bowl results, as was expected, caused by the spin-up of the air charge, as discussed in Chapter 5.

From observation the three different inlet configurations indicated differences in the centring of the charge air motion. Directed inlet flow was more closely centred to the geometric centre of the piston, whereas the helical inlet showed an offset centre to the geometric piston centre. The combined inlet showed a weak centre in the ensemble average results, but in the instantaneous results the centring of the flow is highly variable and in some vector maps two independent swirl centres could be seen (i.e. Figure 7.2). This variable flow would explain the weak centre of the ensemble average for the combined inlet arrangement.

Helical and directed ports demonstrated an increasing velocity with increasing radius, until it reached the boundary layer of the piston bowl wall. The combined inlet illustrated a non-linear relationship between the velocity and the radius, where the maximum velocity of the flow is further from the wall (Appendix A5, Figures A5.1, A5.7, A5.13, A5.19, A5.25 and A5.31). In Section 7.5, where the results are compared with the modelled results from Chapter 5, the reduction in velocity in this region is associated with an increase in the radial component of motion of the velocity vectors. This is attributed to a toroidal flow pattern in the bowl, which in this current work cannot be measured, but there is scope for further investigation by using the Holographic PIV technique which would reveal the z-component of motion.

The following section explains how the vector maps were used to determine the dynamic swirl ratio of the engine and how these results are used to directly compare with the modelled data that was presented in Chapter 5. Velocity profiles will be investigated and a discussion of the results will follow.
7.4 Interpreted results

7.4.1 Introduction

The raw data results presented in Appendix A5 were further processed to gain a deeper understanding of the flow structures, to do this a Matlab program was written which used the original vector maps to determine the following parameters:

i. Ensemble average of the vector maps for each test point (m.s\(^{-1}\))

ii. Tangential velocities about the geometric centre of the piston (m.s\(^{-1}\))

iii. Mean of tangential velocities about the geometric centre of the piston (m.s\(^{-1}\))

iv. Mean radial velocities about the geometric centre of the piston (m.s\(^{-1}\))

v. Dynamic swirl of the flow (rpm)

vi. Mean of the dynamic swirl (rpm)

vii. Normalised dynamic swirl ratio (dimensionless)

viii. Mean normalised dynamic swirl ratio (dimensionless)

ix. Turbulence intensity (m.s\(^{-1}\)).

The Matlab program is shown in Appendix A6 for reference purposes. In the following section the above parameters will be explained further, followed by presentation of these results. The results of the mean tangential velocities are then compared to modelled data, which was presented in Chapter 5, followed by a discussion of the measured data compared to the modelled data. Mean radial velocities are also plotted on these graphs to indicate the level of radial motion. Data is then given which displayed the affect of changing engine speed and different inlet configurations to the velocity profile across the bowl at TDC, followed by a discussion of these results.
7.4.2 Matlab calculations

i. Ensemble average maps that were shown in Section 7.2 were calculated by the TSI Insight software, it was necessary to recompute this average in Matlab for determining other parameters of the flow. The equation for ensemble average was defined in Equation (2.19) as

\[
\bar{U}_{\theta}(\theta) = \frac{1}{N_C} \sum_{j=1}^{N_C} U(\theta, j) \quad \text{for polar co-ordinates}
\]

\[
\bar{U}_{\theta}(x, y) = \frac{1}{N_C} \sum_{j=1}^{N_C} U(x, y, j) \quad \text{for Cartesian co-ordinates}
\]

where \(N_C\) is the number of cycles for which data was taken and \(j\) is the cycle.

ii. Tangential velocities were shown in Chapter 5 in the modelling of the swirl, where the model that was employed in this work only considered the tangential velocities. Hence for accurate comparisons, the raw data needed to be converted to polar co-ordinates. The geometric centre of the piston was determined by inputting a user defined centre in the vector map matrix. As previously discussed in Chapter 6, the reference image of the piston was overlaid onto the recorded data, from this the geometric centre was determined in pixel location. As the vector maps are recorded on a matrix basis, with each velocity vector placed in the centre of each 32 x 32 pixel region, there is a potential maximum error of centre of 0.22 mm for deep bowl, and 0.26 mm for shallow bowl. Overall, this value had a negligible impact on the results. The tangential velocities were determined by computing the position of each vector relative to the user defined centre. Then by employing trigonometry, each tangential velocity vector was calculated. A map of each of these tangential velocities was recorded as a vector map. Figure 7.3 outlines the method for calculating the tangential and radial velocities at each interrogation region in the vector map.
From Figure 7.3, the lengths of $x$ and $y$ are determined from the position of the user defined interrogation region relative to the interrogation region being inspected. The radius from the defined centre can be calculated using trigonometry

$$r = \sqrt{x^2 + y^2} \quad (7.1)$$

and angle $\Psi$ is calculated using trigonometry by the following relationship

$$\Psi = \tan^{-1} \frac{y}{x} \quad (7.2)$$

The tangential velocity acts at $90^\circ$ to the radius and is plotted as $V_\theta$, using similar triangles the angle between $V_\theta$ and the horizontal is equal to $\Psi$. The measured vector being translated is made up of two components of velocity in the $x$ and $y$ direction, from this the angle $\psi$ can be calculated as

$$\psi = \tan^{-1} \frac{U_y}{U_x} \quad (7.3)$$

the actual vector is calculated by using trigonometry as

$$U = \sqrt{U_x^2 + U_y^2}. \quad (7.4)$$

The angle between the tangential velocity ($V_\theta$) and the measured vector can then be calculated as

$$\kappa = \Psi - \psi. \quad (7.5)$$

Hence once the angle $\kappa$ is calculated, the tangential velocity was calculated from
\[
V_\theta = (\cos \kappa) U
\]  
(7.6)

iii. Mean tangential velocities were calculated for each experiment from the sample of tangential velocities that was calculated above and ensemble averaged, as was defined by Equation 2.19.

iv. Mean radial velocities were also calculated for each test point to show the radial component of the flow, these were calculated from trigonometry as discussed for tangential velocities. The radial component of velocity was determined by
\[
V_r = (\sin \kappa) U
\]  
(7.7)
and the ensemble average calculated as was defined in Equation 2.19.

v. Dynamic swirl is an interpretation of the calculated tangential velocities, where the tangential vector is converted from m.s\(^{-1}\) to rpm. This is of particular interest when observing the flow across the piston-bowl to understand the velocity profile.

vi. Mean of the dynamic swirl were calculated for each experiment from the sample of dynamic swirl that were calculated above.

vii. Normalised dynamic swirl is taking the dynamic swirl calculated above and dividing it by the engine speed, as was described in Chapter 5 for defining the swirl ratio of an engine.

viii. Mean normalised dynamic swirl was calculated for each vector from the sample of dynamic swirl that were calculated above.

ix. Turbulence intensity is the root mean square (RMS) of the fluctuating component about the ensemble averaged value and is calculated as Cartesian or polar co-ordinates as follows
\[
u'(x,y) = \sqrt{\frac{1}{N_C} \sum_{j=1}^{N_C} [u(x,y)]^2} = \sqrt{\frac{1}{N_C} \sum_{j=1}^{N_C} [U(x,y)^2 - \bar{U}(x,y)^2]} 
\]  
(7.8)
\[ u'(\theta) = \frac{1}{N_e} \sqrt{ \frac{1}{N_e} \sum_{j=1}^{N_e} [u(\theta, j)]^2 } = \sqrt{ \frac{1}{N_e} \sum_{j=1}^{N_e} [U(\theta, j)^2 - \bar{U}_{\text{ed}}(\theta)^2] } \]  

(7.9)

This RMS value gives an indication of the total fluctuating components in each set of data. This fluctuating component encompasses all the turbulence and the cyclic variability.

As discussed in Chapter 5, the basic swirl modelling suggested a swirl spin-up for a simple flat bottomed piston of 1.6 for the shallow bowl, and 2.5 for the deep bowl. The swirl value for the three different types of inlet was given in Table 5.1. Multiplying the steady-flow swirl number and the basic spin-up will give an estimation of the normalised dynamic swirl at TDC. Data will now be shown for the different test points for the mean normalised dynamic swirl ratio and the turbulence intensity (m.s\(^{-1}\)) for each of the 18 test points.

### 7.4.3 Mean normalised dynamic swirl ratio and turbulence intensity results

The mean normalised dynamic swirl ratios are shown as vector maps, followed by their respective turbulence intensity map. These maps were chosen due to the simplicity of comparing the steady-flow swirl rig results and basic spin-up theory. Since these results have been normalised by the engine speed, the values for the normalised dynamic swirl ratio can be directly compared for different engine speeds. Table 7.2 shows the predicted normalised spun-up values.

<table>
<thead>
<tr>
<th>Normalised spun-up swirl number</th>
<th>Deep bowl piston</th>
<th>Shallow bowl piston</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Combined inlet</td>
<td>Directed inlet</td>
</tr>
<tr>
<td></td>
<td>3.8</td>
<td>7.7</td>
</tr>
<tr>
<td></td>
<td>2.4</td>
<td>4.9</td>
</tr>
</tbody>
</table>

Table 7.2 Predicted normalised spun-up swirl numbers from basic modelling and steady-flow swirl results
7.4.3.1 Combined inlet

Figure 7.4  Comparison of mean normalised dynamic swirl ratio for combined inlet, deep bowl – basic spin-up prediction of 3.8
Combined inlet, deep bowl piston

Figure 7.5  Comparison of turbulence intensities for combined inlet, deep bowl (m.s\(^{-1}\))
Figure 7.6  Comparison of mean normalised dynamic swirl ratio for combined inlet, shallow bowl – basic spin-up prediction of 2.4
Combined inlet, shallow bowl piston

Figure 7.7  Comparison of turbulence intensities for combined inlet, shallow bowl (m.s$^{-1}$)
From the results presented in this section some important findings could be drawn. All of the combined inlet maps exhibited large errors in swirl ratio at the centre of the bowl, however, this is purely due to dividing the tangential velocity by small radii, leading to large errors.

Figures 7.4 and 7.6 showed the normalised dynamic swirl ratio for the deep and shallow bowl piston at TDC. The area of maximum flow occurred away from the piston bowl wall boundary layer. Normalised dynamic swirl ratio was seen to increase with increasing engine speed for both deep and shallow bowl cases. A particular high band of flow was shown that began approximately 5 mm from the piston bowl wall for the deep bowl and approximately 10 mm for the shallow bowl. These velocity profiles are shown in more detail later in this chapter. Velocity vectors towards the centre of the piston exhibited a lower dynamic swirl ratio, this was mainly due to the swirl centre changing with each cycle, hence when the results are averaged the small scale detail of the turbulence is lost. It was shown from individual results for the combined inlets that two swirl centres could be seen (as shown in Figure 7.2), or a large weak swirl centre. In Figure 7.4 for the 1600 rpm result, the normalised dynamic swirl was shown to be relatively low for approximately 10 mm radius from the centre of the piston, as the engine speed increased it was shown in instantaneous vector maps (presented in Appendix A5) that the flow in this region exhibited lower magnitude of velocity compared to the increase in the outer radii.

Figures 7.5 and 7.7 presented the turbulence intensity maps for the combined inlet for deep and shallow piston bowl. Turbulence intensity, which is the RMS of the fluctuating components of the velocity, increased with increasing engine speed, as would be expected. Turbulence intensity maps at 1600 rpm showed significantly higher turbulence intensity, which deviated from a linear relationship with engine speed. The turbulence intensity maps have a similar value to that obtained for the directed and helical inlet regime, whilst exhibiting lower values of normalised dynamic swirl ratio. This is of interest since the flows, from steady-flow rig results, would suggest the swirl speed to be approximately half of the directed and helical results. This suggests that the turbulence intensity in combined inlet configurations is significantly higher compared to single port inlet, which from previous results (as discussed in Chapter 2) leads to improved efficiency and combustion characteristics.
7.4.3.2 Directed inlet

Figure 7.8 Comparison of mean normalised dynamic swirl ratio for directed inlet, deep bowl – basic spin-up prediction of 7.7
Directed inlet, deep bowl piston

Figure 7.9  Comparison of turbulence intensities for directed inlet, deep bowl (m.s\(^{-1}\))
Figure 7.10 Comparison of mean normalised dynamic swirl ratio for directed inlet, shallow bowl – basic spin-up prediction of 4.9
Directed inlet, shallow bowl piston

Figure 7.11 Comparison of turbulence intensities for directed inlet, shallow bowl (m.s\(^{-1}\))
The directed inlet exhibited a different flow structures to the combined inlet that was previously presented. Figures 7.8 displayed a thin (< 5 mm) band of high flow that existed close to the piston bowl which decreased in normalised dynamic swirl ratio with increasing engine speed. The affect of this thin high band of flow led to a higher measured swirl on steady-flow swirl rigs. This would be due to the angular momentum at the outer radii having a substantially greater affect on the total angular swirl momentum of the charge air motion. This leads to the over prediction of normalised dynamic swirl ratio, which deviated further with increasing engine speed. The decrement of swirl with increased engine speed can also be attributed to the flow choking at the inlet valve leading to a lower volumetric efficiency (as was described in Chapter 2).

For the shallow bowl case, as presented in Figure 7.10, a better correlation with the predicted normalised dynamic swirl existed for the 800 rpm test point. This result deviated with increasing engine speed where the normalised dynamic swirl is seen to decrease with increased engine speed. As described above, the decrease in normalised dynamic swirl was attributed to the choking affect at the inlet valve for higher engine speeds. The 1600 rpm test point presented a better correlation with assumption of solid body rotation, with the flow centred about the geometric centre of the piston bowl when compared with the 800 and 1200 rpm test points.

Figures 7.9 and 7.11 showed the respective turbulence intensity maps for deep and shallow bowl pistons for the directed inlet. As with the combined inlet, the turbulence intensity increased with engine speed, but a non-linear relationship was seen, where the results for the 1600 rpm increased significantly. In Figure 7.9 for the deep bowl, areas around the geometric centre of the piston bowl showed low turbulence intensity, which related to low ensemble average data in the same regions. The area directly above the piston pip suffered from laser flare which artificially lowered results (as motion of the particles could not be imaged due to image brightness). In Figure 7.11 for the shallow bowl case, areas of low turbulence intensity are shown toward the piston bowl wall for the 800 rpm test point, which suggests that the charge motion was well ordered in this region. This deviated with increasing engine speed, but lower turbulence intensity was still exhibited to the outer edge compared to the mid portion of the piston bowl.
7.4.3.3 Helical inlet

Figure 7.12 Comparison of mean normalised dynamic swirl ratio for helical inlet, deep bowl – basic spin-up prediction of 8.4
Helical inlet, deep bowl piston

Figure 7.13 Comparison of turbulence intensities for helical inlet, deep bowl (m.s$^{-1}$)
Figure 7.14 Comparison of mean normalised dynamic swirl ratio for helical inlet, shallow bowl – basic spin-up prediction of 5.4
Figure 7.15 Comparison of turbulence intensities for helical inlet, shallow bowl (m.s\(^{-1}\))
The helical inlet results in Figures 7.12 and 7.14 present very different effects according to the piston bowl used. The deep bowl results (Figure 7.12) presented a ramped increase of normalised dynamic swirl from the centre of the piston. This was accentuated with higher engine speed, where the band of flow at the piston bowl wall was shown to exhibit a higher dynamic swirl and the region size also increased. The charge air motion close to the geometric centre of the piston presented areas of low velocity. The off-centre swirl that was introduced with the helical port processes about the geometric centre leading to areas of low flow around the piston pip when the results are averaged. The estimated normalised dynamic swirl ratio was shown to over predict the charge air motion substantially. Similarly to the directed inlet, when testing this type of inlet on a steady-flow swirl test rig, the band of faster flow acting towards the outer radii would have a greater impact on the steady-flow swirl rig result.

The shallow bowl result in Figure 7.14 presented a very different outcome to the deep bowl case. For the 800 rpm test point, the normalised dynamic swirl showed a strong agreement with the basic spin up prediction. The normalised dynamic swirl ratio was seen to decrease with increased engine speed. At the 800 rpm test point, the flow was shown to ramp across the bowl with less gradient (compared to deep bowl), but the velocity banding of the flow was much wider. The approximation of solid body rotation was improved with the shallow bowl, but did not accurately define the flow. At the 1600 rpm test point there were areas of blank data, this was caused by bright spots in the image area. The charge air motion was well centred about the geometric centre of the piston pip, which was in contrast to the deep bowl where areas of low flow are shown about the piston pip. The maximum normalised dynamic swirl did not occur at the bowl edge in the shallow bowl as was the case for the deep bowl piston.

Figures 7.13 and 7.15 presented the respective turbulence intensity maps for the helical inlet with deep and shallow bowl pistons. Figure 7.13 for the deep bowl case presents the highest turbulence intensity of the three inlet configurations and this occurs towards the piston bowl wall. Towards the centre of the piston are areas of low fluctuation, referring to Figure 7.12 the same regions of low flow exhibit low normalised dynamic swirl suggesting this area has low velocity across many vector
maps. The turbulence intensity is seen to increase with engine speed. For the shallow bowl case, Figure 7.15, the turbulence intensity was seen to be well distributed across the piston bowl at lower engine speeds. As the engine speed increased, the turbulence intensity was shown to develop in different regions in the bowl, not just towards the outer edge as occurred in the deep bowl case. The helical inlet exhibited the highest levels of turbulence intensity when compared with the other inlet configurations for the respective piston bowls used.

