AN EXPERIMENTAL AND COMPUTATIONAL STUDY OF FLOW IN THE
SQUISH JET COMBUSTION CHAMBER

by

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ABSTRACT

Fast controlled burning of the fuel-air charge inside combustion chambers of spark ignition engines typically leads to reductions in specific fuel consumption and exhaust emissions. These advantages are especially useful in engines running on natural gas—a fuel with a relatively low flame speed. ‘Squish-Jet’ combustion chambers have a unique geometry that forms jets of gas that converge radially inwards as TDC (Top Dead Centre) is approached. The turbulence generated by the jets has been shown to increase the burn rate and the positive results warrant numerical optimisation of Squish-Jet combustion chambers. In this study a prerequisite to the optimisation was carried out, namely, testing the validity of numerical flow predictions inside Squish-Jet combustion chambers. The validity study also led to a greater understanding of the flow processes inside these complex combustion chambers. The CFD (Computational Fluid Dynamics) code chosen for the numerical work was KIVA-3V.

Ensemble mean velocities and turbulence intensities that were measured in three different combustion chambers exhibiting squish flow were compared with corresponding values from KIVA-3V. One of the combustion chambers was a plain Bowl-in-Piston type while the remaining two were Squish-Jet chambers (named ‘Squish-Jet 1’ chamber and ‘Squish-Jet 2’ chamber). While each Squish-Jet chamber had a 10 mm wide fence surrounding the piston bowl rim, Squish-Jet 1 chamber had a taller fence with wider squish grooves than Squish-Jet 2 chamber. Measurements were made by PIV (Particle Image Velocimetry) and LDV (Laser Doppler Velocimetry) and in order to closely reproduce the initial and boundary conditions set in KIVA-3V, the UBCRICM (University of British Columbia Rapid Intake and Compression Machine) was used. PIV enabled the in-cylinder flow to be visualised in planes parallel to the cylinder head, while LDV produced single-point velocity data that were used to more accurately determine rms velocity fluctuations. The microscopic particles used for PIV and LDV were seeded into each combustion chamber by a novel system developed to suit the momentary flow in the UBCRICM. The combustion chambers did not contain fuel and they were transparent to allow optical access for the laser measurements. A compression ratio of 9.1:1, a squish clearance of 2 mm and a crank speed of 800 rev/min were used for all tests.
The numerical and experimental results indicated that Squish-Jet chambers tend to generate more
turbulence than plain Bowl-in-Piston chambers, even though the former may have smaller squish
velocities. This affirms the potential for Squish-Jet chambers to accelerate combustion; thus
improving thermal efficiency and emissions in reciprocating internal combustion engines.

Although the measured and predicted rms velocity fluctuations were all below 1.7 m/s, the LDV
measurements made at the jet opening of one of the Squish-Jet chambers revealed some flaws in
the $k$-$\epsilon$ turbulence model used in KIVA-3V. The assumption of turbulence isotropy in KIVA-3V
was violated there, as the rms velocity fluctuations in the radial direction were higher than in the
tangential direction especially for crank angles prior to 10 degrees BTDC (Before Top Dead
Centre). In addition, the turbulent kinetic energy was generally overestimated while the turbulent
viscosity may have been underestimated. Consequently, it is likely that KIVA-3V overpredicted
the dissipation rate of the turbulent kinetic energy.

The general mean flow trends (with respect to time and space) measured in important regions of
each squish chamber were well predicted by KIVA-3V. Nevertheless, weaker predictions were
made at the (only measurable) squish groove outlet for each Squish-Jet chamber due to a
suspected local PIV error. Aside from this, the mean radial velocity was overpredicted by at most
17%. The symmetrical inward migration of squish flow that was observed by PIV was present in
KIVA-3V output. KIVA-3V also forecast the abatement of the squish velocity observed at TDC.
The greater turbulence intensity observed just inside the piston bowls near TDC was also
predicted. These flow predictions are valuable for chamber shape optimisation because they can
be used to develop trends governing a chamber’s propensity to produce turbulence and to
transport it toward the ignition point.
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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>2-D</td>
<td>Two dimensional</td>
</tr>
<tr>
<td>3-D</td>
<td>Three dimensional</td>
</tr>
<tr>
<td>( \nabla )</td>
<td>Gradient operator (( \nabla = \frac{\partial}{\partial x} + \frac{\partial}{\partial y} + \frac{\partial}{\partial z} ))</td>
</tr>
<tr>
<td>( a )</td>
<td>Acceleration of gas parcel</td>
</tr>
<tr>
<td>( A )</td>
<td>Area</td>
</tr>
<tr>
<td>ALE</td>
<td>Arbitrary Lagrangian Eulerian method (KIVA numerical technique)</td>
</tr>
<tr>
<td>ATC</td>
<td>At top dead centre</td>
</tr>
<tr>
<td>B</td>
<td>Pixel brightness level</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom dead centre</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake Specific Fuel Consumption</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before top dead centre</td>
</tr>
<tr>
<td>( c )</td>
<td>Constant; specific heat capacity</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>( d )</td>
<td>Diameter; displacement between PIV images; distance from squish-jet opening</td>
</tr>
<tr>
<td>DNS</td>
<td>Direct Numerical Simulation</td>
</tr>
<tr>
<td>( f )</td>
<td>Frequency; focal length</td>
</tr>
<tr>
<td>( \bar{g} )</td>
<td>Specific gravitational force</td>
</tr>
<tr>
<td>IC</td>
<td>Internal combustion</td>
</tr>
<tr>
<td>ID</td>
<td>Inner diameter</td>
</tr>
<tr>
<td>( i )</td>
<td>Position index in x direction</td>
</tr>
<tr>
<td>( \hat{i} )</td>
<td>Unit vector in the x direction</td>
</tr>
</tbody>
</table>
\( I \) Specific internal energy

\( j \) Position index in y direction

\( \vec{j} \) Unit vector in the y direction

\( \bar{j} \) Heat conduction

\( k \) Specific turbulent kinetic energy; Position index in z direction

\( \vec{k} \) Unit vector in the z direction

\( K \) Effective thermal conductivity

\( l \) Characteristic size

\( L \) Length

\( \text{LDV} \) Laser Doppler Velocimetry

\( N \) Number (usually of measurements)

\( \text{NXP} \) Number of computational cell divisions in x direction

\( \text{NYP} \) Number of computational cell divisions in y direction

\( \text{NZP} \) Number of computational cell divisions in z direction

\( \text{OD} \) Outer diameter

\( P \) Pressure

\( Pr \) Prandtl number

\( \text{PIV} \) Particle Image Velocimetry

\( \vec{r} \) Position vector

\( \text{RANS} \) Reynolds Navier-Stokes Simulation

\( \text{RCM} \) Rapid compression machine

\( \text{Re} \) Reynolds number

\( \text{RON} \) Research octane number

\( s \) Measured standard deviation
SI Spark ignition
STP Standard temperature and pressure (273.15K and 101325 Pa)
t Time; Student’s t-value
T Period; Temperature
TDC Top dead centre
u Velocity fluctuation; speed in x direction
\( \ddot{u} \) Fluid velocity vector
U Measured velocity; mean flow
UBCRICM University of British Columbia Rapid Intake and Compression Machine
v Velocity; velocity in y direction
V Volume
x x coordinate
y y coordinate
\( y \) Distance of cell centroid to neighbouring wall
w Velocity in z direction

Greek
\( \Delta \) Finite change
\( \Delta x \) LDV fringe separation
\( \varepsilon \) Dissipation rate of specific turbulent kinetic energy
\( \theta \) Crank angle; half angle
\( \lambda \) Light wavelength
\( \mu \) Viscosity of fluid, true mean
\( \mu_{\text{eff}} \) Effective coefficient of viscosity
\[
\begin{align*}
\rho & \quad \text{Density} \\
\sigma & \quad \text{True standard deviation} \\
\bar{\sigma} & \quad \text{Viscous stress tensor} \\
\tau & \quad \text{Characteristic eddy time} \\
\varphi & \quad \text{Half angle between LDV laser beams} \\
\chi^2 & \quad \text{Dependent variable for the chi square distribution} \\
\omega & \quad \text{Angular frequency}
\end{align*}
\]

**Subscripts**

1. Refers to cylindrical lens; first PIV image; zone 1 (Appendix A)
2. Refers to negative spherical lens; second PIV image; zone 2 (Appendix A)

\(b\) \quad \text{Refers to beam}

\(B\) \quad \text{Refers to Bragg shift}

\(c\) \quad \text{Refers to cycles}

\(d\) \quad \text{Refers to the divergence; Doppler effect in LDV}

\(f\) \quad \text{Refers to fluid surrounding a particle}

\(\text{fluc}\) \quad \text{Refers to fluctuation component}

\(h\) \quad \text{Refers to high}

\(i\) \quad \text{Refers to cycle number; integral scale}

\(j\) \quad \text{Refers to measurement number}

\(\text{int}\) \quad \text{Refers to PIV interrogation region}

\(k\) \quad \text{Refers to Kolmogorov}

\(l\) \quad \text{Refers to leakage}

\(\text{LDV}\) \quad \text{Refers to Laser Doppler Velocimetry}

\(m\) \quad \text{Refers to mean}
n  Refers to normal direction
p  Refers to particle; piston
rel Refers to particle relative to the air
rms Refers to the root of the mean square
s  Refers to the squish flow area
t  Refers to total; turbulent
x  x coordinate
y  y coordinate

Superscripts
a  Refers to the axial (longitudinal) direction
r  Refers to the radial direction
t  Refers to the tangential (azimuthal) direction
iso Refers to the isotropic value

Miscellaneous
Overbars denote the mean.

Periods above variables denote the temporal rate.

Primed variables denote fluctuation about their mean.
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1 INTRODUCTION

1.1 Improving Thermal Efficiency and Exhaust Emissions

Since the automobile’s inception over a century ago, oil reserves have been continually declining and petroleum companies are finding it increasingly difficult to locate feasible deposits (Campbell and Laherrere, 1998). Meanwhile, political differences between major oil consuming nations in the West and key supplier nations in the Middle East threaten daunting price hikes like those witnessed in the seventies oil crisis. Environmental protection agencies world-wide are also enforcing increasingly stringent IC engine emission regulations in response to the alarming level of harmful pollutants in the atmosphere. It is therefore quite evident that petroleum fuels are increasingly posing an economic and environmental threat to society.

In 1998, SI engines fuelled by gasoline accounted for 78% of the total vehicle energy consumption in the USA (United States Energy Information Administration, 1998). Emissions from SI engines can be reduced without necessitating large manufacturing changes by using an alternative combustible fuel that is economically viable. Due to its ready abundance, its low cost and its low carbon/hydrogen ratio, methane seems a promising candidate to satisfy these criteria. Furthermore, as indicated by its RON of 120 (Heywood, 1988) methane exhibits a very high resistance to ‘knock’. Knock is the objectionable noise coming from a combustion chamber in which the burn rate is exceedingly high. The extremely rapid combustion also often causes component damage. Methane’s superior resistance to knock allows engines to run with higher compression ratios that in turn deliver higher thermal efficiencies. Methane fuelled SI engines
with compression ratios and thermal efficiencies close to 15.8:1 and 40% respectively have been reported (Das, 1995). These values represent a considerable improvement on conventional gasoline fuelled SI engines which generally have compression ratios on the order of 9:1 and maximum thermal efficiencies on the order of 30% (Heywood, 1988). Methane is the predominant component in natural gas for vehicles. Despite hydrogen's potential to offer zero tailpipe emissions, it is not readily available and its production plants need energy to electrolyse water or to crack hydrocarbons. This energy represents the ecological and economical price paid for zero tailpipe emissions. Although hydrogen boasts a relatively high flame speed, its low RON of 60 (Ingersoll, 1996) is a further disadvantage, making the fuel very prone to knock. Moreover, SI engines require considerable modification in order to prevent backfire caused by hydrogen's broad flammability limits.

One drawback to methane is its slow combustion rate in SI engines. This is related to the inherently slow laminar flame speed in premixed methane and air which has been documented by Hill et al. (1988) and Glassman (1987). In order to further exploit the advantages of natural gas, its flame speed requires enhancement. Fast combustion is particularly advantageous in SI engines because it causes a pressure rise near top dead centre (TDC) that approaches the form for the Otto cycle. The Otto cycle ideally represents the processes inside the SI engine and it offers maximum thermodynamic efficiency. Despite the increase in peak pressure and end-gas temperature associated with faster burning, the chance of knock decreases because the end-gas is exposed to the extreme conditions for shorter periods (Lappas, 1996). The end-gas is the last portion of charge to be burnt inside the combustion chamber. Fast burning also helps to extend the lean operational limit for SI engines. Lean operation reduces engine pumping losses and tailpipe emissions. In addition, lean mixtures possess high specific heat ratios, which further enhance thermal efficiency.
1.2 The Squish-Jet Combustion Chamber

The rate at which the premixed fuel-air charge inside a combustion chamber is burned depends heavily on the total flame frontal area. There are two general ways of increasing this area:

a) **Introduction of more flame fronts.** This can be achieved by increasing the number of ignition sites. More flame fronts can also be created by splitting up an existing flame front through the action of significant turbulence. The turbulence also acts to distribute the resultant flame fronts into the unburnt mixture.

b) **Increasing the area per flame front.** The first step in achieving this is to ensure that the contact time between the chamber wall and the flame is as short as possible. This usually means that the spark plug is centrally placed within the combustion chamber. The next step is to increase the area in each flame front by wrinkling it with turbulence.

The combustion chamber walls of contemporary multi-valve SI engines have limited room for extra spark plugs. Consequently, combustion research at UBC has focussed more on turbulence to promote burn rates.

It is broadly acknowledged that turbulence inside a SI engine's combustion chamber has a profound effect on the burn rate. As Groff and Matekunas (1980) reiterate, turbulent flames in SI engines possess speeds that are many times the laminar flame speed of the same fuel-air mixture
at normal engine temperatures and pressures. In principle, turbulence in the engine cylinder at
the moment of ignition can be augmented through:

- Higher shear forces in the annular jets that issue from the intake valves,
- Compression and breakdown of large scale eddies that form during the intake stroke (e.g.,
  Swirl and tumble),
- Utilising appropriate combustion chamber geometry to provide squish-generated shear flow.

Squish is the radially inward gas flow that results when a portion of the piston crown and
cylinder head approach each other closely toward the end of the compression stroke
(Heywood, 1988).

The first two processes can be categorised as induction-generated turbulence, because the
induction stroke is responsible for introducing the flow structures that eventually decay to form
the turbulence at the instant of ignition. Arcoumanis and Whitelaw (1987) have observed that for
practical inlet valve lifts, the peak induction-generated turbulence intensity at ignition is between
0.4 and 0.5 times the mean piston speed.

Turbulence from squish is appropriately generated close to the ignition time. Flame enhancement
is desirable at this early stage because the ignition delay greatly affects the total burn duration.
The ignition delay period typically occupies a large portion (30%) of the total combustion
duration and it is defined by Heywood (1988) as the time taken to burn the first 10% of the total
air-fuel charge mass. Squish turbulence can effectively shorten the ignition delay by providing
intense turbulence with an integral scale that is on the order of the flame kernel size. The small
clearance in a squish chamber produces the fine scales. A typical squish chamber is depicted in
Nevertheless, as noted by Anetor (1994), one disadvantage in standard squish chambers is that the turbulence is localised around the bowl lip and not at the central spark plug. Turbulence may be brought nearer to the spark plug by channelling the squish flow into jets that are directed towards the centre of the combustion chamber. Research by Nakamura et al. (1978) supports this remedy because the introduction of small diameter air-jets towards the spark plug just before ignition was found to significantly increase the burning rate due to the enhancement of the turbulence field near the spark plug.

The combination of squish flow and jet formation led to the development of the 'Squish-Jet' combustion chamber, which was patented by Evans (1986). In this chamber type, radial jets of cylinder gas impinge on each other in a central piston bowl as the piston approaches TDC. The turbulence generated by the jets provides a favourable environment for flame growth initiated by a central spark plug. Figure 5.2 shows the key features of a piston used for a Squish-Jet chamber. Previous experiments conducted on a Ricardo test engine running on natural gas show a 5% reduction in the minimum BSFC when it was fitted with a Squish-Jet chamber (Blaszczyk and Evans, 1995).

1.3 Research Objectives

The evident potential for the Squish-Jet chamber warrants pursuit of the two following main objectives in this study:

- To develop a greater understanding of the flow processes inside the complex geometry of Squish-Jet combustion chambers.
To assess the validity of the CFD code, KIVA-3V, as a tool for predicting flow in such complex combustion chambers.

It is worth noting that engaging in the latter objective facilitates the realisation of the former. The KIVA family of codes (Amsden, 1989) was chosen because of its popularity and its documented refinement over 15 years. When combined with an accurate combustion model, KIVA-3V may eventually be used to optimise the geometry of Squish-Jet combustion chambers for thermal efficiency and emissions.
2 LITERATURE REVIEW

KIVA validation work has been done by a host of researchers in the past for a multitude of combustion chamber geometries, fuels and other engine parameters. Despite the range of applications, no record of KIVA validation work for Squish-Jet combustion chambers has been found in the literature. Nevertheless, some previous work has findings that offer foresight to the present objectives.

2.1 Flow Validity Studies on Engines

By far the most flow validation work for the KIVA family of codes has been done on reciprocating IC engines. It should be noted that although versions of the KIVA code vary in detail they share the same fundamental equations. The test engines have been operated in both the fired mode and the motored mode. Validation studies for fired operation are ignored in this review primarily because the conditions at the beginning of the compression stroke are less well known than for a motored engine. The fired initial conditions include many more variables such as gas species concentrations and temperature variations and their added uncertainties further compromise the validation study. Furthermore, since the gas flow modelling abilities of KIVA are only of interest here, combustion and fuel injection is avoided.

Although not as error prone as fired operation, motored engines are not devoid of uncertainty in initial conditions. Accurate and spatially resolved measurements of flow descriptors such as mean velocity, turbulence intensity (root mean square of the turbulent velocity fluctuations) and temperature are not trivial tasks. The cycle-to-cycle variation in these quantities further compounds the problem. Many researchers attempt to filter out the cyclic velocity variations by cycle-resolving techniques. The techniques usually involve choosing a cut-off frequency below which the bulk flow of a cycle is deemed to reside. The choice of the cut-off frequency is an inexact science and as noted by Tabaczynski (1983), it is highly improbable that a separation of turbulence from the cyclic mean flow can be accomplished. This is because the flow frequency components of turbulence are on the order of cyclic velocity changes. The inability to reliably compensate for cyclic changes and uncertainty in the initial conditions warrants their prevention
in the first place. This is more easily achieved with the use of rapid compression machines in lieu of motored engines. Nevertheless, the ready availability of motored reciprocating engines in research facilities makes them desirable for CFD validation work, despite their innate flaws.

The first documented validation work for KIVA was done by McKinley, Primus, O’Rourke and Butler (1988). The researchers used the KIVA-II code with its $k$-$\varepsilon$ turbulence model to predict the swirling air motion in a motored engine with both circular and square piston cups. The pistons were otherwise flat. Their numerical results were compared with cycle-resolved LDV (Laser Doppler Velocimetry) measurements for the same engine geometries. This velocity measurement technique is described in chapter 3 of this document. Quantities that were compared were mean radial and tangential velocities and turbulence intensities.

For the most part McKinley et al. found that KIVA-II reproduced the swirl velocity measurements with a high level of accuracy. The authors were not surprised by the agreement because previous validation studies using the $k$-$\varepsilon$ turbulence model yielded similar matches. The predicted mean radial velocities were also in close agreement to measured values, especially near the bowl lip.