7.5 Comparison of measured results and modelled results

The mathematical model that was presented in Chapter 5 was used in this section to compare the predicted modelled results directly with the measured experimental data. The modelling only accounted for simple flat bottomed piston bowls and not the complex shape of the piston pip employed in testing. In order that the model maintained the correct compression ratio, the bowl depth of the modelled piston bowl was calculated to maintain an identical compression ratio. The model also considered the entire cylinder area and not just the piston bowl, hence, at the piston bowl outer wall the boundary layer was not considered, this was in accordance with the suggested model by Borgnakke (1981), which considered the whole cylinder charge air motion at each crank angle. Three different suggested modelled velocity profiles for each test point are shown, also a basic spin-up is plotted which uses the swirl ratio number from the steady-flow test results and is multiplied by the engine speed and the basic conservation of angular momentum that was suggested by Heywood (1988). The model only considered the tangential velocities of the flow as previously discussed in Chapter 5. The x-axis only considered the flow up to the piston bowl wall, which was 28 mm radius for the deep bowl, and 37.2 mm for the shallow bowl. Average radial component of velocity is also plotted to give an indication of the property of the original measured vector. A discussion of these results will follow each inlet configuration.
7.5.1 Combined inlet

![Graph showing modelled and measured data for deep bowl piston, 800 rpm with combined inlet ports.](image)

**Figure 7.16** Modelled and measured data for deep bowl piston, 800 rpm with combined inlet ports

![Graph showing modelled and measured data for shallow bowl piston, 800 rpm with combined inlet ports.](image)

**Figure 7.17** Modelled and measured data for shallow bowl piston, 800 rpm with combined inlet ports
Figure 7.18  Modelled and measured data for deep bowl piston, 1200 rpm with combined inlet ports

Figure 7.19  Modelled and measured data for shallow bowl piston, 1200 rpm with combined inlet ports
Figure 7.20 Modelled and measured data for deep bowl piston, 1600 rpm with combined inlet ports

Figure 7.21 Modelled and measured data for shallow bowl piston, 1600 rpm with combined inlet ports
Combined inlet, Figures 7.16 – 7.21, showed a correlation with velocity profile 2 of the modelled data up to approximately 80% piston bowl radius, with the measured data lying between profile 2 and the simple swirl theory (based on solid body rotation). This suggested that for the combined inlet the model (for velocity profile 2) under estimated the swirl and the simple swirl theory over estimated it. Alteration to the quadratic formula that describes velocity profile 2 would enable more accurate modelling to suit the measured data of this engine.

The deep bowl piston results (Figures 7.16, 7.18 and 7.20) exhibited a repeatable profile which deviated from the modelled results with increased engine speed. It was also shown for the deep bowl case that the radial velocities increased with increasing piston bowl radius. The level of radial velocity (which the model does not account for) was greater for radius > 20 mm than the directed or helical inlet configurations. This was of particular interest as the tangential velocities for radius > 20mm decreased and were approximately half the value of the helical or directed inlet (as would be expected from the steady-flow swirl rig results). As the model considered the entire cylinder charge (Borgnakke et al. 1981), the boundary condition at the piston bowl wall was not considered. Therefore the model was shown to increase to a maximum velocity at the piston bowl wall, for radii greater than the piston bowl wall, a cubic equation (described in Section 5.6.7) governed the decrease in tangential velocity until the cylinder wall was reached.

Shallow bowl piston results (Figures 7.17, 7.19 and 7.21) exhibited similar results to the deep bowl case. The velocity profile deviated further from the modelled result with increased engine speed. The 1200 rpm result presented greater scattering of the measured result, this was apparent over a number of experiments, showing a complex interaction of the combined inlet at this engine speed. The 800 rpm and 1600 rpm test points presented a more uniform result of the measured data. The radial motion of the measured result was markedly different to the deep bowl case, with the maximum radial component being achieved close to the geometric centre of the piston and remaining relatively constant across the piston bowl. As with the deep bowl, the radial motion was seen to be of similar magnitude to the helical and directed inlets, although the tangential component was approximately half compared to the single port case.
7.5.2 Directed inlet

Figure 7.22 Modelled and measured data for deep bowl piston, 800 rpm with directed inlet port

Figure 7.23 Modelled and measured data for shallow bowl piston, 800 rpm with directed inlet port
Figure 7.24  Modelled and measured data for deep bowl piston, 1200 rpm with directed inlet port

Figure 7.25  Modelled and measured data for shallow bowl piston, 1200 rpm with directed inlet port
Figure 7.26  Modelled and measured data for deep bowl piston, 1600 rpm with directed inlet port

Figure 7.27  Modelled and measured data for shallow bowl piston, 1600 rpm with directed inlet port
Directed inlet, Figures 7.22 – 7.27, showed markedly different results for the deep and shallow bowl pistons. Deep bowl piston results (Figures 7.22, 7.24 and 7.26) presented a similar ramped curve of velocity, which deviated further from the modelled result with increased engine speed. An increased gradient of velocity was observed from radius 20 – 25 mm, where the main part of the charge air motion acted. The model was shown to over predict the tangential velocity for radius > 12.5 mm, however, for radius < 12.5 mm the measured data was shown to be greater than the model result. This dip in the mid-region of results was believed to be due to the interaction of the squish flow. The radial component of motion was seen to remain constant irrespective of piston bowl radius. This result deviated slightly with the 1600 rpm result, where a minor increase of radial velocity occurred in the region of 15 to 25 mm piston bowl radius.

Shallow bowl piston results (Figures 7.23, 7.25 and 7.27) correlated well with the modelled velocity profile 2, showing a slight deviation with increased engine speed, where the modelled result under predicted the measured result with increasing engine speed. At the 1600 rpm test point, the gradient of the profile changed which showed deviation from the measured result compared to the modelled result. The radial component of flow was again shown to remain approximately constant across the piston bowl for all engine speeds (increasing linearly with engine speed). This constant radial motion is of interest when compared to the combined inlet, which showed increased radial motion with increasing radius.

The deviation between the piston bowl results could be due to the interaction of the squish flow, which was greater for the deep bowl piston due to its larger top land area. Since the measurements of the PIV results were taken 1 mm above the pip, the radially acting squish flow would have a greater affect on the mid-flow of the swirl than the outer regions. However, the radial direction of flow was observed to be lower from the measured data when compared with the combined inlet. The directed inlet was expected to present data that would closely approximate solid body rotation, which would be a linear gradient on these graphs. This theory held well for the shallow bowl results, but not for the deep bowl results, suggesting that the difference in the piston bowls (and hence the squish flows) was the altering factor.
7.5.3 Helical inlet

Figure 7.28 Modelled and measured data for deep bowl piston, 800 rpm with helical inlet port

Figure 7.29 Modelled and measured data for shallow bowl piston, 800 rpm with helical inlet port
Figure 7.30  Modelled and measured data for deep bowl piston, 1200 rpm with helical inlet port

Figure 7.31  Modelled and measured data for shallow bowl piston, 1200 rpm with helical inlet port
Figure 7.32  Modelled and measured data for deep bowl piston, 1600 rpm with helical inlet port

Figure 7.33  Modelled and measured data for shallow bowl piston, 1600 rpm with helical inlet port
Helical inlet results in Figures 7.28 – 7.33 presented different velocity profiles according to piston bowl used. The deep bowl piston results (Figures 7.28, 7.30 and 7.32) correlated better with increased engine speed. The 800 rpm result presented a similar ramped curve to the directed inlet measured data (Figure 7.22) for the same engine speed. As the engine speed increased, the velocity profile of the measured data was shown to become a closer approximation to a linear increase and correlated better with the modelled result (velocity profile 2). The peak velocity was shown to occur at approximately 25 mm radius, whereupon the tangential velocity reduced to zero at the piston bowl wall. The radial velocities presented a different profile to the directed inlet, where they were shown to be unsteady for the radius < 7.5 mm, thereafter they increased slightly and remained constant for the rest of the piston bowl. The radial motion was observed to be less than the combined inlet port case. The helical inlet exhibited the highest tangential velocity of all the measured data in the deep bowl case, with maximum measured average velocity of 25 m.s\(^{-1}\).

The shallow bowl piston measured tangential results (Figures 7.29, 7.31 and 7.33) correlated with the modelled velocity profile 2. This correlation is shown to deviate with increased engine speed, where the modelled data over predicted the measured result. Figure 7.33 showed the peak measured tangential result to deviate further from the modelled data at radius > 27.5 mm, whereas the 800 and 1200 rpm results were shown to follow the modelled data until the boundary layer of piston bowl wall. The radial component of velocity was presented, as with the directed inlet, to remain approximately constant for the entire piston bowl radius. Similarly to the directed result, the radial velocity was shown to be of similar magnitude to the combined inlet, whereas the tangential velocity was approximately double the combined inlet.

As previously mentioned, this deviation between the piston bowls would suggest that the interaction of squish with the swirl affected the results for the deep bowl case. The boundary layer towards the piston bowl wall was significantly less than the combined inlet, and slightly less than the directed inlet, where peak velocities occurred in the region of 3 mm from the outer wall for both piston bowl cases.

Since the model was based on a flat bottomed piston and did not account for the piston pip, a further observation is that the shallow bowl piston was more
representative of a simple flat bottomed piston than the deep bowl piston. This
indicates that the model could be further improved by incorporating the piston bowl
design into the model.

7.6 Velocity Profiles

Data that has been presented is now interpreted for comparing directly the affect of
engine speed and inlet configurations for the mean results. Normalised dynamic swirl
ratio \( \left( \frac{\text{Measured Swirl Speed (rpm)}}{\text{Engine Speed (rpm)}} \right) \) shall be used as the main unit since this allows
comparisons to be drawn independent of engine speed. The results will firstly present
the velocity profiles of each inlet regime and respective piston bowl with changing
engine speed. Velocity profiles for each engine speed for the different inlet
configurations are then considered. The first 5 mm of the bowl radius was omitted
from these results due to the large errors in that area and the flow in this region does
not significantly affect the overall properties of the swirl.

7.6.1 Combined inlet velocity profiles

![Combined inlet velocity profiles](image)

Figure 7.34 Variation of normalised dynamic swirl ratio across the piston
bowl for combined inlet, deep bowl piston
Combined inlet, results in Figures 7.34 and 7.35 displayed different velocity profiles across the bowl for the different piston bowls. The deep bowl piston (Figure 7.34) showed a good correlation of the velocity data for the three engine speeds up to a radius of 18 mm. In the outer portion of the bowl (i.e. 18 mm < radius < 28 mm) the swirl speed increased linearly with engine speed. The profile, however, did not display a solid body rotation. From the turbulence intensity map, Figure 7.4, the fluctuating component of the velocity exhibited random areas of low and high turbulence intensity with engine speed for radii < 18 mm. The increase in the turbulence intensity towards the outer radii (18 mm < radius < 28 mm) of the piston bowl suggested a more linear relationship with engine speed, where the fluctuating component was seen to increase in all areas. Since the velocity profiles in the deep bowl indicated a similar profile that increased with engine speed, a more comprehensive model of this data could be created, aiding in better modelled results.

The shallow bowl piston (Figure 7.35) displayed a closer approximation to solid body rotation for radii < 28 mm. Again, the swirl increased with increasing engine speed, however, with the shallow bowl arrangement the majority of the flow (radii < 35 mm) was seen to increase. The mean result presented in Figure 7.35 tended towards
greater fluctuations when compared with the deep bowl piston result in Figure 7.34. This is most evident with the 1200 rpm result, where a highly fluctuating flow was shown. This fluctuating flow was seen to be repeated for different experiments and would suggest that the interaction between the directed and helical ports created a highly fluctuating flow. With reference to Figure 7.6, the turbulence intensity map showed that the area of greatest fluctuation occurred up to a radii of 25 mm. In the outer region of the flow (radii > 25 mm), the normalised swirl ratio decreased, which indicated a closer agreement of the normalised swirl remaining constant for radii < 30 mm for varying engine speed. The turbulence intensity map indicated a less fluctuating flow in this outer region, which was different from the helical and directed regime for the shallow bowl piston. Although the mean flow displayed greater variation in Figure 7.35, there is scope for better modelling of this motion, where there is a dependence on engine speed and a repeatable profile.

7.6.2 Directed inlet velocity profiles

![Graph showing directed inlet velocity profiles](image)

Figure 7.36 Variation of normalised dynamic swirl ratio across the piston bowl for directed inlet, deep bowl piston
Directed inlet results in Figures 7.36 and 7.37 displayed different velocity profiles according to the piston bowl employed. Figure 7.36 for the deep bowl showed an inverse parabolic velocity profile across the bowl, which decayed from the centre to a radius of approximately 17 mm and then increased until the boundary layer of the outer piston bowl wall was reached. This is a significant deviation from the expected solid body rotation that would be assumed in simple modelling. This profile deformed with increased engine speed. For radii > 17 mm, the normalised swirl decayed with increasing engine speed and a smaller boundary layer to the outer piston bowl wall was observed. For radii < 17 mm, the velocity profile displayed a more chaotic profile with increased engine speed. This is confirmed by the turbulence intensity map, Figure 7.8, where as engine speed was increased to 1600 rpm the fluctuating component increased dramatically. The curved fit of the velocity profile showed the mid portion of the piston bowl to be significantly altered, this was expected to be due to the squish flow interaction.

The shallow bowl exhibited a different profile to the inverse parabolic observed for the deep bowl. With lower engine speeds a parabolic curve exists which is the inverse of the parabolic as displayed by the deep bowl piston. The profile changed with
increased engine speed and displayed a better approximation to solid body rotation at 1600 rpm. Overall, normalised swirl rate decreased with increasing engine speed and the boundary layer to the outer piston bowl wall reduced in width. Referring to the turbulence intensity map in Figure 7.10, the fluctuating component of velocity increased rapidly at 1600 rpm, this suggested that the mean normalised swirl result decreased with increasing engine speed and that increased fluctuating velocities account for a greater part of the flow motion. As previously mentioned for combined inlet, the squish interaction had less affect on the flow and this was confirmed by the flatter profile, which represented solid body rotation.

7.6.3 Helical inlet velocity profiles

![Image of data points for normalised dynamic swirl ratio across the piston bowl for helical inlet, deep bowl piston](image)

Figure 7.38 Variation of normalised dynamic swirl ratio across the piston bowl for helical inlet, deep bowl piston
Helical inlet results in Figures 7.38 and 7.39 suggested that the normalised swirl ratio increased with the deep bowl piston for increasing engine speed and decreased with the shallow bowl piston for increasing engine speed. The deep bowl piston presented a poor correlation with solid body rotation. For radii < 13 mm the mean results are highly variable and there are two main factors that may cause this. The first is the squish and secondly the off-centre swirl regime set-up by the helical inlet. Figure 7.12 shows the turbulence intensity maps for the deep bowl piston. It was shown that the largest amount of fluctuating velocity component occurs to the outer radii of the piston bowl, however, about the centre of the piston bowl there were areas of low fluctuation, which would be indicative of the low velocities seen about the radius. The velocity profiles for each engine speed displayed similar gradients for radii > 13 mm, but there is a marked increase in the 1600 rpm result.

Shallow bowl results presented a significantly flatter profile which was a better agreement with solid body rotation theory. Velocity profiles for the three engine speed results exhibited a similar parabolic curve, but a minor decrease with increasing engine speed. This decrease was of interest, since with the deep bowl piston the swirl
increased with engine speed, which was expected to be repeated for the shallow bowl. This result may be due to squish interaction and off centre swirl.

7.6.4 Velocity profiles at engine speed 800 rpm

800 rpm engine speed, deep bowl piston results in Figure 7.40 gave significantly different swirl profiles according to the inlet port tested. The directed inlet produced the most swirl at this engine speed, with particularly higher swirl towards the outer piston bowl wall. It also displayed higher rates of swirl, compared to the helical and combined inlets, for radii < 15 mm. Helical inlet exhibited a similar profile to the directed inlet (for radii > 15 mm) with slightly lower normalised swirl ratio, with the maximum occurring at the outer piston bowl wall. The combined inlet was shown to increase until it coincided with directed and helical results for 15 mm < radius < 20 mm. It then exhibited a decline in swirl from this 20 mm radius until the bowl edge was reached, where as previously discussed the radial component of velocity is greater. This higher rate of swirl towards the piston bowl wall, shown by the helical and directed result, would influence the results of the steady-flow test rig to produce higher mean swirl numbers, due to the outer radii having a greater influence on the overall swirl number indicated.
Figure 7.41  Variation of inlet configurations at 800 rpm, shallow bowl piston

800 rpm engine speed, shallow bowl piston results in Figure 7.41 present a better correlation with the steady-flow test rig results. The combined swirl is seen to be approximately half of the directed and helical results. The helical result showed a slightly increased normalised dynamic swirl ratio over the directed result. The combined inlet gave a better approximation to solid body rotation for the majority of the piston bowl. The helical and directed inlets maintained a similar gradient and boundary layer thickness towards the piston bowl wall.
7.6.5 Velocity profiles at engine speed 1200 rpm

![Graph showing velocity profiles at 1200 rpm](image)

**Figure 7.42** Variation of inlet configurations at 1200 rpm, deep bowl piston

1200 rpm engine speed, deep bowl piston results in Figure 7.42 showed similar trends to Figure 7.40, where both the directed and helical ports maintained a similar gradient towards the outer edge of the piston bowl. Helical inlet had a slightly higher normalised swirl ratio for radii > 13 mm. The combined inlet was seen to increase and show a greater overlap with the other two inlet configurations for 15 mm < radius < 22 mm. The combined inlet then decreased as it approaches the bowl edge.
1200 rpm engine speed, shallow bowl piston results in Figure 7.43 showed a similar trend to Figure 7.41, the main difference being the directed inlet decreased in normalised dynamic swirl ratio, showing the helical inlet to be slightly increased over the directed result. All of the results at this test point exhibited good agreement for a scatter graph plot. The combined inlet normalised swirl is slightly increased when compared to the 800 rpm result, indicating that the swirl is more than half for the majority of the piston bowl in the combined inlet condition compared to the directed or helical inlet conditions.

Figure 7.43  Variation of inlet configurations at 1200 rpm, shallow bowl piston
7.6.6 Velocity profiles at engine speed 1600 rpm

Figure 7.44 Variation of inlet configurations at 1600 rpm, deep bowl piston

1600 rpm engine speed, deep bowl piston results in Figure 7.44 mark a significant change in the three inlet profiles. Directed inlet was shown to decrease in normalised swirl ratio consistently over the three engine speeds and displayed a velocity profile which increased non-linearly with radius. Helical inlet increased in normalised swirl ratio consistently with increasing engine speed and maintained a linear gradient of the flow up to the boundary layer of the piston bowl wall. The combined inlet further increased in normalised swirl ratio with increasing engine speed. In the 1600 rpm result, the combined inlet was shown to have greater swirl than the directed inlet for radii < 23 mm. The profile of the combined inlet had changed with the increasing engine speed, the peak of the curve was seen to become flatter and occur at greater radius with increasing engine speed. It also indicated the normalised swirl ratio was greater towards the piston bowl wall than the previous engine speeds, suggesting a smaller boundary layer.
1600 rpm engine speed, shallow bowl piston results in Figure 7.45 indicate that the directed inlet regime normalised swirl ratio, as was seen in Figure 7.44, decreased the greatest over the three engine speeds. Helical inlet normalised swirl ratio was also seen to decrease over the three engine speeds, but to a lesser extent than the directed inlet. The combined inlet increased normalised swirl ratio with increasing engine speed. Combined inlet also exhibited that for radii > 28 mm the normalised swirl ratio declines. This is not the case for the directed and helical inlet where they are observed to be constant or increase until the boundary layer is reached.

7.7 Holographic PIV (HPIV)

7.7.1 Introduction

In conjunction with this current research, an advanced method of HPIV has been developed to measure the in-cylinder flows on the Perkins Trailblazer engine. This novel technique provides detailed 3-D analysis of the in-cylinder flows within the
piston bowl. The general HPIV technique was described in Section 3.5.4, in this section the specific arrangement of the HPIV set-up is described and the method for obtaining the holograms outlined. A holographic data result will then be presented showing the 3-D flow regime in the piston bowl and compared directly with 2-D PIV data. The HPIV testing described in this section was carried out by Dr. R.D. Alcock of Loughborough University, this section outlines the important aspects of the HPIV technique with respect to the Perkins Trailblazer optical engine. Further information of this technique and the interrogation methods can be found in Alcock et al. (2004).

7.7.2 HPIV experimental set-up

In most parts, the basic experimental set-up is identical to the 2-D PIV that was described in Chapter 6. The engine configuration and auxiliary equipment, deep bowl piston, particle seeding unit, motor controller and timing unit all remained the same. However, in the HPIV technique the optical set-up is considerably different, as are the requirements of light displacement properties of the seeding chosen. The in-cylinder flows for the HPIV testing were measured at 750 rpm (this data point chosen to be the same frequency as the Nd:YAG Spectron laser operating at 12.5 Hz) and used the combined inlet ports regime with the deep bowl piston.

The HPIV technique allowed for accurate recording of images through thick curved cylinder walls, in this case the piston cassette glassware and the optical cylinder extension, which was an advantage over the 2-D PIV technique. Perspective errors (described in Section 6.8.4 for 2-D PIV) do not exist with this method since the reconstruction of the hologram in a dummy piston allowed for true displacement measurement instead of a calibrated image size. Measurement of the third component of velocity enabled greater understanding of the complex flow behaviour. It was shown with the 2-D PIV results that a radial flow component existed and it was believed to be a toroidal flow pattern within the bowl (described in Section 2.4). However, without the third component of velocity or multiple-plane 2-D PIV measurement this result cannot be confirmed. The knowledge of the third component of velocity enables a full picture of the flow behaviour.
HPIV was recorded in an auto-correlation mode, whereby double exposure images of the flow were recorded to a silver halide holographic plate that was mounted behind the engine (arrangement shown in Figure 7.46). Two coherent, high energy laser pulses were fired with a short known time separation between the pulses. The beam was split using beam-splitter optics to create the object beam and the reference beams. Two collimated reference beams exposed the hologram plate from the same side as the scattered light from the particles in the flow. This exposed the holographic plate with two separate hologram images. A known image shift was incorporated into the holograms, thus removing directional ambiguity of the particle displacements.

![Diagram of Hologram Plate and Laser Beams](image)

Figure 7.46 Layout of HPIV method recording holograms (Alcock et al. 2004)

Once holograms were recorded, it was necessary to develop the exposed hologram plate in a similar method to photographic film development. The hologram was then reconstructed in a dummy optical cylinder and inspected by an optical fibre probe using a novel technique called optical conjugate reconstruction (OCR) (Alcock et al. 2004). The layout of this reconstruction is shown in Figure 7.47.
7.7.3 HPIV particle seeding

The HPIV technique required large particles with good flow following capabilities. From the testing which used the high speed flow visualisation, the original preferred seeding was Expancel, (see Table 6.2). However, it was found that Expancel could not survive the compression stroke as the particles were subject to compression ignition. As previously discussed in Section 6.8.2, hollow glass micro spheres could be used but with the detriment of damaging the engine (glass particles damaging cylinder contact surfaces). The hollow glass micro spheres used had a mean diameter of 100 μm with a distribution of 90 to 120 μm diameter, density of 200 kg.m\(^{-3}\) and a crush pressure of approximately 70 bar. These particles were imaged using the flow visualisation equipment and found to survive the compression phase of the engine and were capable of scattering enough light to make them suitable for HPIV.
7.7.4 HPIV result compared to 2-D PIV result

A preliminary 3-D HPIV result is now presented here and compared with a 2-D PIV result under similar engine operating conditions. Figure 7.48 displays the HPIV result, which shows the laser beam passing under the cylinder head window. When a laser capable of producing high energy beams is available, it will enable whole field results in the piston bowl to be measured, whilst using smaller particle seeding with increased flow following capability and increased resolution of vectors per unit volume. This result is followed by the same result arranged orthogonally in Figures 7.49 and 7.50 for direct comparison with the 2-D PIV result. The accuracy of the HPIV system allowed for very small displacements of particle shift (measurable displacements in the region of 0.3 to 10 μm) which was an advantage over the 2-D PIV system where error in small displacements (sub-pixel) can be large (shown in Chapter 6 up to 12% RMS error). HPIV also provided a greater number of vectors per unit volume when compared with the 2-D PIV method, which allowed for analysis of smaller turbulent structures in the flow.

The HPIV result was taken at 750 rpm engine speed, combined inlet and used the deep bowl piston. The 2-D data was obtained at 800 rpm engine speed, combined inlet and deep bowl piston.
Figure 7.48 3-D HPIV results and orientation in engine cylinder

Figure 7.49 Top view of HPIV results

Outline of 2-D PIV cylinder head window

Exhaust valve

Cylinder head window

Directed inlet valve

Helical inlet valve

Velocity (m.s⁻¹)

Velocity (m.s⁻¹)
Figure 7.50 Orthogonal HPIV vector maps of flow in piston bowl

Figure 7.51 Instantaneous 2-D PIV result at 800 rpm, combined inlet, deep bowl piston
Figure 7.52  Ensemble averaged 2-D PIV result at 800 rpm, combined inlet, deep bowl piston

The 3-D HPIV and 2-D PIV results (Figure 7.51 and 7.52) showed a good visual agreement of the flow experienced under similar test conditions. As the flows are highly variable, it was difficult to directly compare results of two instantaneous results. Figure 7.52 shows the ensemble averaged result for 2-D PIV, which gave a good agreement with the 3-D HPIV result. However, from observation, similar velocity magnitudes exist showing the central area of the flow to be of low velocity (< 3 m.s\(^{-1}\)) and the outer edge to have higher velocities (4 – 8 m.s\(^{-1}\)).

The side view of Figure 7.49 showed the z-component of the flow, the motion of this z-component indicated out of plane flows that the 2-D PIV could not measure. This flow shows the toroidal pattern that is set-up in the piston bowl, as was described in Chapter 2. The flow toward the geometric centre of the piston bowl exhibited larger z-component motion relative to the x and y components of motion. Since 2-D PIV images a plane of flow (determined by laser sheet thickness), this result indicates that areas of sparse data towards the centre of the bowl would be due to the out of plane flows experienced. As more HPIV results are obtained, there is considerable scope for more comparison with 2-D PIV data.
7.8 Summary

This chapter presented a variety of results obtained in the optical CI engine. Results were processed using code written in the Matlab language, which enabled further analysis of the flow. The first set of results presented raw data from the engine (a mean and an instantaneous vector map for each engine test point) which was validated with the criteria set out in Table 6.5. Results were then interrogated for their mean normalised dynamic swirl ratio and a turbulence intensity map at each of the 18 test points was presented. These were visually compared to the basic simple swirl theory. A discussion of these results followed. Measured data was then compared with the model that was presented in Chapter 5, which showed varying correlation with the modelled results. The shallow bowl model was observed to exhibit better comparable results with the measured data than the deep bowl model. As the model was based on a simple flat bottomed piston, the shallow bowl piston approximated this design better than the deep bowl piston. The squish interaction with the deep bowl was also greater and was likely to have affected the deep bowl piston results when compared with the model. A discussion of the model and measured data was then presented. An important result was that the suggested velocity profile 2 from the model best represented the measured results in most instances, which was a positive quadratic profile. There is scope for further work to define this velocity profile better. However, it was seen that the velocity profile did change with the inlet port used, hence for a tuneable velocity profile it would have to be based on the type of inlet port and its swirl number and not based on swirl number alone.

The velocity profiles of each inlet were then compared with engine speed. This indicated that a velocity profile existed for each inlet regime that varied but maintained a similar profile with engine speed. Directed inlet velocity profile was seen to decay with increased engine speed. The combined inlet was seen to increase with increased engine speed. Helical inlet deviated in that the velocity profile increased with engine speed for the deep bowl, but decreased with increased engine speed for the shallow bowl.
Velocity profiles for each inlet were then considered for the relative engine speed and piston bowl. The deep bowl piston velocity profiles for the helical and directed inlets showed an increase with radius until the boundary layer was reached. Combined inlet displayed a parabolic curve which had equivalent swirl levels in the mid region of the piston bowl, but decayed toward the outer radius of the piston bowl. The shallow bowl piston exhibited a better approximation towards the theory of solid body rotation than the deep bowl results. Directed and helical inlets were seen to decline in swirl with increased engine speed, whereas the combined inlet swirl rate increased with engine speed.

The technique of holographic PIV, which was also employed on the optical CI engine, was then introduced. The experimental set-up and method for interrogating the information from the holographs was briefly described. An example holographic PIV result was then given for the combined inlet regime at 750 rpm and was compared to an instantaneous 2-D PIV result recorded for the combined inlet regime at 800 rpm. These results exhibited a good correlation of the velocity vectors in the piston bowl with the 2-D data. The holographic PIV technique has significant future potential and is currently still in development.
Chapter 8  Conclusions and Future Work

8.1 Conclusions

The objectives of this work was to use PIV measurement techniques in a CI DI engine for varying inlet configurations, engines speeds and piston bowls. From these measurements a better understanding of the charge air fluid motion at TDC has been gained. These objectives were met, providing a method for measuring in-cylinder flows in a production geometry CI DI engine, with piston-bowl pip and high compression ratio.

An optical cylinder head and piston bowl was developed for this work, with the cylinder head employing a directed and helical inlet port that returned similar steady-flow swirl rig results. These inlet ports were tested both independently and in combination. A shallow bowl and deep bowl piston were used to compare the affect of the piston on the in-cylinder flow. Tests were carried out for three different engine speeds to determine whether normalised dynamic swirl ratio was maintained with crankshaft speed. A swirl spin-up model was proposed based on the steady-flow swirl rig results of the optical cylinder head, these results were directly compared with measured data of the engine. Due to the nature of in-cylinder flows, data at each test point were ensemble averaged for 400 vector maps to show the average flow characteristics and presented the fluctuating component of the flow as turbulence intensity maps. The major findings of this study are summarised as follows:

1. The approximation of solid body rotation of the charge air motion during compression does not hold true. The deep bowl piston for all three inlet configurations showed deviation from solid body theory, which was accentuated with increasing engine speed. The shallow bowl piston exhibited better approximation to solid body rotation. This would suggest that the squish flows, which are considerably higher with deep bowl pistons, have a significant affect on the charge air motion at TDC.
2. Steady-flow swirl rig results suggested that the swirl in the engine for the combined inlet ports would be approximately half of either the helical or directed inlet port. The measured data showed that the combined inlet in the deep bowl piston returned similar swirl levels between 35% and 75% of the bowl radius. The combined inlet presented a weaker level of swirl towards the geometric centre of the piston. However, the turbulence intensity of the combined inlet was observed to be similar to the levels of the single port inlet configurations, which suggested a highly fluctuating flow in this region. Shallow bowl results exhibited a better correlation with the steady-flow swirl rig based modelled results for lower engine speeds. However, with increased engine speed the directed and helical inlet swirl ratio decreased whereas the combined inlet was seen to increase. At crankshaft speed of 1600 rpm, the swirl level of the combined inlet was approximately 25-30% less than the directed or helical inlet, the steady-flow swirl rig results suggested this to be 50% less.

3. The combined inlet ports showed a parabolic velocity profile across the bowl, this was highlighted with the deep bowl piston. Swirl ratio towards the outer region of flow was seen to decay from approximately 75% piston bowl radius to the piston bowl edge. It was observed when comparing the modelled and measured data that the radial velocities in the combined inlet were greater than those exhibited by the directed and helical inlet configurations. The helical and directed inlets indicated increased swirl ratio in this same region until the boundary layer of the piston bowl wall was reached. This greater swirl rate towards the outer radii of the piston bowl would explain why steady-flow swirl rigs return higher swirl ratio numbers than the combined inlet. The effect of this increased swirl ratio towards the bowl edge exaggerated the overall flow profile of the inlet regime.

4. The velocity profile of the combined inlet returned a similar increasing curve which increased with engine speed, with a peak occurring between 70% and 80% piston bowl radii. This was applicable for deep and shallow bowl piston geometry. The directed velocity profile for deep bowl piston displayed a negative parabolic curve, where the mid-section of piston-bowl radii decayed in swirl, this velocity profile held for different engine speeds and presented
decreasing swirl rate with increased engine speed. Shallow bowl result also demonstrated decreased swirl rate with increased engine speed, but the reduced swirl in the mid-section of the piston bowl was substantially less. Helical inlet for the deep bowl piston showed increasing swirl rate across the piston which increased with engine speed. However, the shallow bowl piston exhibited a lower gradient in swirl rate across the bowl, which decreased with increased engine speed.

5. Modelled data correlated well with certain results up to a boundary layer, which the model did not consider. The combined inlet with the deep bowl piston showed a good approximation between measured and modelled data between 0 to 80% piston bowl radii. Directed inlet with deep bowl piston returned a poor correlation with results, where the model over-predicted the swirl rate and the velocity profile gradient increased toward the outer radii. Helical inlet gave a better correlation with results with increasing engine speed for the deep bowl piston. All three inlet configurations for the shallow bowl correlated well with the modelled data.

6. The squish interaction with the swirl flow was greater than first expected. Various literature has reported the squish flow to peak at 10° BTDC and decay rapidly to zero by TDC, whilst this may be true, the effect of the squish flow on the swirl at TDC was clearly seen in the deep bowl results, suggesting the full interaction of squish and swirl had a major affect on the flowfield at TDC.

7. Caterpillar Engines are currently designing new CFD code for advanced design of CI engines. The next stage of this project will be to destructively test the optical cylinder head so as to obtain the exact details of the inlet ports and to accurately model them in this new code. The measured results presented in this thesis will be used as validation of the CFD models.

8. Directed inlet was observed to produce lower swirl motion when compared to helical and combined inlets, this is likely to be due to the volumetric efficiency of directed inlets being less efficient, in effect partially limiting the flow with increased engine speed.
9. Helical inlet displayed the highest turbulence intensity toward the outer edge of the flow. From previous results, helical inlets have returned improved emissions results compared with directed inlets, this may be due to the increased turbulence levels exhibited.

10. The 2-D PIV results compared well with the 3-D HPIV presented in Chapter 7 for the combined inlet, deep bowl piston 800 rpm test point. A comprehensive comparison for the same spatial domain needs to be investigated for a complete analysis of the in-cylinder motion. A larger dataset of the HPIV is required before this can be achieved.

8.2 Future work

This research has aided in furthering the understanding of the air charge motion at TDC. Due to experimental limitations of the engine it was not possible to research the squish flows of the engine, which from this work is seen to be a significant contributor to the TDC charge air motion. With a fuller understanding of the interaction of these two flows, inlet port and piston design can be improved to aid in more efficient combustion and improved emissions.

Due to the cylinder head being modified for optical access in the central fuel injector region, it was not capable of being used on a fired CI DI engine. However, by gaining optical access at the central region it allowed for velocity profiles of the flow to describe the motion across the piston bowl from the geometric centre of the piston. As this work has shown, the air charge motion about the geometric centre has little affect on the overall flowfield. If the optical access was gained through an exhaust valve, or holographic techniques used, the injector could be used to inject an appropriate dopant, with similar properties to diesel fuel, which could be illuminated by the LIF arrangement. This would give an insight into the exact fuel-air mixing experienced within a CI DI engine. To aid further in the understanding of these in-cylinder results, it is suggested that a replica of this cylinder head is manufactured for
use with a fired engine so that engine performance and emissions data can be compared directly with the measured results presented in this thesis.

The holographic PIV technique has been discussed and example data shown. This technique is still being developed and it will allow for full 3-D spatial understanding of the in-cylinder flow.