Although peak and spatially averaged values of predicted turbulence intensity were within 14% of measured values prior to TDC, significant discrepancies were found after TDC. The authors connected them to the $k$-$\varepsilon$ model’s possible inadequacy for representing the change in dissipation rate of turbulent kinetic energy ($\varepsilon$) during expansion.

Although the flow validity study done by McKinley et al. suggested that KIVA represents reality for simple bowl-in-piston chambers up to the ignition point, there was no assurance of similar performance for the more complex squish-jet chambers. It was possible, for example, that the suspected flaws in $\varepsilon$ determination may appear before TDC for squish-jet chambers due to their more complex geometries.

Zur Loye, Siebers, McKinley, Ng, and Primus (1989) also took cycle-resolved LDV measurements in a motored single cylinder engine and compared the results with KIVA (the prototype code of the KIVA family). The engine had an axisymmetric piston with a toroidal bowl placed in the centre. The quantities that were measured and compared with KIVA were
tangential mean velocity and turbulence intensity at various locations in a plane 9 mm below the cylinder head for a range of crank angles.

The researchers measured weak swirling and tumbling motions (resulting from valve flow) which could not be modelled by the 2-D (axisymmetric) KIVA code. Nevertheless, an equivalent 2-D initial flow field was assumed and only the predicted turbulence intensity field at TDC was compared with LDV measurements. The measured turbulence intensity at 60 degrees BTDC and 0.25” away from the bowl centre was approximately 20% lower than the computed value. This difference gradually decreased to 2% at TDC.

These results are highly suspect, given the unrealistic imposition of axisymmetric flow. Furthermore, the initial turbulence intensity at the beginning of the compression stroke was found by estimating the production of turbulence through the inlet valve during the intake stroke. The estimation was made with empirical equations derived by another researcher (El Tahry, 1984) and the turbulent kinetic energy at the start of the intake stroke was unsupportedly assumed to be 10% of the mean piston speed squared. The authors of the study also acknowledged that the uncertainty of the measured turbulence intensity strongly depends on the choice of cut-off frequency for cycle-resolving.

The flaws in this study serve to emphasise the importance of simple and well-defined initial conditions. For example, if the initial flow was quiescent with no turbulent kinetic energy, the 2-D KIVA model would have been applicable and the initial conditions would not have been as uncertain.

Sweetland and Reitz (1994) compared PIV measurements of gas velocity with results from KIVA-3. The measurements were made in a plane inside the toroidal piston bowl of a motored Diesel engine for a number of crank angles. PIV is a measurement technique in which displacements of particles moving in a thin plane of light are globally determined over a small time period. The technique is described in detail in chapter 3 of this document.

Sweetland et al. used two separate PIV velocity plots to obtain approximate values for turbulence intensities that were then used to estimate the turbulence kinetic energies. They found that their estimates compared well with those from KIVA-3. Nevertheless, their finding is not of much
consequence because the small number of samples \((n=2)\) that were used for their fluctuation intensity calculations subjected their estimate to an extremely large uncertainty. For example, statistical analysis (Bhattacharyya and Johnson, 1977) shows that even for six samples the uncertainty is around 100\% and that at least 100 samples are needed to achieve an uncertainty of 13\%. In comparison with LDV measurements, the low data rate and poor spatial resolution of most PIV systems makes PIV a less suitable technique for determining turbulence intensities. The average velocities that Sweetland et al. determined also had very high levels of uncertainty because of the limited amount of data.

A KIVA-3 validity study on a motored engine was also done by Amsden, O'Rourke, Butler, Meintjes and Fansler (1992). The experimental measurements were made by LDV on a crankcase scavenged two-stroke engine with a flat piston. Amsden et al. made no attempt to quantify the differences in velocity they observed between KIVA-3 and measurements; however despite differences in detail, the basic scavenging loop was well replicated. The researchers also noted that the measured turbulence was generally isotropic except for one crank angle (270 degrees ATDC). The largest differences between predicted and measured flow fields were found when the ports were opening on the expansion stroke of the engine. The probable cause presented was that ring crevice volumes and rounded port corners were not accounted for in the model. Weak swirl observed in the measurements that was not predicted by KIVA-3 was attributed to a slightly tilted piston or a pressure differential in the crankcase. The most significant discrepancies between measurements and predictions were for the turbulence intensity. Predicted values were as much as 50\% lower than the measured ones. One of the possible causes cited was the significant error in the measured values.

The explanations offered by Amsden et al. for the discrepancies between KIVA-3 and measurements had a common factor—experimental problems. Many of these problems are avoided if the experimental boundary conditions match those of the computer model more closely.

Auriemma, Corcione, Macchioni and Valentino (1995) conducted a study to check the validity of the \(k-\epsilon\) turbulence model in KIVA-II. The validity study was based on integral length scale measurements made at a fixed location in the cylinder of a motored engine with a two-probe LDV system. Spatial autocorrelation methods were used on the measured velocities to determine
the integral scales. Auriemma et al. found that in order to achieve a close match between numerical and experimental values of length scale, one of the constants in the $k$-$\varepsilon$ model ($C_{\mu e}$) should be increased by 32%. $C_{\mu e}$ is a constant relating the length scale to specific turbulent kinetic energy ($k$) and its dissipation rate ($\varepsilon$). The researchers also found that the predicted mean velocities were insensitive to this change and that they closely agreed with measured values.

Aside from the possibility that the $k$-$\varepsilon$ model in KIVA requires some refinement, the prevalent flaw noted in most of the validity studies was the inadequate numerical replication of experimental initial and boundary conditions. Despite the advantage offered by rapid compression machines in this domain, few reports of their use in CFD validity studies were found.

### 2.2 Flow Validity Studies on Rapid Compression Machines

Jakirlic, Volkert, Pascal, Hanjalic and Tropea (2000) reported LDV measurements made in a unique rapid compression machine (RCM) comprising a cylinder which could be spun on its axis while the piston was at BDC. The RCM had a flat piston crown and was operated in three modes: steady cylinder rotation, transient spin-down without the compression stroke and transient spin-down with the compression stroke. The compression ratio was 5:1 and two cylinder head geometries were tested: a flat cylinder head and a squish configuration with a bowl in the centre. A spin-down test entailed suddenly halting the steady spinning of the cylinder, thus forming a well-defined swirling flow inside it.

The measured mean velocities in the axial and tangential directions were compared with corresponding values from DNS (Direct Numerical Simulation) and from RANS (Reynolds Averaged Navier-Stokes Simulation) codes. Except for the beginning of the compression stroke tests, the measurements agreed very closely to predictions from both models. The results from DNS and the RANS codes were always close. The velocity discrepancies between the simulations and the measurements at the start of compression were attributed to leakage in the gap between the portion of the cylinder that could spin and the stationary portion. While DNS directly solves the Navier-Stokes equations with a spatial resolution that is small enough to avoid the turbulent Reynolds stresses, RANS codes have coarser spatial resolution for which these
stresses must be modelled. The RANS method chosen by Jakirlic et al. was of the second-
moment closure type, which models the Reynolds-stress tensor by accounting for turbulence
anisotropy. Conversely, turbulence isotropy is assumed in the simpler $k$-$\varepsilon$ RANS model that is
available in the KIVA codes. Jakirlic et al. did not make comparisons with the standard $k$-$\varepsilon$
model because of their contention that the experimental flow had strong anisotropic turbulence.
In KIVA, only the anisotropic effects near walls are accounted for with the Prandtl-von Karman
law of the wall.

Although Jakirlic et al. did not present measurements or predictions of turbulence intensity, the
more controlled environment of an RCM (than of motored engines) for creating initial conditions
was apparent. Their realistic numerical predictions may also be attributed to the comprehensive
numerical models they used. Nevertheless, the $k$-$\varepsilon$ turbulence model could not be dismissed
because predictions with this model were not presented for comparison with the second-moment
RANS results.

The only known validity study of the KIVA code involving a rapid compression machine was
conducted by Anetor (1994). Anetor took LDV measurements inside two different combustion
chambers set-up in the University of British Columbia Rapid Intake and Compression Machine
(UBCRICM). One combustion chamber had a re-entrant bowl in the piston while the other had a
simple piston bowl. These geometries produced squish near TDC. While both bowls had flat
bottoms, the re-entrant bowl had concave side-walls and the simple bowl had convex sides. Both
bowls were centrally located and their axisymmetry enabled KIVA-II to be run in the faster 2-D
mode. No Squish-Jet chambers were tested.

Swirl was also introduced into the cylinder via tangential ports just before the compression
stroke. It is strongly suspected that the purpose of the UBCRICM to provide well-defined initial
conditions was defeated because of the poor characterisation of the swirling flow that was
introduced. For example, the peak swirl velocity was estimated by approximating the pressure
drop across the inlet port and applying it to Bernoulli's equation. No discharge coefficient was
used. Furthermore, without experimental confirmation, the swirl profile was assumed to fit a
Bessel function of the first order and the initial turbulence kinetic energy was assumed to be
uniform and equal to 10% of the square of the mean piston speed.
Despite the suspect initial conditions, the mean velocities predicted in both the radial and tangential directions inside both bowl types showed very close agreement with measurements at all crank angles tested. The turbulence intensities predicted inside the re-entrant bowl also matched well with those measured; however, the same was not true for the simple bowl. In this case, the turbulence intensities were underpredicted by at least 20% and by at most 80%. The discrepancies were particularly noticeable near the bowl centre and they may have been caused by swirl centre precession that was not accounted for by KIVA-II. The precession may be more prevalent in the simple bowl because the flow is less confined than in the concave walls of the re-entrant bowl.

2.3 Ramifications for Present Research

The flaws and advantages drawn from validity studies of CFD codes that simulate engine flows have formed the basis for a methodology to satisfy a key objective of this study, namely, to assess the validity of KIVA flow predictions in Squish-Jet combustion chambers.

The recurring problem of unmatched initial and boundary conditions between the numerical model and experiments were to be addressed by using the UBCRICM only in the compression stroke. Each experiment should begin with no flow inside the cylinder. This null flow field is easy to program in KIVA.