The current time resolved PIV equipment that has become commercially available can operate PIV at 5 kHz, this equates to a PIV recording being taken every 1.9° CA at an engine speed of 1500 rpm. This could show the developing flowfield of the engine during the compression stroke and aid in further understanding the cyclic variation and turbulent structures present in the engine. With a time history to the developing flowfield, POD (Proper Orthogonal Decomposition) techniques could be applied to determine cyclic variation, which in this present study was not considered individually but as a part of the turbulence intensity.

A re-design of the optical CI DI engine could allow for greater optical access and improved refit time, allowing the cleaning of the optical components inside the engine and reducing down time of the engine between tests.

A CFD model of the optical engine geometry is planned. This will be carried out by first extracting the exact data of the inlet ports by destructive testing and using the exact inlet port data in the CFD model. Results of this CFD model would be compared with the results presented in this thesis. This would give validation data and aid in the future design of CFD programming with respect to engines.
References


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Macmillan Press Ltd.


Appendix A1  Perkins report on swirl calculations

Author: Penny Emonson
Report No: 17138/PENNY/TR01
Date: 22-04-1998

1. Introduction

This report provides a detailed understanding of the calculations that are used to obtain the swirl results from the swirl measurement rig. It also details the origins of the equations. The approved methods of calculation which use these equations are:
concerto, v:\winmca\mathpack\swirl.mcd and q:\swirl\swirl.xls

2. Flow Rate Calculations

These are taken from the MathCad file which was based upon the original main frame calculations. The equations have been compared to BS1042, and all except the expansibility equation can be traced back to the standard. The nomenclature used is described in Appendix 1.

2.1 Coefficient of Discharge

The discharge coefficient is a comparison of the ideal flow with the actual flow that occurs through the nozzle. The actual flow is always less than the ideal flow due to losses (e.g. frictional).

The coefficient of discharge for the nozzle used in the flow rig is obtained from a best fit line through data obtained off the nozzle at the National Engineering Laboratory in 1973. Details are included in Appendix 2.

\[ C_d = 0.008535 \times \log(P, u.) + 0.9568 \]  
Equation 1

2.2 Effective Diameter

The effective diameter calculates the change in diameter due to the increase in temperature. It uses the linear expansion equation:

\[ \text{expansion} = \alpha L \Delta T \]  
Equation 2

Where \( \alpha \) is the linear expansion coefficient for the materials. The throat of the nozzle is made of brass, and the upstream pipe is made of steel.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Material</th>
<th>Expansion Coefficient[10^{-6}]</th>
</tr>
</thead>
<tbody>
<tr>
<td>d ( d_p )</td>
<td>brass</td>
<td>18.9 \times 10^{-6} K^{-1}</td>
</tr>
<tr>
<td>D ( d_p )</td>
<td>steel</td>
<td>11.7 \times 10^{-6} K^{-1}</td>
</tr>
</tbody>
</table>

\[ d_{\text{eff}} = d \left(1 + \alpha_d \times (T_t - T_{\text{up}})\right) \]  
Equation 3

\[ D_{\text{eff}} = D \left(1 + \alpha_p \times (T_t - T_{\text{up}})\right) \]  
Equation 4

2.3 Velocity Approach Factor

\[ \nu_{\text{eff}} = \frac{1}{\sqrt{1 - \beta^4}} \]  
Equation 4

where diameter ratio \( \beta \): \[ \beta = \frac{d_{\text{eff}}}{D_{\text{eff}}} \]  
Equation 5
2.4 Expansibility Factor

The expansibility factor is the coefficient which takes into account that the fluid in use (air) is actually compressible.

\[ e = 1 - 0.04332 \times \frac{P_{\text{mb}}}{P_t} \]  

Equation 6

I can find no explanation of this equation, in BS1042, the expansibility factor is expressed in terms of the ratio of the pressures either side of the nozzle.

2.5 Density of Air

2.5.1 Relative Humidity

The relative humidity (\(\phi\)) is the amount of moisture in the air (m_v) in comparison to the maximum amount of water the air can hold at the same temperature (m_g)^y.

\[ \phi = \frac{m_v}{m_g} = \frac{P_v V / R_v T}{P_g V / R_g T} \]

Equation 7

The ambient pressure can be considered to be the sum of the partial pressure of air and the partial pressure of vapour:

\[ P_0 = P_a = P_v \]

Equation 8

The saturation pressure can be calculated from a best fit line to the data supplied in the steam tables\(^{\text{vi}}\). See Appendix 3

\[ P_g = 10^{\left( \frac{24315}{-40} \right)} \]

Equation 9

2.5.2 Density

Using PV=mRT

\[ \rho = \frac{P}{RT} = \frac{P_v}{R_v T} + \frac{P_a}{R_a T} \]

Equation 10

But from Equation 8: \( P_a = P_0 - P_v \)

\[ \rho = \frac{P_0}{R_u T} - \frac{P_v}{R_u T} + \frac{P_v}{R_v T} \]

Equation 11

From Equation 7

\[ P_v = \phi P_g \]

Equation 11

\[ R_a \quad 287.1 \text{ J/kgK}^{\text{u}} \]
\[ R_v \quad 461.5 \text{ J/kgK}^{\text{u}} \]
\[
\rho = \frac{P_0}{287.17T_0} - \frac{\phi P_e}{287.17T_0} + \frac{\phi P_e}{461.57T_0} \\
= \frac{3.48 \times 10^{-3}}{T_0} \left( P_0 - 0.3778\phi P_e \right)
\]

But the pressures are measured in mmHg, and the calculation needs them to be in Pa, therefore the Equation 11 needs to be multiplied by 133.322

\[
\rho = \frac{0.4629}{T_0} \left( P_0 - 0.3778\phi P_e \right) \quad \text{Equation 12}
\]

### 2.6 Volume Flow Rate

This is the volume flow rate through the nozzle, it can not be said that it is the same though the valve, due to the compressibility of the fluid.

In BS 1042,

\[
V = E \times d_{eff}^2 \times \varepsilon \times C_d \times \frac{\pi}{4} \times \sqrt{\frac{P_{atm} \times 2}{\rho}}
\]

Bringing together the constants

\[
V = \left( \frac{\pi \sqrt{2}}{4} \right) \times E \times d_{eff}^2 \times \varepsilon \times C_d \times \sqrt{\frac{P_{atm}}{\rho}}
\]

Converting to the correct units

\[
V = \left( \frac{\pi \times \sqrt{2}}{4 \times 10^5} \right) \times E \times d_{eff}^2 \times \varepsilon \times C_d \times \sqrt{\frac{P_{atm} \times 9.9065}{\rho}} \quad \text{Equation 13}
\]

### 2.7 Corrected Volume Flow Rate

This corrects the volume flow rate to standard temperatures and pressures so various tests can be compared.

\[
V_c = V_a \frac{P_2}{P_{atm}} \frac{T_{sp}}{T_2} \quad \text{Equation 14}
\]

### 2.8 Mass Flow Rate

The mass flow rate, allows examination of the valve. The mass flow rate though the nozzle and though the valve are the same due to conservation of mass.

\[
m_u = V \times \rho \quad \text{Equation 15}
\]
3. Valve Calculations

3.1 Effective Area of the Valve

The valve can be treated as a nozzle. The actual flow through the valve is limited by frictional losses. The effective area describes the cross-sectional area of a frictionless nozzle which would flow the actual mass flow.

Using the 1 dimensional steady flow calculation, which can be derived for an isentropic fluid from the continuity equation and the energy equation.

Isentropic fluid \( P v'' = \text{constant} \) \hspace{1cm} \text{Equation 16}

Energy equation:
\[
h_2 + \frac{C_2^2}{2} = h_1 + \frac{C_1^2}{2}
\]

Continuity equation:
\[
\frac{A_2 C_2}{v_2} = \frac{A_1 C_1}{v_1}
\]

If you assume that the initial velocity \( (C_1) \) is zero

\[
\frac{1}{2} (C_2^2 - C_1^2) = -\int_v dp
\]

\[
\frac{1}{2} C_2^2 = \frac{n}{1-n} (p_2 v_2 - p_1 v_1)
\]

substituting with Equation 16

\[
C_2 = \left[ \frac{2n}{1-n} p_1 v_1 \left( \left( \frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right) \right]^{\frac{1}{2}}
\]

Equation 19

\[
m = A_2 C_2
\]

From continuity equation

\[
\frac{A_2}{m} = v_2 C_2 = \frac{v_1}{C_2 (p_2 / p_1)^{\frac{1}{n}}}
\]

Substituting Equation 19 and \( PV = mRT \) gives

\[
m = P_1 \left[ \frac{2\gamma}{1-\gamma} \frac{1}{RT_1} \left( \left( \frac{P_2}{P_1} \right)^{\frac{\gamma+1}{\gamma}} - \left( \frac{P_2}{P_1} \right)^{\frac{2}{\gamma}} \right) \right]^\frac{1}{2}
\]

\[
A = \frac{m}{P_1 \left[ \frac{2\gamma}{1-\gamma} \frac{1}{RT_1} \left( \left( \frac{P_1}{P_0} \right)^{\frac{\gamma+1}{\gamma}} - \left( \frac{P_1}{P_0} \right)^{\frac{2}{\gamma}} \right) \right]^\frac{1}{2}}
\]

Equation 20
The original programme used a slightly form of the equation, but the results obtained are the same.

3.2 Valve Port Flow Coefficient

The valve port flow coefficient is an indication of how efficient the valve is at various valve lifts. It is calculated by comparing the actual valve area with the effective area calculated in Equation 20.

\[ C_f = \frac{A_v}{n_{\text{valves}} A_{\text{valve}}} \]

Equation 21

Where

\[ A_{\text{valve}} = \frac{\pi}{4} D^2 \]

The nearer the figure to one, the more efficient the valve.

4. Swirl Calculations

The swirl calculations used are based on Tippelmann's equations and AVL's method of comparing swirl numbers across various engines.

4.1 Swirl Rate

Tippelmann's theory assumes that the angular momentum inside the cylinder can be equated to the force exerted on the rectifier inside the cylinder.

\[ T = l \]

Where \( T \) - torque, and \( l \) angular momentum flux.

Assuming the flow can be split into two velocities, the axial, and the circumferential velocity.

The circumferential velocity \( (cu) \) is responsible for the angular momentum

\[ dl = r \cdot Cu \cdot (r, \varphi) \, dm \]

The mass flow is caused by the axial velocity \( (ca) \)

\[ dm = C_{ca} (r, \varphi) \rho \cdot rdr \cdot d\varphi \]

Combining the two equations gives:

\[ T = l = \int \int r^2 ds = \int \int \rho C_s (r, \varphi) C_{ca} (r, \varphi) r^2 \, drd\varphi \]

Equation 22

Assuming:

\[ C_u = \text{const} \]

\[ C_{ca} = \text{const} \]
and substituting into equation 22:

\[ T = \omega p C_a \frac{R^4}{4} \cdot 2\pi \]

Equation 23

Now:

\[ C_a = \frac{V}{\pi R^2} \]

Therefore

\[ T = \omega p V \frac{R^2}{2} \]

Re-arranging:

\[ \omega = \frac{2T}{mR^2} \]  

Equation 24

The swirl Rate (\( \omega \)) is the velocity that the rectifier spins at.

From the measurements during the test:

\[ T = k_{reg} \cdot \text{swirl}_v \cdot K_{method} \cdot g \]  

Equation 25

Combining this into equation 24, (assuming units are as stated in appendix 1):

\[ \text{swirl(rads} / s) = \omega = \frac{39.165 \times 10^{-3} \times k_{reg} \times k_{method} \times v_{\text{swirl}}}{D_{core} \times m_a} \]  

Equation 26

4.2 Swirl Ratio

The swirl ratio is a method of linking the cylinder charge movement (swirl) calculated above with the engine speed. The method used is the AVL method\(^8\) of comparing the swirl with a fictitious engine speed.

\[ \text{swirl ratio} = \frac{n_{\text{swirl}}}{n} \]  

Equation 27

Where \( n_{\text{swirl}} \) is the swirl speed (swirl), \( n \) is the fictitious engine speed gained from equating the mean piston speed with the axial component of velocity.

Mean velocity in axial direction \( c_a = \frac{V}{\pi R^2} \)

Mean Piston Speed: \( c_m = 2sn \)

\[ n = \frac{c_m}{2s} \]  

(rps)

\[ = \frac{c_m \times \text{rev}}{2s} \]  

(rad / s where rev = 2\pi)
Equating the mean piston velocity to the mean axial velocity. Assuming that the positive direction is upwards, the mean piston speed is negative when the axial velocity is positive, therefore:

\[ n = \frac{V_\text{rev}}{V_\text{oil} \cdot 2} \]

Substituting this into equation 27:

\[ \text{Swirl ratio } R_{\text{ndn}} = \frac{n_d}{n} = -\frac{\text{swirl} \cdot V_\text{oil} \cdot 2}{V_\text{a} \cdot \text{rev}} \]

Equation 28

5. Mean Swirl Number and Flow Coefficient Calculations

According to the AVL method, the entire movement of the cylinder charge can be characterized by the mean swirl number.

Mean swirl number \( \left( \frac{n_d}{n} \right) \)

This is based on the following assumptions:

- the sum of moments of momentum during the intake process is equal to the moment of momentum of the entire cylinder load
- each moment of momentum can be characterized by the swirl ratio \( (n_d/n) \) for one particular valve lift.
- Instantaneous piston speed is taken into account, but the compressibility of air is neglected.

This results in the following equation:

\[ \left( \frac{n_d}{n} \right) = \frac{1}{\pi} \frac{\text{RXX} (\alpha - n)}{\text{SDX} (\alpha - 0)} \int n_d \left( \frac{c}{c_m} \right)^2 d\alpha \]

Equation 29

5.1 Instantaneous Crank Speed to Mean Speed Ratio

From the AVL report, the instantaneous speed to mean speed ratio is:

\[ \frac{c}{c_m} = \frac{\pi}{2} \sin \alpha \left( 1 + \frac{\cos \alpha}{\sqrt{R^2 - \sin^2 \alpha}} \right) \]

Equation 30

where \( R = \text{length of con rod} / \text{stroke} \)

5.2 Swirl Number

The swirl number for each individual valve lift can then be calculated by combining equation 30 into equation 29.

\[ \text{swirl number} = \left( \frac{c}{c_m} \right)^2 \left( \frac{n_d}{n} \right) \]

Equation 31
5.3 AVL Standard Valve Lifts

To be able to compare various ports, it is necessary to have the same non-dimensional cam to compare them with. AVL investigated the effect of valve lift on the mean port calculations and found that a movement of the inlet curve by 15° CA could decrease the mean port flow coefficient by 6%, and the mean swirl ration by 5%.

For this reason they developed a standard exhaust and inlet curve. Within the PTL calculations the inlet curve is used for both the inlet and exhaust.

The equation to describe this curve is:

\[
\frac{h_v}{D_v} = -3 \times 10^{-3}(\alpha - 105)^2 + 0.275
\]

where \( h_v \) is the absolute valve lift and \( D_v \) the inner seat diameter of the valve.

Using equation 32, a standard set of values for the non-dimensional valve lift can be obtained.

<table>
<thead>
<tr>
<th>° Crank Angle</th>
<th>Subscript</th>
<th>( h_v/D_v )</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1</td>
<td>0.032</td>
</tr>
<tr>
<td>30</td>
<td>2</td>
<td>0.106</td>
</tr>
<tr>
<td>45</td>
<td>3</td>
<td>0.167</td>
</tr>
<tr>
<td>60</td>
<td>4</td>
<td>0.214</td>
</tr>
<tr>
<td>75</td>
<td>5</td>
<td>0.248</td>
</tr>
<tr>
<td>90</td>
<td>6</td>
<td>0.268</td>
</tr>
<tr>
<td>105</td>
<td>7</td>
<td>0.275</td>
</tr>
<tr>
<td>120</td>
<td>6</td>
<td>0.268</td>
</tr>
<tr>
<td>135</td>
<td>5</td>
<td>0.248</td>
</tr>
<tr>
<td>150</td>
<td>4</td>
<td>0.214</td>
</tr>
<tr>
<td>165</td>
<td>3</td>
<td>0.167</td>
</tr>
</tbody>
</table>

To convert the tested valve lifts to this standardised form we need to divide them by the inner seat diameter, and perform a linear interpolation to calculate the swirl ratio at these lifts provided above (\( r_{nd,n,\text{std}} \)).

5.4 Mean Swirl Number

Using the standardised swirl ratios (\( r_{nd,n,\text{std}} \)), a mean swirl number can now be calculated using the Simpson's rule.

The area to be integrated (TDC to BDC) is split into twelve sections relating to 0, 15, 30...165, 180°. The swirl number (equation 31) is then calculated for each of these valve lifts, and then using equation 32, a mean swirl number is obtained.
mean swirl number = \[ \left( y_0 + y_2 + y_4 + y_6 + y_8 + y_{10} \right)^{4/3} \frac{4 + 2(y_1 + y_3 + y_5 + y_7 + y_9)}{36} \]

5.5 Mean Flow Coefficient

From AVL the equation for the mean flow coefficient, using the same assumptions as for the mean swirl number:

\[
\text{mean flow coefficient} = \frac{1}{\sqrt{\pi}} \left( \frac{c(\alpha)}{c_m} \right)^{3/2} \int_0^1 \left( \frac{c(\alpha)}{c_m} \right)^{1/2} d\alpha
\]

Equation 33

\(c/c_m\) is the ratio of instantaneous piston speed to mean piston speed, which has already been calculated.

Again to compare to a standard cam profile, a linear interpolation is carried out to find the values of \(c_t\) for the standard non-dimensional valve lifts.

If then the following calculation is carried out for each of these new flow coefficients, the simpsons rule can once again be used to calculated the mean value.

\[
c_{fad} = \left( \frac{1}{c_f} \right)^2 \left( \frac{c}{c_m} \right)^3
\]

Mean Flow Coefficient:

\[
c_{\text{mean}} = \frac{1}{\sqrt{\text{simpsons}(c_{fad})}}
\]

Equation 34

6. Recommendations

- Both the mean port coefficient and mean swirl number are integrated between TDC and BDC. In any future programs the option of using the actual valve event should be allowed. This would allow for the effect to be analysed to the datum of TDC to BDC.
- In the MathCad file only the AVL standard camshaft can be used. In any future program the option to use the actual camshaft details should be included. This again could then be referenced to the datum of the standard AVL cam Profile.
- At present only crankshaft and conrod data are provided for the 700 and New 1000 Series engines in concerto. Other engine series data should be available. Also only the AVL standard camshaft is used.
Appendix A: Measurement Positions and Normenclature

7. Measurement Positions

![Diagram showing measurement positions](image)

8. Normenclature

- Barometric Pressure: $P_0$ (mmHg)
- Ambient Temperature: $T_{0,c}$ (°C)
- Relative Humidity: $h_a$
- Chamber Depression: $P_{dep}$ (mmH$_2$O)
- Rig Calibration Constant: $k_{rig}$ (gm/volt)
- Bore Diameter: $D_{bore}$ (mm)
- Stroke: $L_{stroke}$ (mm)
- Con-Rod Length: $L_{rod}$ (mm)
- Valve inner seat diameter: $D_{valve}$ (mm)
- Number of inlet valves: $n_{valves}$
- Valve lift: $L_v$ (mm)
- Swirl Speed: $v_{swirl}$ (volt)
- Upstream gauge pressure: $P_{nu}$ (mmHg)
- Upstream temperature: $T_{2c}$ (°C)
- Head across nozzle: $P_{nh}$ (mmH$_2$O)

9. Conversion Factors

- $1\text{mmHg} = 133.322\text{Pa}$
- $1\text{mmH}_2\text{O} = 9.9065\text{Pa}$
• \( t(°C) = T(°K) - 273.15 \)

10. **Calculation of related inputs**
- Absolute pressure inside chamber \( P_1(Pa) \):
  \[
P_1 = (P_0 \times 133.322) - (P_{dep} \times 9.9065)
\]
- Absolute upstream pressure \( P_2(Pa) \):
  \[
P_2 = (P_0 - P_{nu}) \times 133.322
\]
- Absolute ambient temperature \( T_0(°K) \): \( T_0 = T_{oc} + 273.15 \)
- Absolute upstream temperature \( T_2(°K) \): \( T_2 = T_{2c} + 273.15 \)
- Swept Cylinder Volume \( V_{cyl} (mm^3) \):
  \[
  V_{cyl} = \left(\frac{\pi}{4} \times D_{bore}^2\right) \times L_{stroke}
  \]
Appendix B: Calibration of Coefficient of Discharge

The calibration of the nozzle was carried out by the department of Trade and Industry National Engineering Laboratory (NEL) in 1973. The nozzle was fitted into the NEL calibration line, and taper pieces were added to mimic the Perkins set-up.

The NEL calibration circuit flows water through the test line from a tank in a sump, where it is pumped back into the header tank. The flow is regulated by a control valve. The differential head across the nozzle was measured during the testing, and the results were calculated.

<table>
<thead>
<tr>
<th>Point No.</th>
<th>Flowrate (m³/s x 10⁵)</th>
<th>Line Temp (°C)</th>
<th>Flowmeter differential reading (mmH₂O)</th>
<th>Discharge Coefficient</th>
<th>Reynolds No. (x10⁶)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.635</td>
<td>25.9</td>
<td>0.107</td>
<td>0.9722</td>
<td>0.052</td>
</tr>
<tr>
<td>2</td>
<td>2.606</td>
<td>26.0</td>
<td>0.288</td>
<td>0.9790</td>
<td>0.099</td>
</tr>
<tr>
<td>3</td>
<td>5.744</td>
<td>23.3</td>
<td>0.103</td>
<td>0.9827</td>
<td>0.206</td>
</tr>
<tr>
<td>4</td>
<td>8.871</td>
<td>23.4</td>
<td>0.244</td>
<td>0.9860</td>
<td>0.318</td>
</tr>
<tr>
<td>5</td>
<td>12.60</td>
<td>23.7</td>
<td>0.491</td>
<td>0.9876</td>
<td>0.455</td>
</tr>
<tr>
<td>6</td>
<td>14.72</td>
<td>23.9</td>
<td>0.668</td>
<td>0.9891</td>
<td>0.535</td>
</tr>
<tr>
<td>7</td>
<td>10.72</td>
<td>24.5</td>
<td>0.355</td>
<td>0.9878</td>
<td>0.395</td>
</tr>
<tr>
<td>8</td>
<td>7.207</td>
<td>24.8</td>
<td>0.161</td>
<td>0.9860</td>
<td>0.267</td>
</tr>
<tr>
<td>9</td>
<td>10.73</td>
<td>25.4</td>
<td>0.354</td>
<td>0.9894</td>
<td>0.412</td>
</tr>
<tr>
<td>10</td>
<td>21.59</td>
<td>26.5</td>
<td>1.433</td>
<td>0.9898</td>
<td>0.830</td>
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<tr>
<td>11</td>
<td>18</td>
<td>26.8</td>
<td>0.998</td>
<td>0.9887</td>
<td>0.697</td>
</tr>
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<td>12</td>
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<td>1.211</td>
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</tr>
<tr>
<td>13</td>
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<td>1.433</td>
<td>0.9898</td>
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<tr>
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<td>13.65</td>
<td>27.8</td>
<td>0.573</td>
<td>0.9892</td>
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</tr>
<tr>
<td>16</td>
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</tr>
<tr>
<td>17</td>
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<td>0.592</td>
</tr>
<tr>
<td>18</td>
<td>3.580</td>
<td>25.6</td>
<td>0.505</td>
<td>0.9799</td>
<td>0.135</td>
</tr>
<tr>
<td>19</td>
<td>4.549</td>
<td>26</td>
<td>0.811</td>
<td>0.9821</td>
<td>0.173</td>
</tr>
<tr>
<td>20</td>
<td>5.742</td>
<td>26.2</td>
<td>1.292</td>
<td>0.9824</td>
<td>0.219</td>
</tr>
<tr>
<td>21</td>
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<td>26.3</td>
<td>1.525</td>
<td>0.9837</td>
<td>0.239</td>
</tr>
<tr>
<td>22</td>
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<td>26.4</td>
<td>0.178</td>
<td>0.9760</td>
<td>0.081</td>
</tr>
<tr>
<td>23</td>
<td>9.889</td>
<td>22.9</td>
<td>0.302</td>
<td>0.9880</td>
<td>0.365</td>
</tr>
<tr>
<td>24</td>
<td>16.20</td>
<td>23.8</td>
<td>0.811</td>
<td>0.9875</td>
<td>0.587</td>
</tr>
<tr>
<td>25</td>
<td>17.96</td>
<td>24.9</td>
<td>0.995</td>
<td>0.9886</td>
<td>0.667</td>
</tr>
<tr>
<td>26</td>
<td>19.63</td>
<td>25</td>
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<td>0.9889</td>
<td>0.738</td>
</tr>
<tr>
<td>27</td>
<td>21.62</td>
<td>25.3</td>
<td>1.438</td>
<td>0.9897</td>
<td>0.810</td>
</tr>
</tbody>
</table>
The results with a pressure drop up to 500 mHg were taken, and a line fitted to them.

![Plot of Cd vs. pressure difference across the nozzle](data:image/png;base64,iVBORw0KGgoAAAANSUhEUgAA...)

<table>
<thead>
<tr>
<th>Pressure Drop (mmHg)</th>
<th>Cd</th>
<th>Excel suggested best fit</th>
<th>$0.008535\log(P_{nh})+0.9568$</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.107</td>
<td>0.9722</td>
<td>0.988</td>
<td>0.949</td>
<td>2.436</td>
</tr>
<tr>
<td>0.268</td>
<td>0.979</td>
<td>0.989</td>
<td>0.952</td>
<td>2.766</td>
</tr>
<tr>
<td>7.652</td>
<td>0.9827</td>
<td>0.992</td>
<td>0.964</td>
<td>1.868</td>
</tr>
<tr>
<td>18.128</td>
<td>0.986</td>
<td>0.993</td>
<td>0.968</td>
<td>1.872</td>
</tr>
<tr>
<td>36.478</td>
<td>0.9876</td>
<td>0.994</td>
<td>0.970</td>
<td>1.769</td>
</tr>
<tr>
<td>26.374</td>
<td>0.9878</td>
<td>0.994</td>
<td>0.969</td>
<td>1.910</td>
</tr>
<tr>
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<td>0.993</td>
<td>0.966</td>
<td>2.029</td>
</tr>
<tr>
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<td>0.969</td>
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</tr>
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<td>0.970</td>
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</tr>
<tr>
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<td>0.990</td>
<td>0.954</td>
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</tr>
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<td>1.292</td>
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</tr>
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<td>1.525</td>
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<td>0.958</td>
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</tr>
<tr>
<td>0.178</td>
<td>0.976</td>
<td>0.989</td>
<td>0.950</td>
<td>2.623</td>
</tr>
<tr>
<td>22.437</td>
<td>0.988</td>
<td>0.994</td>
<td>0.968</td>
<td>1.991</td>
</tr>
</tbody>
</table>

Average: 2.240273
Appendix C: Calculation of Saturation Pressure

The saturation pressure was calculated from a best fit line through the data provided in the steam tables. A comparison of the actual temperatures and the predicted temperatures is shown below.

The equation was then re-arranged to provide the saturation pressure for a given temperature.
Appendix D: References

1 v:/mathcad/mathpak/swirl.mcd
1 Department of Trade and Industry National Engineering Laboratory - East Kilbride, Glasgow - Certificate of Calibration - September 1973
1 Bosch Automotive Handbook - 3rd Edition
1 BS 1042. Section 1.1 1992 - Measurement of Fluid flow in closed conduits - Pressure differential devices - Specification for square-edged orifice plates, nozzles, and Venturi tubes inserted in circular cross-section conduits running full
1 Thermodynamics - Cengal and Boles
1 Thermodynamic & Transport Properties of Fluids - CFC Rogers & YR Mayhew)
1 Engineering Thermodynamics - Work and Heat Transfer - Rogers and Mayhew
1 A New Method of Investigation of Swirl Ports - SAE 770404 - Götz Tippelmann
1 Derivation of a Standard Valve Lift Curve for the Calculation of the Mean Flow Parameters of Intake and Exhaust Ports - Memo 294/Gen./118
1 Derivation of a Standard Valve Lift Curve for the Calculation of Mean Flow Parameters of Intake and Exhaust Ports - Memo 294/GEN/118
% Input variables
bore = 0.1; % input('Enter the bore of the cylinder : ');
d = 0.07378; % input('Enter the bowl diameter : ');
h = 0.010292; % input('Enter the bowl height : ');
 rpm = 800; % input('Enter the engine speed in rpm : ');
swirlnumber = 1.512; % input('Enter the engine swirl number in rpm : ');
G = 0.0008; % input('Enter the bump clearance : ');
S = 0.127; % input('Enter the stroke length: ');
RI = 0.9; % input('Input value for r/R at max velocity (0.6 - 1) : ');
epsilon = 16.4; % input('Enter the compression ratio: ');
temp = 298; % input('Enter the initial temperature: ');
press = 101.325; % input('Enter the initial pressure: ');

omega_ivc = (swirlnumber * rpm * 2 * pi) / 60;

for velprof = 1:3,
    fname = strcat('c:\', sprintf('%d', velprof));
    spinup(bore, d, h, omega_ivc, G, S, RI, rpm, cr, temp, press, velprof, fname);
end

function [DH] = dhht(alpha, Vmax, dr, Rm, viscosity, density, Zp, R, rb, h, dt, velprof)
    DH = 0;
    ir = 0.000000001; % 'i' is another symbol for the radius of the cylinder, to avoid confusion with 'r'.
    m = 0;
    mt = 0;
    volt = 0;

    % Losses over piston and cylinder heads
    while (ir < Rm) % Region up to Rm (v = ar^2 + br)
        if (velprof == 1)
            a = -(Vmax/(Rm*Rm)); % Profile 1
            b = -2*a*Rm; % Profile 1
        end
        if (velprof == 2)
            a = Vmax/(Rm*Rm); % Profile 2
            b = 0; % Profile 2
        end
        if (velprof == 3)
            a = Vmax/((Rm*Rm)-(2*R*Rm)); % Profile 3
            b = -2*a*R; % Profile 3
        end

    end

    V = (a*ir*ir)+(b*ir); % Derive velocity equation
    if (ir < rb)
        vol1 = 2*pi*dr*(Zp+h);
    else
        vol1 = 2*pi*dr*Zp;
    end

end
volt = vol1 + volt;
m = density * vol1;
mt = mt + m;
Re = density * V * ir / viscosity;
tau = (0.055/2) * density * (V * ir) * (Re * (Re * (-0.2)));
%Calculate Shear Stress
Ts = (4 * tau * pi * (ir * ir)) * dr;
%Calculate Torque
dh = Ts * dt;
DH = DH + dh;
%Sum Losses

%Increment Radius by small value
ir = ir + dr;
end

while(ir < R)
%Region after Rm (v = dr^3 + er^2 + fr)
y = (Vmax / ((y * Rm * Rm) * Rm) - (y * Rm) - (y * Rm) - (3 * y * Rm) + 2);
d = e * y;
f' = -(y * Rm)^2 * 2 * e;
V = (d * ir * ir * ir) + (e * ir * ir) - (f * ir);
%Derive velocity equation
vol2 = 2 * ir * pi * dr * (Zp);
volt = vol2 + volt;
m = density * vol2;
mt = mt + m;
Re = density * V * ir / viscosity;
tau = (0.055/2) * density * (V * ir) * (Re * (Re * (-0.2)));
%Calculate Shear Stress
Ts = (4 * tau * pi * (ir * ir)) * dr;
%Calculate Torque
dh = Ts * dt;
DH = DH + dh;
%Sum Losses
%Increment Radius by small value
ir = ir + dr;
end

%Losses over cylinder walls
Re = density * Vmax * Rm / viscosity;
tau = (0.055/2) * density * (Vmax * ir) * (Re * (Re * (-0.2)));
Ts = (tau * Rm * 2 * pi * Rm * Zp) - (tau * rb * 2 * pi * rb * h);
dh = Ts * dt;
DH = DH + dh;

function [res] = dtime(previous_Zp, Zp, speed)
res = (previous_Zp - Zp) / speed;

function [res] = mean_swirl(Vmax, dr, Rm, R, velprof)
%Declaration of variables
Vtot = 0;
ir = 0.000000001;
%i is another symbol for the radius of the cylinder, to avoid confusion with 'r'.
%It has a very small initial value to avoid the problem of dividing by zero
while(ir < Rm)
%Region up to Rm (v = ar^2 + br)
if(velprof == 1)
a = -(Vmax / (Rm * Rm));
%Profile 1
b = -2 * a * Rm;
%Profile 1
end
if(velprof == 2)
\[ a = \frac{V_{\text{max}}}{R_m R_m}; \quad \text{%Profile 2} \]
\[ b = 0; \quad \text{%Profile 2} \]
\end{cases}

\begin{cases}
\text{if}(\text{velprof} == 3)
\begin{align*}
a &= \frac{V_{\text{max}}}{((R_m R_m)-(2*R_m R_m))}; \quad \text{%Profile 3} \\
b &= -2*a*R; \quad \text{%Profile 3}
\end{align*}
\end{cases}

\begin{align*}
V &= (a*ir*ir)+(b*ir); \quad \text{%Derive velocity equation} \\
V_{\text{tot}} &= V_{\text{tot}}+V; \\
ir &= ir + dr;
\end{align*}
\end{align*}

\begin{align*}
\text{while}(ir < R) \quad \%\text{Region after } R_m (v = dr^3 + er^2 + fr) \\
y &= ((2*R_m)-R))/((R*R)-(3*R_m*R_m)); \\
e &= \frac{V_{\text{max}}}{((y*R_m*R_m*R_m)-(R_m*R_m)-(R_m*R_m*((3*y*R_m)+2))); \\
d &= e*y; \\
f &= -(R_m^2*(3*d*R_m)+(2*e)); \\
V &= (d^*ir*ir*ir)+(e^*ir*ir)+(f^*ir); \quad \text{%Derive Velocity equation} \\
V_{\text{tot}} &= V_{\text{tot}}+V; \\
ir &= ir + dr;
\end{align*}
\end{align*}

\begin{align*}
\text{res} &= \frac{V_{\text{tot}}}{(R/dr)}; \\
\%	ext{Function Piston Speed calculates the velocity of the piston at each crank angle displacement. Time calculates the time step, taking into account this piston speed}
\end{align*}

\begin{align*}
\text{function } \text{[res] = piston\_speed}(\text{rpm, alpha, S, G}) \\
\quad \text{mean\_speed} &= 2*(S+G)*(\text{rpm}/60); \\
R &= 3.504; \\
\text{alpharad} &= \alpha\pi/180; \\
a &= (\pi/2)*\sin(\text{alpharad}); \\
b &= \cos(\text{alpharad}); \\
c &= R*R; \\
d &= \sin(\text{alpharad})*\sin(\text{alpharad}); \\
e &= ((c-d)^.5); \\
\text{res} &= -a*(1+(b/e))*\text{mean\_speed};
\end{align*}

\begin{align*}
\text{function } \text{[res] = pressure}(\text{press, previous\_vol, current\_vol}) \\
\text{res} &= \text{press}*((\text{previous\_vol}/\text{current\_vol})^1.35); \\
\%	ext{Swirl Spin-up in IC Engine}
\text{function } []=\text{spinup}(\text{bore, d, h, omega\_ivc, G, S, R1, rpm, cr, temp, press, velprof, fname});
\%	ext{Main Program}
\text{dr} &= 0.0002; \\
V_{\text{max}} &= 0; \\
dalpha &= 1; \\
\%	ext{Define remaining constants}
\text{R} &= \text{bore}/2; \\
\text{rb} &= \text{d}/2; \\
\text{previous\_Zp} &= \text{S}+\text{G}; \\
\text{omega} &= \omega\text{omega\_ivc}; \\
\text{bowl\_move} &= (\text{R1}-(\text{d}/\text{bore}))/(30/dalpha);
\%	ext{Calculation of initial H and initial volume:}
\end{align*}
density = press/(0.287*temp);
H = ((pi*density/2)*((R*R*R*R*(S+G))+(rb*rb*rb*rb*h)))*omega_ivc;
previous_vol = (pi*R*R*(S+G)+(pi*rb*rb*h));