The KIVA-3V version of the code should be used to more easily model the complex 3-D geometry inherent in Squish-Jet chambers.

A PIV system should be developed because it was recognised as a useful means of capturing the evolution of global velocity information in a plane. Although the method is adequate for mean velocity determination, its lack of spatial resolution makes single point measurements more suitable for turbulence intensity determination. Consequently, the PIV measurements should be complemented by LDV data.
3 EXPERIMENTAL AND DATA ANALYSIS METHODS

In this chapter a description is made of the experimental apparatus and the procedures used to make raw measurements. The techniques developed to process these measurements in order to yield meaningful data for comparison with KIVA-3V are also explained.

3.1 The Test Volume

The primary component of the experimental system is the UBCRICM because it contains the test volume. The UBCRICM was designed and assembled by Dohring (1986). This machine generates the physical phenomenon that KIVA-3V output is compared against: turbulent squish flow in a combustion chamber. The initial flow field in the UBCRICM is quiescent and it can be easily and accurately matched in KIVA-3V.

The UBCRICM essentially comprises a single cylinder whose piston follows the motion of the gudgeon pin on a connecting rod and crank linkage. The crank is turned by the linear motion of a rack that meshes with a crankshaft pinion. The rack is driven from one end by a pneumatic cylinder whilst the other end is connected to an hydraulic ram. The ram pushes hydraulic fluid through an orifice at a constant flow rate, thus ensuring a constant rack speed. The UBCRICM is activated by first pressurising the pneumatic cylinder, then suddenly opening the constant flow orifice with a solenoid valve. A braking pin is attached to the end of the ram to restrict flow through the orifice as the rack completes its stroke. In this way, damage by collision between the ram and the orifice is prevented. The major components of the UBCRICM (including the crankcase, the pneumatic cylinder and the hydraulic ram) are mounted on a raised platform that is heavy to damp vibrations during operation. A schematic diagram of the UBCRICM is shown in figure 3.1, while a list of key specifications is shown in table 3.1.

The rack of the UBCRICM is long enough to rotate the crank by 360 degrees. This enables both the intake and compression strokes to be executed. Valve actuation during the inlet stroke is facilitated by a pin (which is restricted to move in a direction normal to the rack) that follows a groove machined in the side of the rack. The pin is analogous to a pushrod of a reciprocating IC
engine, with the groove in lieu of a cam. In this research, the valve train was dismantled because only the compression stroke was of interest. The compression stroke was set-up by moving the rack carefully with the pneumatic cylinder until BDC was reached. A small port drilled through the cylinder wall just above the piston crown allowed air to enter the cylinder at BDC. This ensured ambient cylinder pressure at the start of the compression stroke. The port was also used to introduce microscopic particles (for velocity measurements) before it was plugged for executing the compression stroke.

3.1.1 Modifications made to the UBCRICM

The most significant modifications the UBCRICM underwent were on the cylinder, the cylinder head and the piston. The objective was to maximise the optical access of the combustion chamber whilst ensuring structural stability and sealing. All three components were made of transparent acrylic for visual access. While the piston crown and the cylinder head changed in shape depending on the test performed, the cylinder was permanent. Figure 3.2 shows the drawing for the cylinder. The piston shape changed depending on the combustion chamber tested.

Three different combustion chambers were used for this KIVA-3V validity study:

a) The ‘Bowl-in-Piston’ combustion chamber. This chamber demonstrates the simplest form of squish generation and it doesn’t have squish grooves. It is intended to test the basic squish prediction ability of KIVA. An illustration of this chamber is shown in figure 5.1.

b) The ‘Squish-Jet 1’ combustion chamber. This chamber has four equi-spaced squish grooves in a fence surrounding the lip of the piston bowl. This chamber is illustrated in figure 5.2.

c) The ‘Squish-Jet 2’ combustion chamber. This chamber also has four equi-spaced squish grooves; however, the grooves are narrower and the fence is 40% shorter than for Squish-Jet 1. This chamber is illustrated in figure 5.3.

In this document, frequent references are made to common geometric features of the combustion chambers. The following list is meant to familiarise the reader with these features:

- ‘Squish-fence’—The fence surrounding the lip of the bowl in Squish-Jet chambers.
• 'Squish-grooves'—The slots found in the squish-fence that issue jets of squished gas.
• 'Bowl area'—The area at the base of the piston bowl.
• 'Squish area'—The cylinder bore area minus the bowl area.
• 'Squish ratio'—The ratio of squish area to bowl area (≈75% for all chambers in this study).
• 'Raised bowl volume'—The cylindrical volume inside the combustion chamber that is directly above the bowl base.
• 'Squish Volume'—The volume of the combustion chamber minus the raised bowl volume.
• 'Squish flow area'—The area that is formed by the intersection of the squish volume and the raised bowl volume. This is the area through which the squished gases flow when exiting the squish volume.

Unless described otherwise, the squish velocity for the Squish-Jet chambers is the ensemble mean velocity of gas directly exiting the midline of the squish-grooves. For the Bowl-in-Piston, it is the ensemble mean velocity of gas through the squish flow area.

Instead of having to remove the gudgeon pin to replace the piston each time another chamber is tested, a permanent piston base was used into which the desired piston crown could be screwed. Another advantage of this design is that spacers can be installed between the piston base and the crown to make compression ratio adjustments. Figure 3.3 shows the drawing for the piston base. Figures 3.4, 3.5 and 3.6 show the piston crown dimensions for the Bowl-in-Piston, Squish-Jet 1 and Squish-Jet 2 chambers respectively. For each of the piston crowns, the surfaces that were parallel to the cylinder head were painted black. This reduced diffuse reflections that would otherwise have introduced noise to the optical velocity measurement techniques employed.

All three combustion chambers had a squish clearance of 2mm, a compression ratio of 9.1:1 and a squish ratio of 75%.

One of the techniques used to measure the velocity in the combustion chamber was PIV (Particle Image Velocimetry). The technique is explained later in this chapter. A requirement of the method was to pass a thin laser sheet parallel to and less than a millimetre in from the cylinder head. In order to achieve the proximity of the laser sheet to the head, the cylinder and head were sandwiched between the crankcase and a steel retainer plate. A shallow annular recess was also machined into one edge of the disk shaped head so that it snugly fit into the cylinder. This fit
restricted lateral movement between the head and the cylinder. For the same reason, a similar recess was made on the other side of the head adjoining the retainer plate. Figure 3.7 shows the design drawing of the acrylic cylinder head while the head retainer plate is shown in figure 3.8. The retainer plate was fastened onto four steel studs protruding from the crankcase. The retainer plate had to be a fixed distance from the crankcase to ensure a consistent compression ratio. Furthermore, in order to prevent damage and to prevent leakage at the cylinder head, the plate had to be parallel to the crankcase surface supporting the cylinder. These geometric constraints were satisfied by sliding four steel collars of identical length onto the studs and by installing a stiff rubber gasket between the base of the cylinder and the crankcase. Figure 3.9 shows a plan view of the cylinder assembly.

The other velocity measurement technique was LDV (Laser Doppler Velocimetry). The technique is also explained later in this chapter. In this method, two thin and intense laser beams passed through the clear cylinder head and converged on a tiny point (approximately 100 microns wide) where the measurement was made. The point is known as the 'probe volume'. In order to minimise the deviation and attenuation of the beams, an optical grade flat quartz window was used for the cylinder head instead of the acrylic head. Since it was made of quartz, the disk shaped window (measuring 50.4 mm thick by 114.3 mm diameter) could not be machined to the shape of the acrylic head it replaced, so a PVC case was made to secure the quartz in place over the cylinder. To prevent chipping, the window’s perimeter was lined with 1/8” (3.18 mm) rubber sheet before it was installed into the case. Figure 3.10 shows the drawing for the window case.

The piston and the head/cylinder joint were fitted with O-rings to ensure adequate sealing of the combustion chamber. The combustion chamber was pressurised with helium to 300 psi absolute (2.07 MPa absolute) and leak tested with soapy water, yet no leaks were detected. The test pressure was 36% greater than the measured peak cylinder pressure during the compression stroke.

Another important modification made to the UBCRICM concerned its vibration. A high shutter speed digital photograph of the cylinder head when the piston was at 20 degrees BTDC revealed a displacement of 2mm (from its rest position) in a direction normal to the head. A similar displacement was observed in the direction parallel to the rack. Given that the squish clearance has a similar thickness, this movement was clearly unacceptable for optical velocity
measurements because the light sources were mounted on the ground. As a result, the base of the UBCRICM was braced with cross-members to prevent sway. The machine was then bolted solidly to the ground to stop it from sliding. Subsequent photographs showed undetectable displacements.

3.2 The PIV Measurement System

The majority of the velocity measurements inside the combustion chamber were made with a PIV system. The system was developed from separate components and it was specifically built for its application to the UBCRICM. PIV was chosen as a measurement method because it is capable of providing global velocity plots of planes inside the cylinder. Subsequently, velocity structures that are important to the development of in-cylinder flow can be observed. For comparison, similar velocity plots can be extracted from KIVA-3V output. Although PIV can be used to determine ensemble mean velocities, the limited spatial resolution and data rate makes PIV inadequate for finding turbulence intensities. As a result, LDV was used as a complementary flow measurement technique. LDV is described in section 3.3.