I = (pi*density/2)*((R*R*R*R*(S+G))+(rb*rb*rb*rb*h));
turb KE = 0.5*I*omega_ivc*omega_ivc*0.3;

% Open log file
fid = fopen(strcat(filename,'_spinup.csv'),'w');

% Display header
Varlist={'G','dr','S','crank angle','omega','H','KE','squish KE','psq','squish velocity','squish mom','V','turb KE','dturb KE','swirl rpm','swirl ratio','mean swirl vel','Vmax','piston speed','time step'};

txt = "";
for n = 1:length(Varlist);
    txt = strcat(txt,strprintf('%15s,',Varlist{n}));
end;

disp(txt);
fprintf(fid,'%s',txt);

% m_bowl_previous=0;
for alpha = 180:360,
    Rm = R1*R;

% Calculate current conditions in the cylinder
    Zp = G + ((S/2)*(1+cos((alpha+180)*pi/180)));
    current_vol = (pi*R*R*Zp)+(pi*rb*rb*h);
    temp = temperature(temp, previous_vol, current_vol);
    viscosity = (0.00000152*temp.^1.5)/(124.856+temp);
    press = pressure(press, previous_vol, current_vol);
    density = press/(0.287*temp);
    mass = density*current_vol;

% Calculate piston speed hence timestep
    speed = piston_speed(rpm, alpha, S, G);
    if (alpha<(361-da))
        dt = dtime(previous_Zp, Zp, speed);
    end

% Calculate squish conditions
    m_bowl_current = pi*rb*rb*(h+Zp)*density;
    m_cylinder = pi*((R*R)-(rb*rb))*Zp*density;
    m_total = m_cylinder+m_bowl_current;

    dm_bowl = m_bowl_current - m_bowl_previous;
    squish_velocity = squish(Zp, h, d, bore, speed);
    squish_momentum = squish_velocity*dm_bowl;
    squish KE = 0.5*dm_bowl*squish_velocity*squish_velocity ;
    H = H + (0.03*squish_momentum);  // Squish momentum addition
    I = (pi*density/2)*((R*R*R*R*(Zp))+(rb*rb*rb*rb*h));

% Calculate Vmax and momentum losses, followed by new H
    Vmax = vel_iterator_parab(rb, Vmax, Rm, R, density, Zp, h, dr, H, velprof);
    dh = dhh(alpha, Vmax, dr, Rm, viscosity, density, Zp, R, rb, h, dt, velprof);
    H = H - dh;

% Calculate equivalent solid-body angular velocity
omega = H/I;

%Calculate turbulence conditions
KE = 0.5*omega*omega;
dturb KE = turbulence(turb KE, cr, rpm, S, bore, squish velocity, dt);
%KE = KE - 0.3*dturb KE;               //Kinetic Energy Losses
psq = squish turbulence(squish velocity, dt);
turb KE = turb KE + dturb KE + (psq*mass);

%Angular Velocity after turbulent losses
omega = sqrt(2*KE/I);
H = 2*KE/omega;
%disp(sprintf("b omega : %f",omega));

%Calculate further parameters for analysis
swirl rpm = (omega/(2*pi))*60;
swirl ratio = swirl rpm/rpm;
mean swirl vel = mean swirl(Vmax, dr, Rm, R, velprof);

%Output Results
txt = sprintf('%15g','G,dr,Zp,alpha,omega,H,KE,squish KE,psq,squish_velocity,squish_momentum,curr
ent_vol,turb KE,dturb KE,swirl rpm,swirl_ratio,mean_swirl vel,Vmax,speed,dt);
disp(txt); fprintf(fid,%s',txt);

%Increment Cylinder Conditions
previous_vol = current_vol;
previous Zp = Zp;
m_bowl previous = m_bowl current;
alpha = alpha + dalpha;

%Move location of Vmax
if(alpha > 330)
    RI = R1-bowl_move;
end
alpha = alpha-dalpha;
vel_prof(Rm, Vmax, alpha, R, dr, velprof,fname);
fclose(fid);
disp('Simulation complete.);

%Function squish calculates and returns the squish velocity, based on the
cyllinder dimensions and piston speed. Turbulence calculates the change
%in the turbulent and kinetic energy due to production and dissipation, as
%well as squish effects.

function res=squish(Zp,h,d,bore,speed)
    vol_bowl = pi*d*d*h/4;
    Ac = pi*bore*bore/4;
    a = d/(4*Zp);
    b = ((bore*bore)/(d*d))-1;
    c = vol_bowl/((Ac*Zp)+vol_bowl);
    res = a*b*c*speed;
function [psq] = squish_turbulence(squish_velocity,dt)
    psq = 0.01*squish_velocity*squish_velocity*dt; %Squish turbulence

%Calculates the temperature and pressure in the cylinder using the
%standard polytropic relationships with k = 1.35
function [res] = temperature(temp,previous_vol,current_vol)
    res = temp*((previous_vol/current_vol).^0.35);

function [dturb KE] = turbulence(turb KE,cr,rpm,S,bore,squish_velocity,dt)

    if (S<=bore)
        beta = 1;
    else
        beta = bore/S;
    end
    dturb KE = (((rpm/45)*turb KE*log(cr))- 
                (120*turb KE*turb KE/(S*S*beta*beta*rpm)))*dt;

%Function vel_iterator_parab calculates the angular momentum of the flow by integrating
%the velocity curve, and compares to angular momentum of the actual flow. The maximum
%velocity is changed until the two angular momentums are equal (within 99.99%) and
%returns the value of Vmax.
function [Vmax] = vel_iterator_parab(rb,Vmax,Rm,R,density,Zp,h,dr,H,velprof)

    %Declaration of variables
    H2 = 0; %Angular momentum of flow with parabolic velocity profile
    Vmax = Vmax + 0.01; %Increment maximum velocity
    r = 0.00000001;
    v = 0;
    ms = 0;
    while(r<Rm) %Region up to Rm (v = ar^2 + br)
        if(velprof == 1)
            a = -(Vmax/(Rm*Rm)); %Profile 1
            b = -2*a*Rm; %Profile 1
        end
        if (velprof == 2)
            a = Vmax/(Rm*Rm); %Profile 2
            b = 0; %Profile 2
        end
        if(velprof == 3)

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\[ a = \frac{\text{Vmax}}{(Rm \times Rm) - (2 \times R \times Rm)}; \quad \text{Profile 3} \]
\[ b = -2a \times R; \quad \text{Profile 3} \]

velocity = \( a \times r + b \times r \); \quad \text{%Derive velocity equation}
\[
\omega_2 = \frac{\text{velocity}}{r};
\]
if \( r < rb \)
\[
\text{volume} = 2 \times \pi \times r \times dr \times (Zp + h);
\]
else
\[
\text{volume} = 2 \times \pi \times r \times dr \times Zp;
\]
end

mass = \text{volume} \times \text{density};
\[
ms = ms + mass;
\]
I = mass \times (r + dr) \times (r + dr);
\[
dH2 = l \times \omega_2;
\]
%Add H of individual rings to find total H
\[
H2 = H2 + dH2;
\]
\[
\nu = \nu + \text{volume};
\]
\[
r = r + dr; \quad \text{%Increment radius}
\]
end
end

%Region after Rm (\( \nu = dr^3 + er^2 + fr \))
\[
y = \frac{(2 \times Rm - R)}{((R \times R) - (3 \times Rm \times Rm))};
\]
\[
e = \frac{\text{Vmax} \times ((y \times Rm \times Rm) + (Rm \times Rm) - (Rm \times Rm \times ((3 \times y \times Rm) + 2)))}{(R \times R) - (3 \times Rm \times Rm)};
\]
\[
d = e \times y;
\]
\[
f = -Rm \times ((3 \times d \times Rm) + 2 \times e);
\]
\[
\text{velocity} = \frac{(d \times r + e \times r) + (e \times r + f \times r)}{r}; \quad \text{%Derive velocity equation}
\]
\[
\omega_2 = \frac{\text{velocity}}{r};
\]
\[
\text{volume} = 2 \times \pi \times r \times dr \times Zp;
\]
\[
\text{mass} = \text{volume} \times \text{density};
\]
\[
ms = ms + mass;
\]
\[
I = mass \times (r + dr) \times (r + dr);
\]
\[
dH2 = l \times \omega_2;
\]
%Add H of individual rings to find total H
\[
H2 = H2 + dH2;
\]
\[
\nu = \nu + \text{volume};
\]
\[
r = r + dr; \quad \text{%Increment radius}
\]
end

while \( r \leq R \) \quad \text{%Region after Rm (\( \nu = dr^3 + er^2 + fr \))}
\]
\[
y = \frac{(2 \times Rm - R)}{((R \times R) - (3 \times Rm \times Rm))};
\]
\[
e = \frac{\text{Vmax} \times ((y \times Rm \times Rm) + (Rm \times Rm) - (Rm \times Rm \times ((3 \times y \times Rm) + 2)))}{(R \times R) - (3 \times Rm \times Rm)};
\]
\[
d = e \times y;
\]
\[
f = -Rm \times ((3 \times d \times Rm) + 2 \times e);
\]
\[
\text{velocity} = \frac{(d \times r + e \times r) + (e \times r + f \times r)}{r}; \quad \text{%Derive velocity equation}
\]
\[
\omega_2 = \frac{\text{velocity}}{r};
\]
\[
\text{volume} = 2 \times \pi \times r \times dr \times Zp;
\]
\[
\text{mass} = \text{volume} \times \text{density};
\]
\[
ms = ms + mass;
\]
\[
I = mass \times (r + dr) \times (r + dr);
\]
\[
dH2 = l \times \omega_2;
\]
%Add H of individual rings to find total H
\[
H2 = H2 + dH2;
\]
\[
\nu = \nu + \text{volume};
\]
\[
r = r + dr; \quad \text{%Increment radius}
\]
end
end

%Return vmax

%Velocity profile 1 is a negative parabolic with a local maximum at \( r = Rm \)
%Velocity profile 2 is a positive parabolic
%Velocity profile 3 is a negative parabolic with a local maximum at \( r = R \)

function \( J = \text{vel PROF}(Rm, \text{Vmax}, \alpha, \text{R}, \text{dr}, \text{velprof}, \text{fname}) \)

\[
fid = \	ext{fopen}('\text{strcat(fname,'}_\text{velprofexc}.\text{csv}{', 'w');}
\]
\[
\text{fprintf}(\text{fid},'\text{Crank_Angle, radius, velocity', 'r});
\]
\[
\text{%Declaration of variables}
\]
\[
r = 0;
\]

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while(r<Rm) %/Region up to Rm (v = ar^2 +br)
    if(velprof == 1)
        a = -Vmax/(Rm*Rm));  %Profile 1
        b = -2*a*Rm;          %Profile 1
    end
    if (velprof == 2)
        a = Vmax/(Rm*Rm);    %Profile 2
        b = 0;               %Profile 2
    end
    if (velprof == 3)
        a = Vmax/((Rm*Rm)-(2*R*Rm)); %Profile 3
        b = -2*a*R;          %Profile 3
    end
    velocity = (a*r*r)+(b*r); %Derive velocity equation
    fprintf(fid,'%f,%f,%f,%f
',alpha,r,velocity);
    r = r + dr; %increment radius
end

while (r>R) %Region after Rm (v = dr^3 + er^2 + fr)
    y = ((2*Rm)-R)/(y^Rm*Rm^Rm));
    e = Vmax/((y*Rm*Rm*Rm)+(Rm*Rm)-(Rm*Rm*((3*y*Rm)+2)));
    d = e*y;
    f = -Rm*(((3*d*Rm)+(2*e)));
    velocity = (d*r*r*r)+e*r*r)+(f*r); %Derive velocity equation
    fprintf(fid,'%f,%f,%f,%f
',alpha,r,velocity);
    r = r + dr; %Increment radius
end
fclose(fid);
M3 Tapped holes to depth 11mm, equispaced on 91.7mm PCD to centreline of pillars.

3 Holes, 4.3mm countersunk to 9mm, equispaced on 75.86mm PCD

<table>
<thead>
<tr>
<th>BJS008</th>
<th>Scale 11</th>
<th>Mat. Alu. BST490 LM 13 TE</th>
</tr>
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<tbody>
<tr>
<td>4 Pillar Deep Piston Bowl</td>
<td>Drawn By: Brian Stapleton</td>
<td>28/08/2002</td>
</tr>
<tr>
<td></td>
<td>Checked By: Colin Garner</td>
<td></td>
</tr>
</tbody>
</table>
Deep Piston Bowl Gloss

Notes:
1. All edges to be chamfered
2. Stress relieving if possible
3. All notches same radius
   equispaced on 98.3mm PCD
4. Optical polished finish

<table>
<thead>
<tr>
<th>LBO-CAT-G2A</th>
<th>27 Jan. 2003</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deep Piston Bowl Glass</td>
<td></td>
</tr>
<tr>
<td>Drawn by: Brian Stapleton</td>
<td></td>
</tr>
<tr>
<td>Checked by: Colin Garner</td>
<td></td>
</tr>
</tbody>
</table>
3.1mm countersunk through hole on PCD 91.7mm, equilaterally spaced

36.1mm Diameter, recess
15mm, 45 degree chamfer, perpendicular spaced on PCD
63.64mm

BJ5007  Scale 11  Material: Titanium
4 Pillar Deep Bowl Piston Lid

<table>
<thead>
<tr>
<th></th>
<th>Drawn By</th>
<th></th>
<th></th>
</tr>
</thead>
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<tr>
<td></td>
<td>Brian Stapleton</td>
<td>28/08/2002</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Checked By</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Colin Garner</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Appendix 3

Shallow piston bowl

Piston cassette drawings (deep and shallow bowl)

M3 Tapped holes to depth 11mm on 92.3mm PCD, placed on centreline of pillars.

3 Holes, 4.3mm countersunk to 9mm, equispaced on 85mm PCD.

120° ±0.05

Pillars R6.0 on a 98.3mm PCD

<table>
<thead>
<tr>
<th>BJS010</th>
<th>Scale 11</th>
<th>Material: Aluminium</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 Pillar Shallow Pistons Bowl</td>
<td>Drawn By: Brian Stapleton 28/08/2002</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Checked By: Colin Garner</td>
<td></td>
</tr>
</tbody>
</table>
Notes:
1. All edges to be chamfered
2. Stress relieving if possible.
3. All notches same radius on 98.3mm PCD
4. Optical polished finish

Shallow Piston Bowl Glass

LBD-CAT-BRIAN 27 Jan 2003

Drawn by: Brian Stapleton
Checked by: Colin Garner
Appendix 3

Piston cassette drawings (deep and shallow bowl)

Shallow bowl piston top

<table>
<thead>
<tr>
<th>BJS011</th>
<th>Scale 11</th>
<th>Material: Titanium</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 Pillar Shallow Bowl Piston Lid</td>
<td>Drawn By: Brian Stapleton</td>
<td>28/08/2002</td>
</tr>
<tr>
<td></td>
<td>Checked By: Colin Garner</td>
<td></td>
</tr>
</tbody>
</table>
## Appendix A4  Engine upfit procedure

### Piston Cassette Upfit

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Turn Engine to TDC, using crankshaft encoder indication (green and red light on), use 16mm socket on balancing shaft to turn engine.</td>
</tr>
<tr>
<td>ii</td>
<td>Fit appropriate Piston Bowl to Piston, align markings and tighten countersunk Allan Bolts (x 3).</td>
</tr>
<tr>
<td>iii</td>
<td>Clean glass prior to fitting using a clean lint free cloth. Fit Piston Annulus Glass to Piston bowl, a Gaskoid 0.15mm gasket to be fitted either side of glass to remove metal to glass contact. Ensure glass is centralised between piston bowl pillars (using piston ring clamp if necessary). Note, the shallow bowl glass only fits one way, and the piston crown has one valve recess missing which fits where the optical window in the cylinder head is positioned.</td>
</tr>
<tr>
<td>iv</td>
<td>Fit appropriate Piston annulus lid to piston bowl. Tighten lid to piston bowl by Allan bolts (x 4).</td>
</tr>
</tbody>
</table>

### Fitting Cylinder Glass

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Fit bottom Cylinder glass gasket (rubber type joint) to the top of the cylinder, ensuring no creases in gasket.</td>
</tr>
<tr>
<td>ii</td>
<td>Clean cylinder glass thoroughly with a clean lint free cloth.</td>
</tr>
<tr>
<td>iii</td>
<td>Gently place cylinder glass onto cylinder top (recess) ensuring it does not enter at an angle.</td>
</tr>
</tbody>
</table>

### Fitting Cylinder Head

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Ensure that the cylinder glass is in place and is in a clean state.</td>
</tr>
<tr>
<td>ii</td>
<td>Ensure that steel cylinder head spacers are in place in their correct positions (as marked) and not hard against the cylinder glass.</td>
</tr>
<tr>
<td>iii</td>
<td>Clean cylinder glass window (top and bottom) with a clean lint free cloth. Inset a clean cloth into void space between window and opening at top of cylinder head for protection of glass during upfit.</td>
</tr>
<tr>
<td>iv</td>
<td>Fit graphite gasket to the top of the cylinder glass ensuring it is centralised.</td>
</tr>
<tr>
<td>v</td>
<td>Place cylinder, with running gear facing the front of the engine, onto the spacers until it is correctly seated. Fit cylinder head bolts, (3 * ½” UNF x 6½” bolts with washers and 1 * ½” UNF x 3” Allen bolt)</td>
</tr>
<tr>
<td>vi</td>
<td>Tighten cylinder head bolts equally, tightening to a torque of 100N.m</td>
</tr>
<tr>
<td>vii</td>
<td></td>
</tr>
</tbody>
</table>

### Fitting of Cam Carrier

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Remove front and rear thrust bearings.</td>
</tr>
<tr>
<td>ii</td>
<td>Remove camshafts and clean surfaces with a clean cloth. Clean roller bearings with cloth and ensure no swarf or dirt are left on bearing surfaces.</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>iii</td>
<td>Re-grease camshaft surfaces and roller bearings with high temp grease.</td>
</tr>
<tr>
<td>iv</td>
<td>Fit thrust bearings back into correct positions (note that front thrust bearings are to be fitted with spacer to the front.</td>
</tr>
<tr>
<td>v</td>
<td>Ensure that tappet props are in correct positions (3 off).</td>
</tr>
<tr>
<td>vi</td>
<td>Lightly grease the top of the tappet props and the valve stems with high temp grease.</td>
</tr>
<tr>
<td>vii</td>
<td>Place cam carrier onto cylinder head, aligning with dowels in cam carrier.</td>
</tr>
<tr>
<td>viii</td>
<td>Fit timing plate to cam carrier (bolting into place) setting the camshafts to the TDC position. Wedge the sides of the cam carrier up to be able to fit finger followers to be fitted to the cam carrier. Ensure no pre-loading from camshaft onto the valves. If there is too much play, shim the tappet props from access point.</td>
</tr>
<tr>
<td>ix</td>
<td>Remove all wedges, ensuring appropriate finger followers are securely in place with valve stems and tappet props. Check for pre-loading of valves.</td>
</tr>
<tr>
<td>x</td>
<td>Bolt down cam carrier by 4 x corner allen bolts. Tighten to hand tight, unnecessary to torque.</td>
</tr>
<tr>
<td>xi</td>
<td>Fit camshafts gears onto the end of the camshafts loosely, fit spacer and torque bolt. Note there is no location for camshaft gear. Ensure that gear wheels can still turn.</td>
</tr>
<tr>
<td>xii</td>
<td>Loosely fit the cambelt tensioner to the cylinder head, ensuring the idler gear is lubricated and free running.</td>
</tr>
<tr>
<td>xiii</td>
<td>Fit cambelt around tensioners and camshaft gears, using a lever, tighten the belt to remove any slack and bolt the cambelt tensioner into place. Ensure during this process that crankshaft is not rotated.</td>
</tr>
<tr>
<td>xiv</td>
<td>Once the belt is tensioned sufficiently (should be able to turn the belt approx. 90 degrees). Torque the camshaft gear bolts to 140 N.m, beware of timing plate crushing/buckling.</td>
</tr>
<tr>
<td>xv</td>
<td>Remove the timing plate (may need mallet and screwdriver to remove) and ensure there is nothing fouling the cambelt.</td>
</tr>
<tr>
<td></td>
<td><strong>Turn engine over by hand.</strong></td>
</tr>
<tr>
<td>i</td>
<td>Place 16mm socket onto crankshaft balance shaft. Turn engine over slowly by hand. Using a torch, look at the valve clearances through the cylinder glass.</td>
</tr>
<tr>
<td>ii</td>
<td>Ensure a full 720 degrees of the engine is turned to ensure no collision of running gear.</td>
</tr>
<tr>
<td>iii</td>
<td></td>
</tr>
</tbody>
</table>
Running under motored conditions

<table>
<thead>
<tr>
<th>Test</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Start oil pump, (switch located on rear of engine housing) and ensure oil pressure at gauge in block, if necessary, lower/raise oil pressure by adjusting relief valve located under rear of the housing. exhaust manifold and inlet manifold are to be fitted to the engine. The exhaust extractor fan is to be powered up to remove the seeded air from the engine. Ensure crankshaft and camshaft encoder are fitted to the engine and timing box and that leads are free and clear. Power up the timing box and observe the engine speed. Power up the motor control unit, ensure that the speed control dial is set zero (all the way anti-clockwise). Reset the emergency stop and reset the motor. Ensure all panels and doors are properly shut and locked in place. Turn the speed dial and wind the engine up to 100rpm, listening carefully for any unusual noises. In the event of unusual noises, hit the emergency stop and wait for the engine to stop leaving the extractor fan on in the event of glass breaking.</td>
</tr>
<tr>
<td>ii</td>
<td>Ensure crankshaft and camshaft encoder are fitted to the engine and timing box and that leads are free and clear. Power up the timing box and observe the engine speed.</td>
</tr>
<tr>
<td>iii</td>
<td>Power up the motor control unit, ensure that the speed control dial is set zero (all the way anti-clockwise). Reset the emergency stop and reset the motor.</td>
</tr>
<tr>
<td>iv</td>
<td>Ensure all panels and doors are properly shut and locked in place.</td>
</tr>
<tr>
<td>v</td>
<td>Turn the speed dial and wind the engine up to 100rpm, listening carefully for any unusual noises. In the event of unusual noises, hit the emergency stop and wait for the engine to stop leaving the extractor fan on in the event of glass breaking.</td>
</tr>
</tbody>
</table>

Run with Cylinder Glass in place

<table>
<thead>
<tr>
<th>Test</th>
<th>Description</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Assemble engine without piston cassette, include the cylinder glass (ensuring gaskets are correctly placed top and bottom). Care to exercised when torquing down cylinder head to not exert too much pressure on glass.</td>
<td>To check the engine for low engine speed and leakage from the cylinder through the gaskets.</td>
</tr>
<tr>
<td>ii</td>
<td>Follow steps ii to xii from Initial Run with running valve gear.</td>
<td></td>
</tr>
<tr>
<td>iii</td>
<td>Apply leak detection fluid (&quot;Snoop&quot;) about top and bottom cylinder glass gaskets.</td>
<td></td>
</tr>
<tr>
<td>iv</td>
<td>Run engine to 100 rpm and inspect through door of engine housing for evidence of leakage from cylinder.</td>
<td></td>
</tr>
<tr>
<td>v</td>
<td>Increase speed of engine to 500 rpm and hold for 1 min.</td>
<td></td>
</tr>
<tr>
<td>vi</td>
<td>Monitor camshaft-bearing temps.</td>
<td></td>
</tr>
<tr>
<td>vii</td>
<td>Wind down engine to stop and inspect for leakage.</td>
<td></td>
</tr>
</tbody>
</table>
### Run with Cylinder Glass and Piston Cassette (with optic)

<table>
<thead>
<tr>
<th>Test</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Assemble engine with piston cassette, piston glass, piston lid, and cylinder glass (ensuring gaskets are correctly placed top and bottom). Care to be exercised when torqueing down cylinder head to not exert too much pressure on cylinder glass. To run the engine in it’s final configuration, to check for leakage at high engine speeds, ensure the optics are not scuffing and the engine is configured correctly for future PIV testing.</td>
</tr>
<tr>
<td>ii</td>
<td>Follow steps ii to xiii from Initial Run with running valve gear.</td>
</tr>
<tr>
<td>iii</td>
<td>Apply leak detection fluid (‘Snoop’) about top and bottom cylinder glass gaskets.</td>
</tr>
<tr>
<td>iv</td>
<td>Run engine to 100 rpm and inspect through door of engine housing for evidence of leakage from cylinder.</td>
</tr>
<tr>
<td>v</td>
<td>Increase speed of engine to 500 rpm and hold for 1 min.</td>
</tr>
<tr>
<td>vi</td>
<td>Monitor camshaft-bearing temps.</td>
</tr>
<tr>
<td>vii</td>
<td>Increase engine speed to 1000 rpm and hold for 1 min.</td>
</tr>
<tr>
<td>viii</td>
<td>Increase engine speed to 1500 rpm and hold for 1 min.</td>
</tr>
<tr>
<td>ix</td>
<td>Increase engine speed to 2000 rpm and hold for 1 min.</td>
</tr>
<tr>
<td>x</td>
<td>Increase engine speed to 2300 rpm and hold for 1 min.</td>
</tr>
<tr>
<td>xi</td>
<td>Continual monitoring of camshaft bearing temps through engine speed runs.</td>
</tr>
<tr>
<td>xii</td>
<td>Wind engine down to a stop and check further for evidence of any leakage through cylinder glass gaskets.</td>
</tr>
</tbody>
</table>

### Fitting of Camera

<table>
<thead>
<tr>
<th>Test</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Camera to be mounted to slide mounted to beam on engine housing with associated post and arm arrangement of camera. To focus and take a reference image inside the bowl.</td>
</tr>
<tr>
<td>ii</td>
<td>Camera to be connected to computer for live image.</td>
</tr>
<tr>
<td>iii</td>
<td>Fit the piston bowl female with reference graph paper into the piston bowl. Cylinder glass does not need to be fitted in this arrangement.</td>
</tr>
<tr>
<td>iv</td>
<td>Illuminate piston bowl area with a lamp to show focus plane for camera and take a reference image.</td>
</tr>
<tr>
<td>v</td>
<td>Mark position of camera on the slide and slide camera out of the way to allow upfit of engine.</td>
</tr>
<tr>
<td>vi</td>
<td>Carry out upfit of engine as per instructions.</td>
</tr>
<tr>
<td>vii</td>
<td>Move camera back to mark on slide and lock into place using bolt on side of slide to lock position.</td>
</tr>
</tbody>
</table>
### Setting of Laser

<table>
<thead>
<tr>
<th>Test</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Laser to be mounted inside engine housing across the back table. Mirrors to be used to beam laser into piston bowl. To position laser beam to correct plane.</td>
</tr>
<tr>
<td>ii</td>
<td>Beam stops to be mounted into bowl to show position of laser inside the engine bowl.</td>
</tr>
<tr>
<td>iii</td>
<td>Alter the position of mirrors until correct alignment of beam is achieved inside the piston bowl.</td>
</tr>
</tbody>
</table>

### Post Test Inspection

<table>
<thead>
<tr>
<th>Test</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>Dismantle cam carrier and camshafts and inspect bearings for wear or heat damage during engine running. To inspect all new parts of the engine for any signs of wear or damage from testing.</td>
</tr>
<tr>
<td>ii</td>
<td>Inspect the finger followers and valve stems for any damage during running.</td>
</tr>
<tr>
<td>iii</td>
<td>Check cylinder glass and piston glass for any signs of cracking or scuffing damage.</td>
</tr>
<tr>
<td>iv</td>
<td>Check bolts for piston lid and piston cassette for any noticeable signs of stretching or damage.</td>
</tr>
</tbody>
</table>
Figure A5.1  Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, combined inlet, 800 rpm crankshaft speed

Figure A5.2  Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, combined inlet, 800 rpm crankshaft speed
Figure A5.3  Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, directed inlet, 800 rpm crankshaft speed

Figure A5.4  Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, directed inlet, 800 rpm crankshaft speed
Figure A5.5 Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, helical inlet, 800 rpm crankshaft speed

Figure A5.6 Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, helical inlet, 800 rpm crankshaft speed
Figure A5.7  Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, combined inlet, 1200 rpm crankshaft speed

Figure A5.8  Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, combined inlet, 1200 rpm crankshaft speed
Figure A5.9 Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, directed inlet, 1200 rpm crankshaft speed

Figure A5.10 Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, directed inlet, 1200 rpm crankshaft speed
Figure A5.11 Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, helical inlet, 1200 rpm crankshaft speed

Figure A5.12 Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, helical inlet, 1200 rpm crankshaft speed
Figure A5.13 Ensemble average vector map (m.s⁻¹) - deep bowl piston, combined inlet, 1600 rpm crankshaft speed

Figure A5.14 Instantaneous vector map (m.s⁻¹) - deep bowl piston, combined inlet, 1600 rpm crankshaft speed
Figure A5.15 Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, directed inlet, 1600 rpm crankshaft speed

Figure A5.16 Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, directed inlet, 1600 rpm crankshaft speed
Figure A5.17 Ensemble average vector map (m.s\(^{-1}\)) - deep bowl piston, helical inlet, 1600 rpm crankshaft speed

Figure A5.18 Instantaneous vector map (m.s\(^{-1}\)) - deep bowl piston, helical inlet, 1600 rpm crankshaft speed
Figure A5.19 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, combined inlet, 800 rpm crankshaft speed

Figure A5.20 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, combined inlet, 800 rpm crankshaft speed
Figure A5.21 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, directed inlet, 800 rpm crankshaft speed

Figure A5.22 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, directed inlet, 800 rpm crankshaft speed
Figure A5.23 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, helical inlet, 800 rpm crankshaft speed

Figure A5.24 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, helical inlet, 800 rpm crankshaft speed
Figure A5.25 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, combined inlet, 1200 rpm crankshaft speed

Figure A5.26 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, combined inlet, 1200 rpm crankshaft speed
Figure A5.27 Ensemble average vector map (m.s$^{-1}$) - shallow bowl piston, directed inlet, 1200 rpm crankshaft speed

Figure A5.28 Instantaneous vector map (m.s$^{-1}$) - shallow bowl piston, directed inlet, 1200 rpm crankshaft speed
Figure A5.29 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, helical inlet, 1200 rpm crankshaft speed

Figure A5.30 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, helical inlet, 1200 rpm crankshaft speed
Figure A5.31 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, combined inlet, 1600 rpm crankshaft speed

Figure A5.32 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, combined inlet, 1600 rpm crankshaft speed
Figure A5.33 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, directed inlet, 1600 rpm crankshaft speed

Figure A5.34 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, directed inlet, 1600 rpm crankshaft speed
Figure A5.35 Ensemble average vector map (m.s\(^{-1}\)) - shallow bowl piston, helical inlet, 1600 rpm crankshaft speed

Figure A5.36 Instantaneous vector map (m.s\(^{-1}\)) - shallow bowl piston, helical inlet, 1600 rpm crankshaft speed
Appendix A6  Matlab program for interpreting vector files

% Author Tim Justham, Loughborough University, July 2005

function varargout = GUISwirlProgAll180705(varargin)

% Begin initialization code - DO NOT EDIT
gui_Singleton = 1;
gui_State = struct('gui_Name', mfilename, ... 
    'gui_Singleton', gui_Singleton, ... 
    'gui_OpeningFcn', @GUISwirlProgAll180705_OpeningFcn, ... 
    'gui_OutputFcn', @GUISwirlProgAll180705_OutputFcn, ... 
    'gui_LayoutFcn', [], ... 
    'gui_Callback', []);
if nargin && ischar(varargin{1})
    gui_State.gui_Callback = str2func(varargin{1});
end

if nargin
    [varargout{1:nargin}] = gui_mainfcn(gui_State, varargin{:});
else
    gui_mainfcn(gui_State, varargin{:});
end

% End initialization code - DO NOT EDIT

% --- Executes just before GUISwirlProgAll180705 is made visible.
function GUISwirlProgAll180705_OpeningFcn(hObject, eventdata, handles, varargin)

% Choose default command line output for GUISwirlProgAll180705
handles.output = hObject;

% Update handles structure
guidata(hObject, handles);

% --- Outputs from this function are returned to the command line.
function varargout = GUISwirlProgAll180705_OutputFcn(hObject, eventdata, handles)
% varargout cell array for returning output args (see VARARGOUT);
% hObject handle to figure
% eventdata reserved - to be defined in a future version of MATLAB
% handles structure with handles and user data (see GUIDATA)

% Get default command line output from handles structure
varargout{1} = handles.output;

function NoDP_Callback(hObject, eventdata, handles)

% --- Executes during object creation, after setting all properties.
function NoDP_CreateFcn(hObject, eventdata, handles)

if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor',get(0,'defaultUicontrolBackgroundColor'));
end

function EngSpeed_Callback(hObject, eventdata, handles)

% --- Executes during object creation, after setting all properties.
function EngSpeed_CreateFcn(hObject, eventdata, handles)
if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor'.get(0,'defaultUiControlBackgroundColor'));
end

function FileName_Callback(hObject, eventdata, handles)
% --- Executes during object creation, after setting all properties.
function FileName_CreateFcn(hObject, eventdata, handles)
if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor'.get(0,'defaultUiControlBackgroundColor'));
end

function StartCount_Callback(hObject, eventdata, handles)
% --- Executes during object creation, after setting all properties.
function StartCount_CreateFcn(hObject, eventdata, handles)
if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor'.get(0,'defaultUiControlBackgroundColor'));
end

function EndCount_Callback(hObject, eventdata, handles)

function EndCount_CreateFcn(hObject, eventdata, handles)
if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor'.get(0,'defaultUiControlBackgroundColor'));
end

function ULoc_Callback(hObject, eventdata, handles)
% --- Executes during object creation, after setting all properties.
function ULoc_CreateFcn(hObject, eventdata, handles)
if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor'.get(0,'defaultUiControlBackgroundColor'));
end

function VLoc_Callback(hObject, eventdata, handles)
% --- Executes during object creation, after setting all properties.
function VLoc_CreateFcn(hObject, eventdata, handles)
if ispc
    set(hObject,'BackgroundColor','white');
else
    set(hObject,'BackgroundColor'.get(0,'defaultUiControlBackgroundColor'));
end
% --- Executes on button press in RunButton.
function RunButton_Callback(hObject, eventdata, handles)