The PIV system worked by pulsing a thin laser sheet twice through the transparent combustion chamber. The sheet, which was approximately 0.5mm thick, was aligned parallel to the cylinder head and the pulse separation time was typically below 200 microseconds. The double pulsing sequence was initiated by the crank angle trigger that was set on an optical encoder mounted on the crankshaft of the UBCRICM. The crank trigger would send a signal to a pulse generator that would in turn send timed pulses to the laser unit. Regardless of crank angle, the sheet was placed halfway between the piston crown and the head for the Bowl-in-Piston chamber, whilst for both Squish-Jet chambers the sheet always bisected the squish fence. Figures 5.1 to 5.3 show these laser sheet positions. Microscopic particles that were seeded into the combustion chamber and that followed the local flow inside it were illuminated each time the laser pulsed. For each of the two laser pulses, a synchronised digital camera recorded an image of the illuminated particles. The resulting image pair was later analysed using cross-correlation software, which tracks the displacements of small groups of particles. In the cross-correlation process, the pattern formed by each particle group serves as the identifying feature with which their movements are tracked. The outcome is a velocity field map of the flow parallel to and inside the laser sheet. Figure 3.11
shows an overview of the PIV system, while table 3.2 lists the hardware and software specifications of the system. Figure 3.12 shows a typical image pair taken with the PIV system and the resulting velocity plot after processing the pair.

The following list is a summary of the conditions for the PIV experiments:
- The effective crank speed was fixed at 800 rev/min.
- The crank angles tested for each of the three chamber types were 30, 25, 20, 15, 10, 5 and 0 degrees BTDC. Approximately 10 tests were performed for each crank angle.
- The PIV observation planes for each chamber type are specified in figures 5.1 to 5.3.

3.2.1 The Pulsed Laser

A New Wave Gemini twin pulsed laser designed for PIV applications was used as the illumination source for the laser light sheet. It has two pulsed Nd:YAG infrared laser heads that are mounted on a single baseplate. The 1064 nm wavelength beam coming from each laser head entered a second harmonic generator to produce visible green light (of wavelength 532 nm). Dichroic mirrors separated the visible from the residual infrared light and directed the beam to the laser sheet optics. The delay between the two pulses could be varied using an external programmable trigger source (a Berkeley Nucleonics 8 channel pulse generator). The pulses from this source were triggered by the crank angle sensor, which was set at the desired angle before the experiment. The laser pulse separation time was verified by pointing the laser output to an Aluminum surface, then orienting a photodiode to detect the diffuse reflection from the Aluminum. The two voltage pulses from the photodiode that corresponded to the laser pulses were observed on an oscilloscope and their separation time was less than 1% of the period set on the pulse generator.

Each laser head was capable of delivering 120 mJ per pulse. With a pulse width of approximately 5 ns, this equates to a mean light power of 24 MW during illumination. At typical in-cylinder velocities of 10 m/s, the 5 ns pulse is short enough to freeze the particle locations to within 0.05 μm. This represents less than 2% of a particle diameter. Full pulse energy was not necessary for the experiments; instead, the energy was adjusted for each laser head to provide
ample contrast in each of the two images. The contrast ensured that particles were distinguishable for PIV analysis.

A disadvantage of the independent laser heads was the need to maintain the co-alignment of the two separate laser beams. If the laser beams were not aligned to each other, the thin laser sheets crossing the test volume would not overlap. This meant that particle groups from the first image would not correlate with those from the second to yield meaningful particle displacements.

3.2.2 Beam Delivery and Light Sheet Optics

The purpose of the optical assembly was to form the thin laser sheet necessary for the PIV measurements and to provide a means with which the sheet could be easily translated to a desired plane in the combustion chamber.

The translation was achieved by mounting a high intensity laser mirror on a traverse that moved parallel to the cylinder axis. The laser was also oriented to point into the mirror in a direction parallel to the cylinder axis and the mirror angle was adjusted to deflect the incident laser beam by 90 degrees. The mirror was placed at the same height as the cylinder axis and the distance between the mirror and the cylinder (approximately 1 m) was minimised to reduce the directional sensitivity of the laser sheet to any pointing instabilities in the laser. The traverse was mounted exclusively on a steel tower secured to the ground, to reduce mirror vibration that would cause unwanted translation of the light sheet.

The laser sheet was formed by two lenses that were mounted on the traverse between the mirror and the test volume. The pulsed laser output was a collimated beam with a nominal diameter, \( d \), of 4.5 mm. After leaving the mirror, the laser beam entered a positive cylindrical lens that caused the beam to converge on a vertical focal line 200 mm (the focal length, \( f_1 \), of the cylindrical lens) away. The transformed beam then entered a negative spherical lens 25 mm (its focal length, \( f_2 \)) downstream from the focal line.
The final result was a vertically diverging light sheet of thickness, \( W \), and divergence half angle, \( \theta_d \), where:

\[
W = \left| d \frac{f_2}{f_1} \right| = 0.56 \text{ mm} \quad \text{Equation 3.1}
\]

and,

\[
\theta_d = \tan^{-1}\left( \frac{d}{2f_2} \right) = 5.1^\circ \quad \text{Equation 3.2}
\]

Essentially, the lens pair contracts and then collimates the laser beam in the horizontal direction while it diverges the beam in the vertical direction. Since the cylinder axis was horizontal, the light sheet illuminated a plane parallel to the cylinder head. Figure 3.13 shows an optical ray diagram for the lens pair while a diagram of the traverse and optics in relation to the UBCRICM is shown in figure 3.14.

In order to reduce stray illumination of the laser sheet caused by fine scratches in the cylinder wall, it was imperative to periodically polish the cylinder.

3.2.3 Particles and Flow Seeding

The microscopic particles that were used to seed the flow inside the combustion chambers were a critical component to the PIV system and they needed to satisfy a host of requirements in order to conduct reliable measurements. The following is a list of the more important requirements.

The particles should:

1) be able to adequately follow the local flow.
2) scatter an adequate amount of light toward the camera.
3) be homogeneously distributed in the test volume.
4) be distributed at a density that maximises the validity of the results.
5) not have a tendency to foul or abrade the inner walls of the test volume.
6) not have a tendency to clump up.
7) be able to withstand the experimental conditions.
8) be chemically inert during experiments.

Flow-Following Ability

Before the process of PIV particle selection began, it was worthwhile estimating the type of flow the particles were meant to follow.

A numerical simulation, based on balancing the mass flow in the squish volume for the Bowl-in-Piston chamber, was created to estimate the magnitude of squish velocities during the compression process. The key assumptions in the simple analysis were that the pressure and density were uniform in the combustion chamber at any one time. The uniform pressure assumption is plausible as the gas flow in non-reacting combustion chambers is subsonic; however, the uniform density assumption is questionable, because temperatures may vary in the chamber due to non-uniform heat transfer. Nevertheless, the simplistic assumptions have been used before with reported success (Heywood, 1988). Furthermore, the current objective is only to estimate the velocity rather than to perform a detailed simulation. The following shows the theoretical analysis behind the simple simulation.

Consider the motion of a piston (with a centred and symmetrical bowl) during the compression stroke. The process is shown in figure 3.15.

As shown in figure 3.16, the combustion chamber can be divided into two separate volumes. One can find the squish velocity by first applying a mass balance to zone 1 in figure 3.16:

\[
V_1(\frac{d\rho_1}{dt}) = v_pA_1\rho_1 - v_sA_s\rho_2
\]

Equation 3.3

Similarly a mass balance applied to zone 2 yields:

\[
V_2(\frac{d\rho_2}{dt}) = v_pA_2\rho_2 + v_sA_s\rho_2
\]

Equation 3.4
Where:

- $V_1$ and $V_2$ are the volumes of zone 1 and zone 2 respectively.
- $\rho_1$ and $\rho_2$ are the densities in zone 1 and zone 2 respectively.
- $t$ denotes time.
- $v_p$ is the piston speed.
- $A_1$ is the area of the squish surface on the piston crown.
- $A_2$ is the area bounded by the rim of the piston bowl.
- $v_s$ is the squish velocity (radially inwards).
- $A_s$ is the area of intersection between zone 1 and zone 2.

Since the squish ratio is always 75% in this study, $A_2/A_1$ equals 1/3.

If the assumption is made that the density in the cylinder is uniform, division of equation 3.4 by equation 3.3 yields:

\[
\frac{v_s}{v_p} = \frac{\frac{V_2 - A_2}{V_1}}{A_1} = \frac{\frac{V_2 - 1/3}{V_1}}{A_1} = \frac{A_s}{A_1} \left(1 + \frac{V_2}{V_1}\right)
\]

Equation 3.5

Since $A_1$ is a constant and $V_2, V_1, v_p$ and $A_s$ are known functions of crank angle, the squish velocity can be determined for any given crank angle.

Figure 3.17 shows the predicted squish velocity during the compression process for a crank speed of 800 rev/min. It is evident in the figure that the maximum bulk squish velocity predicted by the simulation was 9.2 m/s at a crank angle of 12 degrees BTDC. One would also expect this to be the largest velocity in all three combustion chambers, because the squish volume and flow area are smallest for the Bowl-in-Piston chamber. Assuming zero velocity at the cylinder wall, the mean acceleration, $a_m$, a parcel of gas would undergo in traversing the squish zone is 2120 m/s$^2$. A more realistic representation of the parcel’s acceleration should also include a term for the turbulent velocity fluctuations. Combustion chamber observations reported by Tabaczynski (1976) show an upper bound on turbulence frequency (corresponding to the Kolmogorov frequency, $f_k$) of 10 kHz. Furthermore, Anetor (1994) reports tangential and radial turbulence...
intensities \((u_{rms})\) as high as the mean piston speed, which in the present case is 2.66 m/s. In the extreme case, the fluctuating velocity component, \(v_{\text{fluc}}\), can therefore be modelled as:

\[
v_{\text{fluc}} = u_{\text{max}} \sin(2\pi f_k t) = u_{\text{max}} \sin(\omega_k t)
\]

Equation 3.6

where the angular frequency, \(\omega_k = 2\pi f_k = 62.8 \times 10^3\) rad/s

and \(u_{\text{max}} = u_{\text{rms}} \times 2^{0.5} = 3.76\) m/s

By differentiation with respect to time, the acceleration of the fluctuating term is:

\[
a_{\text{fluc}} = u_{\text{max}} \omega_k \cos(\omega_k t)
\]

Equation 3.7

Therefore in the extreme case, the total acceleration of the gas parcel was modelled as:

\[
a_{a} = a_{m} + a_{\text{fluc}} = a_{m} + u_{\text{max}} \omega_k \cos(\omega_k t)
\]

\[= 2120 + 2.37 \times 10^5 \cos(62.8 \times 10^3 t)\]

Equation 3.8

The next step was to determine the properties (density and aerodynamic diameter) a particle must have in order to adequately follow the gas parcel. This was achieved by modelling the forces the particle is subjected to and dividing by its mass to yield the acceleration. The particle displacement was then found by twice integrating the acceleration with respect to time. A particle was deemed to be adequately following a parcel of gas surrounding it, if the particle’s displacement was between 99% and 101% of the parcel’s displacement.

An isolated and inert solid or liquid particle in a gas can be subjected to two types of forces:

- Aerodynamic forces, which arise from the particle’s interaction with air.
- External non-contact forces, such as gravitational, magnetic or electrostatic forces.

Dust particles typically used for PIV applications in air have small settling velocities relative to the air velocities they’re meant to follow, indicating a negligible gravitational influence for fast flows. Furthermore, charge reduction techniques (like the use of grounded copper tubing to guide
the particles into the test volume) and ensuring that particles are non-magnetic make electrostatic and magnetic forces negligible too.

Thus, as shown in the following force balance equation, only aerodynamic forces are considered.

\[
\frac{\pi d_p^3 \rho_p}{6} \frac{dv_p}{dt} = 3\pi \mu d_p \nu_{rel} + \frac{\pi d_p^3 \rho_f}{6} \frac{dv_f}{dt}
\]

Equation 3.9

Where \( \nu_r \) (which is found from equation 3.8) is the velocity of the fluid surrounding the particle

\( \nu_p \) is the velocity of the particle

\( \nu_{rel} = \nu_f - \nu_p \)

\( d_p \) is the particle diameter

\( \mu \) is the viscosity of the surrounding fluid

\( \rho_f \) is the fluid density

\( \rho_p \) is the density of the particle

\( t \) denotes time

The term on the left side of equation 3.9 is the rate of change of the particle’s momentum, which by Newton’s second law is equal to the sum of the forces the particle is subjected to. The first term on the right side of the equation is the viscous drag according to Stokes’ law. The law applies for particle Reynolds numbers (Re\( p \)) smaller than unity, where:

\[
Re_p = \frac{\nu_{rel} d_p \rho_f}{\mu}
\]

Equation 3.10

The condition is satisfied for microscopic particles and low relative velocities. The second term on the right of equation 3.9 is the force on the particle from the pressure gradient developed when the fluid accelerates.

Equation 3.9 was numerically integrated on a computer spreadsheet. The resultant displacement histories for three different particle sizes (2 \( \mu \)m, 4 \( \mu \)m and 10 \( \mu \)m), each having a density of 1000 kg/m\(^3\), are shown in figure 3.18. The displacement history for the air that influences the particles
is also shown on the same set of axes. The results for the same three particle sizes, but with four times the density (4000 kg/m\(^3\)) are shown in figure 3.19. It should be noted that for a particle closely following the flow, the last term in equation 3.9 is negligible because of the large density of the particle with respect to the air.

It is evident from the graphs that particles with small densities and size follow the predicted flow more faithfully. For example, the smallest and lightest particle simulated (2 micron diameter with density of 1000 kg/m\(^3\)) not only follows the mean acceleration of the fluid, but also follows the high frequency fluctuations. Since LDV can resolve such fluctuations, the particle is suitable for LDV. The particle was calculated to have a gravitational settling velocity in air (at STP) of 1.27x10\(^{-4}\) m/s. The suitability of this particle for 10 kHz flows is also documented by other researchers, including Melling (1997). Although it is unresponsive to the 10 kHz fluctuations, the 4 micron particle follows the mean acceleration. This particle has a settling velocity of 5.06x10\(^{-4}\) m/s. Since PIV cannot resolve much finer flow detail than the mean acceleration, the 4 micron particle is adequate for PIV. A more detailed explanation of PIV limitations follows.

In order to detect the displacements of faithfully tracking particles like the 2 micron particles, the PIV system must have a high enough spatial and temporal resolution to discern the microscales associated with the velocity fluctuations. Time measurement in most velocimetry systems can be in the order of nanoseconds, so the required temporal resolution can be easily achieved. Spatial resolution is adequate with LDV, because the tiny probe volume is typically about 100 microns wide, which is sufficiently smaller than the 0.3 mm microscale cited by Tabaczynski (1976). Unfortunately, the spatial resolution for PIV systems with 1000x1000 pixel cameras is inadequate, even for a quadrant of a typical combustion chamber. Since PIV cannot resolve the high frequency flow component, use of 'high fidelity' particles was considered redundant. As a result, the large particles that tracked the mean flow were considered superior for PIV because of their ability to scatter more light. For this reason, 4 micron particles with a specific gravity of 1 were sought for PIV experiments to determine mean velocity components. Even if the PIV resolution was high enough to warrant the use of high fidelity particles, the number of PIV experiments (per crank angle) needed to find rms velocity fluctuations with a reasonable confidence interval would have been impractical. Furthermore, the particle concentration needed to resolve microscales may have been high enough to adversely affect the image quality. Nevertheless, based on ideal particle concentrations (discussed later in this chapter), such a high
concentration was not expected to significantly increase the effective gas density. If found, larger particles than 4 microns were preferable for PIV, so long as their material density was accordingly smaller (for a fixed settling velocity, $\rho_p \propto 1/d_p^2$). The high fidelity 2 micron particles were sought for turbulence intensity measurements with LDV not only because of the method’s spatial resolution but also because of its abundant output.

**Light Scattering Considerations**

The light intensity scattered by particles typically used for PIV is a complicated function of many variables including the incident light intensity and frequency, the observation position, the refractive index and the size of the particle. The relationship was originally derived by Gustav Mie using Maxwell's equations for electromagnetic radiation.

Mie scattering applies to spherical particles whose diameters, $d_p$, have the same order of magnitude as the incident light’s wavelength, $\lambda$. Mie scattering theory shows that the average intensity of the light scattered by a particle is roughly proportional to the square of its diameter (van de Hulst, 1981). Even though larger particles are desirable for photographic detection, the size limit was already set (for a specific gravity of unity) by flow following requirements. Therefore, the particles were to preferably have a monodisperse size distribution close to the maximum allowable size.

Camera visibility of the chosen PIV particles was to be ensured by progressively increasing the pulse energy of the laser to an appropriate level. If the pulse energy were to reach a high enough level where stray light should become problematic, corrective measures were to be employed to improve the light contrast (eg. Painting surfaces black or polishing scratches out).

**Particle Concentration in the Test Volume**

An important consideration used to select the PIV particles was the ability to control the particle concentration in the test volume. A review of the literature has shown that PIV work on combustion chambers is done much more frequently in reciprocating engines than in rapid
compression machines. Commonly available constant flow-rate particle seeders are suited to the continuous air consumption of reciprocating engines, because the ratio of particles delivered to the air admitted is kept at a desired level. These types of seeders, however, are not suitable for the momentary event executed by the UBCRICM. Instead, a method was devised to 'prime' the cylinder with a metered number of dispersed particles before 'firing' (executing the compression stroke). The method entailed squirting dry air into a fixed number of solid seed particles resting inside a flask. Dry air was used to avoid condensation on the cylinder walls and to prevent possible agglomeration of the seed particles. The source of dry air was from a commercially available pressurised can used for dusting sensitive equipment. Liquid particles were avoided to prevent evaporation during the compression stroke. Using electrically grounded Aluminum tubing, the agitated mixture would exit the flask and flow through a fine mesh. After the mesh the aerosol would flow into the cylinder through its small inlet port.

The grounded tube was meant to prevent charge build-up, while the mesh filter was intended to break up any potential clumped particles. The mesh was called a Rigimesh class K medium and it was made from sintered woven wire by the Pall company. The mesh was also meant to block particles larger than 3 microns. In order to ensure a homogeneous distribution of seed particles, a nozzle was used to issue the particles deep into the cylinder. The nozzle was made by perforating a 75 mm long tube and blocking one end. The OD of the tube was smaller than the port diameter to provide an exit for the displaced cylinder air while the seed particles were being injected. Figure 3.20 shows a schematic diagram of the particle seeding system that was devised.

The number of particles initially in the flask before mixing depends on the desired concentration inside the combustion chamber. Keane and Adrian (1992) performed extensive PIV simulations to observe the effect of the number of particles per interrogation region ($N_{int}$) on the validity of the velocity measurements. The interrogation region is a space chosen in the first PIV image containing a particle pattern for which a match is sought in the second image. The investigation by Keane et al. revealed that $N_{int}$ should be at least equal to 10 for valid velocity detection. Up until that value, more particles per window impart more uniqueness to the particle pattern, and the probability of finding that pattern elsewhere from its true displacement diminishes.

An upper limit for $N_{int}$ could be designated for the point when the effective density of the particle/air mixture is high enough to affect the flow being investigated (e.g., when the mixture
density is 2% greater than that for plain air). Keane et al. did not specify the optimum value for $N_{int}$, because it depends on experimental conditions. For example, experimental conditions can influence the degree of out-of-plane motion, which is a phenomenon that reduces the validity of velocity measurements due to the disappearance (and extraneous materialisation) of particles in the second image.

Given a typical interrogation square of 16 x 16 pixels, a light sheet thickness of 0.5mm and a conversion factor of 15 pixels for every millimetre, the image density of 10 particles per region corresponds to a particle density in the chamber of $2.0 \times 10^{10}$ m$^{-3}$. In order to achieve this density at TDC, the mass of particles needed is only a fraction of a milligram. Nevertheless, flow-back and filtering losses would require an increased initial mass and it was reasoned that the best way to achieve the optimum concentration was to progressively increase the particle mass from the ideal value until the images yielded consistently valid vectors.