MainSwirl(hObject, eventdata, handles);
function MainSwirl(hObject, eventdata, handles)

opengl neverselect %never select opengl driver and hence default to 1 - will stop warnings when plotting graphs

%Data Entry Section, Definition of variables etc.
cd (get(handles.Path,str)); %Enter path to vec files
StartFileName=get(handles.FileName,str); %Start/Initial Part of the File Name without identifying numbers
Ext='.TOOO.DOOO.POO0.HOOO.L.vec'; %File Extension
FileNameCount=str2double(get(handles.StartCount,str)); %First Identifying Number (file number) no preceding zeros
EndFileNameCount=str2double(get(handles.EndCount,str)); %End identifying number - must be continuous run from start number (no breaks)
u_location=str2double(get(handles.ULoc,str)); %Specify location of the swirl centre
v_location=str2double(get(handles.YLoc,str)); %Specify location of the swirl centre
Engine_Speed=str2double(get(handles.EngSpeed,str)); %Engine speed for normalization
int_region_mass=1; %Mass of interrogation region for Inertia Calculations
number_of_dp=str2double(get(handles.NoDP,str)); %this controls the number of points that are plotted on the profile plots (it does not have to be an integer)

%----------------------------------------------------------
%Beginning of program to extract vector file information into different matrices
%Converts to U velocity and V velocity matrices (can make one complex matrix easier to handle
%sometimes in the form U+iV)

Count=0;
while FileNameCount<=EndFileNameCount;
    FileNameCountStr=sprintf('%06. or',FileNameCount);
    FileNameCount=FileNameCount+1;
    Count=Count+1;
    FileName=strcat(StartFileName,FileNameCountStr,Ext);
    fid=fopen(FileName,'r'); %Open File
    fgetl(fid); %Discard First Row of Headers
    TempVecFile=(fscanf(fid,'%f%*s %f%*s %f%*s %f%*s %f\n',[5,inf]))'; %Read vec file row at a time into a 5 x N matrix
    fclose(fid);
    CountX=1; %Preset Count as =1
    while TempVecFile(1+CountX,1)>TempVecFile(1,1); %Count number of X positions in the array
        CountX=CountX+1;
    end

299
if Count<=1
    xValues=reshape((TempVecFile(:,1)),CountX,[]);
    yValues=reshape((TempVecFile(:,2)),CountX,[]);
    uValues=reshape((TempVecFile(:,3)),CountX,[]);
    vValues=reshape((TempVecFile(:,4)),CountX,[]);
else
    xValues=cat(3,xValues,(reshape((TempVecFile(:,1)),CountX,[])));
    yValues=cat(3,yValues,(reshape((TempVecFile(:,2)),CountX,[])));
    uValues=cat(3,uValues,(reshape((TempVecFile(:,3)),CountX,[])));
    vValues=cat(3,vValues,(reshape((TempVecFile(:,4)),CountX,[])));
end

clear TempVecFile;

TempLocation=find(uValues>=9000000000);
%Sets spurious/missing data locations to nan
uValues(TempLocation)=nan; %As nan they take no part in the averages etc. when nanmean is used
TempLocation=find(vValues>=9000000000);
vValues(TempLocation)=nan;
end

countz=1;
while countz<=size(xValues,3);
    x=xValues(:,:,countz);
    y=yValues(:,:,countz);
    u=uValues(:,:,countz);
    v=vValues(:,:,countz);
    count=1;
    while count<=size(xValues,1);
        xValues(((size(xValues,1)))+1-count,:,countz)=x(count,:);
        yValues(((size(yValues,1)))+1-count,:,countz)=y(count,:);
        uValues(((size(uValues,1)))+1-count,:,countz)=u(count,:);
        vValues(((size(vValues,1)))+1-count,:,countz)=v(count,:);
        count=count+1;
    end
    countz=countz+1;
end
clear x; clear y; clear u; clear v;

%Basic Calculations

MagValues=sqrt((uValues.^2)+(vValues.^2)); %Velocity magnitude

Data_Amount_U=sum(abs((isnan(uValues))-1),3)/size(uValues,3)*100;
%Number of good data points in percent
Data_Amount_V=sum(abs((isnan(vValues))-1),3);
Data_Amount_Mag=sum(abs((isnan(MagValues))-1),3);

%Calculating Mean Values

CountZ=1;
uValues_DIM321=permute(uValues,[3,2,1]);
vValues_DIM321=permute(vValues,[3,2,1]);
MagValues_DIM321=permute(MagValues,[3,2,1]);

%Calculating Fluctuating Values
%Cycle through each set of U and V values and minus the mean
if size(uValues,3)>1
while CountZ <= size(uValues,1)
    Mean_uValues(CountZ,:) = nanmean(uValues_Dim321(:, :, CountZ));
    Mean_vValues(CountZ,:) = nanmean(vValues_Dim321(:, :, CountZ));
    Mean_MagValues(CountZ,:) = nanmean(MagValues_Dim321(:, :, CountZ));

    Std_uValues(CountZ,:) = nanstd(uValues_Dim321(:, :, CountZ));
    Std_vValues(CountZ,:) = nanstd(vValues_Dim321(:, :, CountZ));
    Std_MagValues(CountZ,:) = nanstd(MagValues_Dim321(:, :, CountZ));

    CountZ = CountZ + 1;
end

for CountA = 1:Count
    if CountA <= 1
        Fluct_uValues = uValues(:, :, (CountA)) - Mean_uValues;
        Fluct_vValues = vValues(:, :, (CountA)) - Mean_vValues;
        Fluct_MagValues = MagValues(:, :, (CountA)) - Mean_MagValues;
    else
        Fluct_uValues = cat(3, Fluct_uValues, (uValues(:, :, (CountA)) - Mean_uValues));
        Fluct_vValues = cat(3, Fluct_vValues, (vValues(:, :, (CountA)) - Mean_vValues));
        Fluct_MagValues = cat(3, Fluct_MagValues, (MagValues(:, :, (CountA)) - Mean_MagValues));
    end
end

% nanmean squared fluctuating values calculated for Turbulence intensity calculation

CountZ = 1;
Square_Fluct_uValues_Dim321 = ((permute(Fluct_uValues, [3, 2, 1])).^2);
Square_Fluct_vValues_Dim321 = ((permute(Fluct_vValues, [3, 2, 1])).^2);
Square_Fluct_MagValues_Dim321 = ((permute(Fluct_MagValues, [3, 2, 1])).^2);

while CountZ <= size(Fluct_uValues, 1)
    Mean_Square_Fluct_uValues(CountZ,:) = nanmean(Square_Fluct_uValues_Dim321(:, :, CountZ));
    Mean_Square_Fluct_vValues(CountZ,:) = nanmean(Square_Fluct_vValues_Dim321(:, :, CountZ));
    Mean_Square_Fluct_MagValues(CountZ,:) = nanmean(Square_Fluct_MagValues_Dim321(:, :, CountZ));

    CountZ = CountZ + 1;
end

% Turbulence Intensity
RMS_Fluct_uValues = (sqrt(Mean_Square_Fluct_uValues));
RMS_Fluct_vValues = (sqrt(Mean_Square_Fluct_vValues));
RMS_Fluct_MagValues = (sqrt(Mean_Square_Fluct_MagValues));

% Relative Intensity
Rel_Intensity_uValues = RMS_Fluct_uValues ./ Mean_uValues;
Rel_Intensity_vValues = RMS_Fluct_vValues ./ Mean_vValues;
Rel_Intensity_MagValues = RMS_Fluct_MagValues ./ Mean_MagValues;

% Variance
Var_uValues = Std_uValues.^2;
Var_vValues = Std_vValues.^2;
Var_MagValues = Std_MagValues.^2;
Mag_Var = sqrt((Var_uValues.^2) + (Var_vValues.^2));

% Turbulence Kinetic Energy
Fluct_wValues = ((Fluct_uValues.^2) + (Fluct_vValues.^2) + (Fluct_wValues.^2)) / 2;
Turb_Kin_Energy = ((Fluct_uValues.^2) + (Fluct_vValues.^2) + (Fluct_wValues.^2)) / 2;
Co unt Z = 1;
Turb_Kin_Energy_Dim321 = (permute(Turb_Kin_Energy,[3,2,1]));
while CountZ <= size(Turb_Kin_Energy,1)
    Mean_Turb_Kin_Energy(Cou n tZ,:)=nanmean(Turb_Kin_Energy_Dim321(:, :, CountZ));
    CountZ = CountZ + 1;
end
end

% House Keeping
clear TempLocation, clear fid, clear FileNameCount, clear CountA, clear Count, clear EndFileNameCount;
clear CountX, clear FileName, clear Ext, clear FileNameCountStr;
clear uValues_Dim321, clear vValues_Dim321, clear MagValues_Dim321, clear Turb_Kin_Energy_Dim321;
clear Square_Fluct_uValues_Dim321, clear Square_Fluct_vValues_Dim321, clear Square_Fluct_MagValues_Dim321, clear CountZ;
clear Mean_Square_Fluct_uValues, clear Mean_Square_Fluct_vValues, clear Mean_Square_Fluct_MagValues;
clear ans;

% Main program to derive tangential velocities about a swirl centre
% breaks a velocity field down to get m/s about centre
i = sqrt(-1); % set i to be equal to complex operator should be this already but just safe to do this here
velocities = uValues + (vValues * i); % set velocities to be complex variable ust because i want to
velocities_angle = angle(velocities); % gives vector angles in x and y
velocities_angle = angle(velocities); % gives vector angles
Swirl_Centre = [u_location v_location]; % uses defined centre to specify a centre location

% Swirl Centre defined then calculate position vectors from this
% point to each individual interrogation region
position = xValues + (i*yValues);
Swirl_Centre_Position = position(Swirl_Centre(1,2), Swirl_Centre(1,1));
% Take defined swirl centre and finds its position
position_vectors = position - Swirl_Centre_Position;
% Position vectors for each velocity vector relative to centre
unit_position_vectors = position_vectors ./ (abs(position_vectors)); % Unit position vectors
unit_angle = angle(unit_position_vectors); % Angle of the unit position vectors

tangent unit_vectors = imag(unit_position_vectors) + (real(unit_position_vectors)*-i); % unit vector in the tangent direction (normal to the position vector
% Angle of the tangent unit vector
alpha = (velocities_angle) - (tangent_unit_angle); % Difference between velocity angle and tangent angle

tangent_mag = abs(velocities) .* cos(alpha); % Tangent vector magnitude by trig
Vel_Tang = (tangent_mag .* real(tangent_unit_vectors)) + (tangent_mag .* imag(tangent_unit_vectors)*i); % Turn tangent magnitude into tangent vector using the tangent unit vector in the x and y components

radial_mag = abs(velocities) .* sin(alpha); % Radial vector magnitude by trig
Vel_Radial = (radial_mag .* real(unit_position_vectors)) + (radial_mag .* imag(unit_position_vectors)*i); % Turn radial magnitude into radial vector using the unit position vector in the x and y components

circ = (2.*pi.*abs(position_vectors))./1000; % Circumference in Meters
ang_vel = (tangent_mag ./ circ) / (2*pi); % Angular Velocity in Radians
rev_per_min=(Vel_Tang./circ).*60; % RPM of the flow
% rev_per_min2=(tangent_mag./circ).*60;

Swirl_No=rev_per_min/Engine_Speed; %Normalized RPM of the flow

l_dot=(((abs(position_vectors)*1000).*(ang_vel))*int_region_mass); %Inertia of the flow

CountZ=1;
Vel_Tangent_Dim321=permute(Vel_Tang,[3,2,1]);
Vel_Radial_Dim321=permute(Vel_Radial,[3,2,1]);

rev_per_min_Dim321=permute(rev_per_min,[3,2,1]);
Swirl_No_Dim321=permute(Swirl_No,[3,2,1]);

if(size(uValues,3)>1
    while CountZ<=size(uValues,1)
        Mean_Vel_Tang(CountZ,:)=nanmean(real(Vel_Tangent_Dim321(:,CountZ)))+nanmean(imag(Vel_Tangent_Dim321(:,CountZ)))*i;
        Mean_Vel_Radial(CountZ,:)=nanmean(real(Vel_Radial_Dim321(:,CountZ)))+nanmean(imag(Vel_Radial_Dim321(:,CountZ)))*i;
        Mean_rev_per_min(CountZ,:)=nanmean(real(rev_per_min_Dim321(:,CountZ)))+nanmean(imag(rev_per_min_Dim321(:,CountZ)))*i;
        Mean_Swirl_No(CountZ,:)=nanmean(real(Swirl_No_Dim321(:,CountZ)))+nanmean(imag(Swirl_No_Dim321(:,CountZ)))*i;

        CountZ=CountZ+1;
    end
end

%SaveVectors in TecPlot
if size(uValues,3)>1
    SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),Mean_uValues,Mean_vValues,'EnsembleAverage');
    SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),RMS_Fluct_uValues,RMS_Fluct_vValues,'TurbulenceIntensity');
    SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),real(Mean_Vel_Tang),imag(Mean_Vel_Tang),'MeanTangentialVelocities');
    SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),real(Mean_Vel_Radial),imag(Mean_Vel_Radial),'MeanRadialVelocities');
    SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),real(Mean_rev_per_min),imag(Mean_rev_per_min),'MeanRPM');
    SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),real(Mean_Swirl_No),imag(Mean_Swirl_No),'MeanSwirlNo');
end

SaveForTechPlot_20050227(xValues(:,:,1),yValues(:,:,1),real(Vel_Tang),imag(Vel_Tang),'TangentialVelocities');
% Radial Profiles Part of the program to generate the information for the profile plots.

clear RadialOrder  %clear theses for safely as if they exist already then it messes the rest up

clear RadialMean

for Count=1:size(xValues,3);
    RadialOrder(:,1,Count)=reshape(abs(position_vectors(:,Count)),[],1,1);
    RadialOrder(:,2,Count)=reshape(abs(Vel_Tang(:,Count)),[],1,1);
    RadialOrder(:,3,Count)=reshape(abs(Vel_Radial(:,Count)),[],1,1);
    RadialOrder(:,4,Count)=reshape(abs(rev_per_min(:,Count)),[],1,1);
    RadialOrder(:,5,Count)=reshape(abs(Swirl_No(:,Count)),[],1,1);

    RadialOrder(:,Count)=sortrows(RadialOrder(:,Count));
end

RadialMean=ones(size(RadialOrder,1),5,Count)*NaN;

Count=1;
while Count<=size(xValues,3);
    CountN=0;
    CountR=1;
    while CountN<size(RadialOrder,1)-1;
        CountZ=1;
        while round(RadialOrder(CountN+CountZ,1,Count)*(10^number_of_dp))/(10^number_of_dp)>=round(RadialOrder(CountN+CountZ+1,1,Count)*(10^number_of_dp))/(10^number_of_dp);
            CountZ=CountZ+1;
        end
        if CountZ<=1;
            RadialMean(CountR,:,Count)=RadialOrder(CountN+1,:,Count);
        else
            RadialMean(CountR,:,Count)=nanmean(RadialOrder(CountN+1:CountN+CountZ,:,Count));
        end
        CountN=CountN+CountZ;
        CountR=CountR+1;
    end
    Count=Count+1;
end

RadialMean_Dim321=permute(RadialMean,[3,2,1]);
RadialMean_Avg=nanmean(RadialMean_Dim321,1);
RadialMean_Avg=permute(RadialMean_Avg,[2,1,3]);

RadialMean=permute(RadialMean,[2,1,3]);
RadialMean_Avg=permute(RadialMean_Avg,[2,1,3]);

Count=1;
for FileNameCount=1:size(xValues,3);
    FileNameCountStr=sprintf('%05.0f',FileNameCount);
    FileName=strcat('BowProfil Data',FileNameCountStr,'dat');
    HeaderData=strcat('TITLE=',FileNameCountStr,'n VARIABLES = "',startFileName,'", u_location =',u_location,' v_location =',v_location,Engine_Speed=',Engine_Speed,' No_dp =',number_of_dp,'Radius (mm), Tangential Vel (m/s), Radial Vel (m/s), RPM, Swirl Number');

fid=fopen(FileName,'w');
fprintf(fid,'%s
',HeaderData);
fprintf(fid,'%6f %6f %6f %6f %6f
',RadialMean(:,:,Count));
close(fid);
Count=Count+1;
end

FileNameCount = 1;
FileNameCountStr = sprintf('%05.0f',FileNameCount);
FileName = strcat('BowiProfileMeanData',FileNameCountStr,'.dat');
HeaderData = strcat('TITLE=', FileName, '\n','VARIABLES = "StartFileName","u_location = ', u_location, ', v_location = ', v_location, ', Engine Speed = ', Engine_Speed,' ,
No_dp = ', number_of_dp,'\nRadius (mm), Tangential Vel (m/s), Radial Vel (m/s), RPM, Swirl Number"');
fid=fopen(FileName,'w');
fprintf(fid,'%s
',HeaderData);
fclose(fid);

%----------------------------------------------------------------------------------------------------------------------

if size(uValues,3)>1
   %Plot Figures
   figure(1)
   clf
   subplot(2,2,1);
   quiver(Mean_uValues,Mean_vValues)
   axis('image');
   title('Ensemble Average');
   subplot(2,2,2);
   quiver(real(Mean_Vel_Tang(:,:,1)),imag(Mean_Vel_Tang(:,:,1)),2);
   axis('image');
   title('Mean Tangential Velocities');
   subplot(2,2,3);
   plot(RadialMean_Avg(:,:,1),RadialMean_Avg(:,:,2),'xr');
   title('Mean Tangential Velocities');
   subplot(2,2,4);
   plot(RadialMean_Avg(:,:,1),RadialMean_Avg(:,:,4),'xb');
   title('Mean RPM');
end

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