Having selected the phase and the approximate size and density of a suitable particle for the PIV experiments in the UBCRICM, the options remaining for particle selection were narrowed down considerably. A number of micron sized metal oxides were evaluated, including TiO$_2$, MgO and ZrO$_2$, but they tended to clump and they easily fouled the cylinder wall. Eventually nylon particles were found to satisfy all the criteria outlined at the beginning of this section. The particles, which were supplied by TSI Inc., had a nominal diameter of 4 microns and a density of around 1100 kg/m$^3$. Approximately 2 milligrams of the particles were used per PIV test. As shown in the microscope image in figure 3.21, the particles were also spherical and they had a relatively narrow size distribution.

3.2.4 Camera and Frame Grabber

The light side-scattered from the particles illuminated by the laser sheet was recorded by a Roper Scientific ES 1.0 digital camera. A list of specifications for this camera is shown in table 3.3. The monochrome camera was fitted with a 50 mm zoom lens, which was locked in the maximum zoom position to maximise the image resolution. The image that was focussed on the CCD at this zoom setting covered a slightly greater area than one quadrant of the illuminated chamber. The camera, which was pointed at a normal direction toward the cylinder head, was mounted on
a 3-axis traverse approximately 400 mm away from the laser sheet. The camera was focused on
the lower left quadrant, because this region tended to have the clearest illumination and the most
uniform particle concentration. Sample particle images recorded by the ES1.0 are shown in
figures 5.4 and 5.5. The images are 1008 pixel widths across and 1018 pixel widths tall, with
each square pixel having 256 possible greyscale levels. Each millimetre in the photographed
light sheet corresponds to 15 pixel widths.

The camera was set to dual exposure mode and it was triggered by the frame grabber that resided
in a dedicated PC. The exposure mode and exposure times were set by commands sent in RS-232
protocol from a serial port of the PC to the camera. The frame grabber (Matrox Meteor II digital)
was triggered by a pulse generator that synchronised the camera with the laser pulses. When the
frame grabber received the trigger from the pulse generator, it executed two functions:

1) It sent a trigger to the camera to capture the PIV image pair.
2) It initiated custom computer code (written in MIL script) to grab the two separate images
   from the camera.

A digitizer configuration format (DCF) file that contained information for interpreting the camera
output was created in the PC before the experiments began. The delay time between the trigger
input from the pulse generator and the trigger output to the camera was also set in this DCF file.
This delay was set to the minimum possible value (approximately 20 μs).

3.2.5 Camera and Laser Synchronisation

Since velocity data were sought at particular crank angles during the compression stroke, the
crank angle trigger was used as a time datum and it initiated a synchronised activation sequence
for the lasers and camera. The activation sequence came in the form of five carefully timed
pulses (5V) sent from the Berkeley Nucleonics pulse generator. Two of the pulses were delivered
to laser 1, two others were sent to laser 2 and the fifth pulse went to the frame grabber. The only
input to the pulse generator was the TTL trigger signal from the crank trigger. Each laser
required two pulses. The first pulse was for the Q-switch while the second pulse actually fired
the laser. The delay time between Q-switching and firing for each laser was set to 150 μs. This
setting provided ample laser pulse energy for image recording. With the zero datum time at the crank trigger event, the Q-switch pulse and the fire pulse for laser 1 were set to 0 s and 150 µs respectively. Typically, the laser pulse separation time between the lasers was 200 µs, so the Q-switch pulse and the fire pulse for laser 2 were set to 200 µs and 350 µs respectively. The laser pulse separation period was chosen such that adequate particle movement occurred between the two subsequent PIV images. Excessively large separation times allow particle groups to lose their unique patterns, thus lessening the chance of calculating the true particle displacement. Conversely, very short separation times produce particle displacements approaching the spatial resolution of the images, thus making the particles appear ‘frozen’—clearly a condition from which displacements cannot be determined. The rule of thumb used for selecting the laser pulse separation was that particles in relatively high speeds of the flow should move by approximately 20% of the interrogation window size between images. The out-of-plane particle motion was not excessive for the chosen separation time. The pulse for the frame grabber was dispatched at the instant (0 s) the pulse generator received the crank trigger. The widths of the pulses were set to satisfy the requirements of the devices they fed.

At a crank speed of 800 rev/min, the 150 µs delay between the crank trigger and laser 1 firing meant that the first laser sheet pulsed 0.72 crank degrees after the trigger. This delay was compensated for by advancing the crank trigger approximately 0.7 degrees more than the desired measurement crank angle. This action ensured that the laser sheet pulsed at the desired crank angle. The compensation technique was verified by pointing the camera to the crank angle indicator and firing the UBCRICM. The image recorded consistently showed the crank indicator aligned with the desired angle.

3.2.6 Recording Procedure

In order to minimise the chance of equipment damage and null experiments, a systematic procedure for PIV measurements was devised. This was particularly important given the investment of time required to polish the cylinder and to adjust the laser and optics. The procedure is now presented:
- Ensure lasers are warmed up.
- Turn compressor on.
- Ensure that lasers and crank trigger is connected to pulse generator.
- Turn pulse generator on.
- Turn PC on.
- Retract piston to BDC.
- Clean and reassemble cylinder.
- Move optics traverse so that laser sheet misses cylinder and strikes a black board.
- Connect camera and focus it on black board.
- Equalise power for both laser sheets and ensure that they are co-aligned.
- Move laser sheets to desired plane in cylinder.
- Focus camera.
- Equalise laser power (by comparing brightness of cylinder wall in both images).
- Load 4 micron nylon particles in mixing flask
- Insert injection tube into the cylinder's particle port.
- Activate external trigger on pulse generator.
- Ensure that air pressure in compressor has reached level needed for 800 rev/min.
- Activate image-grab software.
- Do mock trigger and image capture by flicking the crank trigger's power supply on and off.
- Check crank position.
- Check UBCRICM valve positions are set for firing.
- Check that the UBCRICM solenoid switch is powered.
- Check UBCRICM head tightness.
- Turn out lights.
- Squirt particles into cylinder and wait for flow to subside.
- Activate image-grab software and fire UBCRICM.

3.2.7 PIV Processing

As mentioned in section 3.2.3, PIV processing of an image pair to yield velocity data entails isolating a portion of the first image (called the interrogation region, \( R_1 \)) and trying to find a match between the particle pattern inside \( R_1 \) with a particle pattern in a similar location in the
second image. Given that a valid match is made, the displacement of the matched region in image 2 relative to R₁ is a direct measure of the true displacement of the particle group. Knowledge of the time elapsed between the two images can therefore be used to determine the average velocity of the particle group in that time. If the image separation time and R₁ are adequately small and the spatial resolution is high enough, the determined velocity can be considered to be the instantaneous velocity of the particle group. Moreover, if the particles faithfully follow the local flow, this velocity also is the fluid’s instantaneous velocity at R₁. This velocity determination method which is the essence of PIV processing, can be repeated on a systematic collection of interrogation regions covering all of image 1. The result is a 2-D global velocity map of the photographed plane of flow.

The method most often used in 2-image PIV processing to find a match for R₁ in image 2 is cross-correlation. In effect, the cross-correlation between R₁ and an identically sized arbitrary region in image 2 (R₂) is a ‘match indicator’ between R₁ and R₂. Higher cross-correlation values indicate greater degrees of matching. Assuming cartesian coordinates and a square interrogation region (L x L pixels), the cross-correlation between R₁ and R₂ can be represented by:

\[ C(x₁, y₁, d_x, d_y) \]

where \( x₁ \) and \( y₁ \) describe the centroid position of R₁ and \( d_x \) and \( d_y \) are the x and y displacements of R₂ relative to R₁.

The cross-correlation of R₂ with respect to R₁ can be defined as the sum of the product of the brightness levels of corresponding pixels in R₁ and R₂:

\[ C(x₁, y₁, d_x, d_y) = \sum_{y=(y₁-L/2)}^{y=(y₁+L/2)} \sum_{x=(x₁-L/2)}^{x=(x₁+L/2)} B₁(x, y) \ast B₂(x+d_x, y+d_y) \]

Where B denotes the brightness level and the subscripts 1 and 2 refer to the first and second images respectively.

The brightness level of a pixel is typically a binary value—1 if a bright spot exists and 0 otherwise. When a match occurs, the bright pixels overlap and a large contribution is made to the cross-correlation. Conversely, if the bright pixels don’t overlap there’s no contribution.
Figure 3.22 shows an example of computing the cross-correlation. The process of selecting a pixel in $R_1$ and multiplying the corresponding pixel in $R_2$ is illustrated. Repeating this process for all pixels in $R_1$ and summing the products yields the cross-correlation between $R_1$ and $R_2$.

Figure 3.23 shows the result of cross-correlating $R_1$ with a range of different $R_2$ regions that are displaced $d_x$, $d_y$ relative to $R_1$. The result is shown as a surface plot where the $x$ and $y$ axes are used to define the displacement of $R_2$ relative to $R_1$. The height of the surface at a chosen displacement is the cross-correlation value for that displacement. A distinct peak in the cross-correlation surface indicates the displacement of the particle group in $R_1$ during the time elapsed between image 1 and image 2. When the cross-correlation surface is interpolated by quadratic or Gaussian functions, sub-pixel accuracy in locating the displacement is achieved. A commonly accepted displacement uncertainty is 0.1 pixels (Keane and Adrian, 1992; Westerweel et al., 1996).

Computer code was written that cross-correlates the image pairs from the experiments to yield numerical velocity data. The raw images had to be converted to ASCII-PGM graphical format with separate software before the cross-correlation processing. The numerical output from the cross-correlation code was then used as input to data plotting software for visualising the velocity vectors. Commercially available software by Dantec called Flowmanager replaced this procedure because the three separate stages were combined. In addition, many useful features such as adaptive correlation and masking made this software a convincing choice. Masking is the term used for ignoring a selected area in the PIV images, such as the region outside the cylinder. Adaptive correlation is an iterative form of cross-correlation in which the interrogation window progressively shrinks, thus providing more velocity vectors per unit area of the images. Each time the window is shrunk, the velocity information from the larger window is used to place a refined search boundary for the shrunk window. Even though the shrinking process reduces $N_{int}$ (the number of particles per interrogation region), the information transfer between stages eases the criterion that $N_{int}$ should be at least 10 for a valid velocity determination. Consequently, this feature is extremely useful for velocity determination in portions of the image pair where the particle distribution is relatively scant.
The image pair for each PIV test was processed with the Flowmanager software, the essential output of which was a two dimensional array of vectors that described the velocity at the centroids of the interrogation windows covering the observation plane. This amounted to a vector array of 83 rows (x positions) by 84 columns (y positions): a total of 6,972 evenly spaced velocity measurement points per PIV test.

Typically, for each crank angle tested on the Squish-Jet chambers about 10 PIV tests were conducted. The ensemble mean velocity vector at a point in the measurement plane was calculated by averaging the velocities at the same point from all the tests. The 10 tests were enough to get a typical 90% confidence interval of 0.3 m/s for the mean velocity. It is emphasised that the averaging process was applied to the *same point in the combustion chamber*, which is not necessarily the same point in the image frame. This is because the camera was sometimes readjusted or the piston would rotate slightly due to the design of the UBCRICKM gudgeon pin bearing.

In order to identify a point relative to the combustion chamber, a new cartesian reference frame was created, the origin of which was placed in the centre of the squish-jet outlet for the groove closest to the laser source. The line on the PIV plane connecting this origin with the cylinder centre was the x-axis of the new reference frame. The key advantage of implementing this reference frame is the ease of access to velocity information at the region of interest (the centreline of the squish-jet), regardless of chamber type or orientation. Computer code was written in Pascal, called ‘PIVVelMap’, that used the coordinates (in the camera’s reference frame) of three points on the bowl rim to identify the new reference frame’s origin and x-axis. The new y-axis was accordingly placed perpendicularly to the new x-axis. The orientation of the new universal reference frame is illustrated in figure 3.24. A grid with 40 divisions in the x direction and 50 divisions in the y direction was also created in the new reference frame. The grid covered more than an entire quadrant of each photographed combustion chamber. Since the coordinate transformation relating the two reference frames was known, PIVVelMap used linear interpolation in the old reference frame to find the velocity at each new grid point. Having assigned velocities to the universal grid for each PIV test (for a chosen crank angle and chamber shape), the code then determined the ensemble velocities for each grid point. The result was a new set of grid points (in a universal reference frame) whose mean velocities can be displayed in
vector plotting software. The standard deviation of the radial and tangential velocity components could also be determined for chosen points of interest.

For the simplest combustion chamber tested, the Bowl-in-Piston chamber, its symmetry was exploited to determine the mean velocity. The cylinder plane was divided into a number of 2 mm thick imaginary concentric rings that contained velocity vectors. The arithmetic mean of the vectors was used to represent the mean velocity at the mid-circumference of each ring. The maximum variation observed (between UBCRICM test runs) in mean velocity at fixed positions was 4%. Other useful quantities such as the standard deviation of the vectors per ring were also determined.

The computer software chosen to plot the resultant vector fields was ‘Tecplot’. PIVVelMap contains a subroutine that outputs the velocity data in a format required by Tecplot.

3.2.8 PIV Error Sources

Assuming valid determinations of particle group displacements, the sources of error in PIV measurements can be grouped into three categories:

a) Errors due to particle infidelity to the flow.

b) Errors in tracking the particle displacements.

c) Statistical uncertainty of the mean flow.

As previously mentioned, timing errors are disregarded because they are relatively insignificant.

Group a) errors have already been discussed with the conclusion that there is no significant discrepancy between the motion of the Nylon particles used for testing and the low frequency component of the flow.

Assuming a typical laser pulse separation time of 200 μs, a displacement resolution of 0.1 pixels and an image length conversion factor of 15 pixels/mm, the group b) uncertainty in a velocity measurement is 33 mm/s. This represents a relative uncertainty of 3.3% for 1 m/s flows.
The range of velocities in which the probability of finding the true ensemble mean is 90% is known as the 90% confidence interval for this mean. The interval can be found with the use of Student's t-distribution. This statistical distribution is similar in form to the normal distribution, but it depends on the number of measurements, N, used to find the mean.

The 90% confidence interval can be described by the following probability statement, that is similar to the one found in Bhattacharyya and Johnson (1979):

\[
\Pr \left[ V_{mean} - t_{0.05} \frac{s}{\sqrt{N}} < \mu_v < V_{mean} + t_{0.05} \frac{s}{\sqrt{N}} \right] = 90\% 
\]

Equation 3.12

where \( V_{mean} \) is the measured mean velocity,
\( \mu_v \) is the true mean velocity,
\( t_{0.05} \) is the t value above which the area under the t-distribution is 0.05
and \( s \) is the standard deviation in the sample set of velocities.

Therefore it follows that the 90% confidence interval is:

\[
V_{mean} \pm t_{0.05} \frac{s}{\sqrt{N}} 
\]

Equation 3.13

As the above equation shows, the statistical uncertainty depends on the rms velocity fluctuation. This value varies with crank angle and location. Therefore the statistical uncertainties will be reported in the results section of the dissertation along with the rms values.

As a rough guide, however, the ensemble mean velocity in both the radial and tangential directions had a typical total uncertainty of around 0.3 m/s.
3.3 LDV Measurement System

As a supplement to the mean velocity determined by PIV, another desirable parameter used to assess the flow validity of KIVA-3V (especially the code’s turbulence model) was the rms velocity fluctuation. As explained, LDV was the preferred measurement technique for this quantity. Ensemble-mean velocities were also determined by LDV and compared with PIV measurements. The system used was a TSI 9100-7 velocimeter. A detailed description of the system’s components can be found in the TSI user’s manual. The LDV measurements were made at a fixed point and the directions for the velocity measurements were: a) radially inward toward the chamber centre and b) tangentially clockwise (as viewed through the cylinder head). The measurement location and the velocity directions are shown in figure 5.44. Only one measurement location was used because an rms value determination requires a large number of tests to keep the statistical uncertainty at a reasonable level. For example, from classical statistical theory (Bhattacharyya and Johnson, 1977) one can show that at least 100 velocity samples are needed to be 90% certain that the rms velocity fluctuation is within 13% of the true rms value. Given that velocities are determined one direction at a time, it follows that at least 200 samples are needed for the 2 orthogonal rms velocities. Given the research time constraint and the considerable set-up time required for each measurement point and UBCRICM firing, it was unrealistic to conduct tests at more than one location. Nevertheless, the point chosen for rms velocity validation was considered to be an ‘acid test’ for KIVA-3V because a changeable history of turbulence production and dissipation was expected near the jet opening of the Squish-Jet 2 chamber.

The LDV system worked by using a large convex lens to focus two parallel monochromatic and coherent laser beams of similar thickness and intensity so that they crossed at the measurement location inside the chamber. The tiny ellipsoidal volume that was formed by the intersection of the two beams is called the probe volume and it was approximately 100 microns wide. Before they were focussed, the two parallel beams travelled in a direction normal to the cylinder head and toward it. A photomultiplier was exposed to the scattered light at the midline between the parallel beams. This orientation constitutes the backscatter configuration, because only light scattered back through the focussing lens and into the photomultiplier was observed. The
configuration is shown in figure 3.25. Even though the detection of forward scattered light is preferred because it is orders of magnitude more intense than back scattered light, this configuration was clearly not possible because the UBCRICH piston and crank case would block the photodetector’s view. When a micron sized particle crossed the probe volume, the light it scattered back toward the photomultiplier from each beam was Doppler shifted by an amount depending on the particle’s velocity and the beam’s incidence angle (relative to the velocity). The photomultiplier would thus be exposed to the sum of the two Doppler shifted frequencies. The summed frequencies produce a beat frequency that is proportional to the speed of the particle in a certain direction. This direction is normal to the line bisecting the two beams and in the common plane of the two beams.

It follows that if the beams are placed in a horizontal plane, the horizontal speed is detected. Similarly, the speed in the vertical direction is sensed by the photomultiplier if the beams were placed in a vertical plane. This method was used to measure either the radial or the tangential velocity components.

A more common explanation of the working principle behind LDV uses light interference. As shown in figure 3.26, the interference pattern produced in the probe volume comprises a series of equi-distant small dark planes (or fringes) that are parallel to the bisector of the two laser beams. The fringe planes are also normal to the plane connecting the two laser beams. The fringe separation, $\Delta x$, is:

$$\Delta x = \frac{\lambda}{2 \sin \varphi}$$

Equation 3.14

where $\lambda$ is the wavelength of the laser beams and $\varphi$ is the half angle between the beams.

When a particle crosses these fringes, it intermittently scatters light for a finite period of time with a blinking frequency (or Doppler frequency), $f_d$, which is proportional to the speed component normal to the fringes ($v_n$). Thus,

$$v_n = \frac{\Delta x}{T_d} = f_d \Delta x$$

Equation 3.15
where $T_d$ is the blinking period. The short period of pulsing light is often called a ‘Doppler burst’.

The photomultiplier amplifies the weak backscattered Doppler burst and converts it into an electrical signal whose amplitude is proportional to the intensity of the light signal. Figure 3.27 shows an example of a signal recorded on an oscilloscope during preliminary tests with water mist sprayed from a pump-action perfume dispenser. The raw signal from the photomultiplier (at the top of the oscilloscope screen) shows the Doppler burst resulting from a water droplet’s transit through the probe volume. It is worth noting the raw signal’s Gaussian envelope signifying the intensity profile of the laser beams. The lower signal results from filtering out frequencies lower than 10% of the Doppler frequency, which explains the disappearance of the Gaussian profile. In addition, a high frequency cut-off filter was used to eliminate extraneous noise.

An acousto-optic device known as a Bragg cell was used on one of the beams to shift its light frequency by 40 MHz. This caused the interference fringes to move in time, so that even a stationary particle inside the probe volume would produce an artificially biased Doppler shift (of 40 MHz). If the particle moved against the direction of the moving fringe pattern with speed $v_B$, the Doppler frequency would increase by an amount $v_B/\Delta x$. If the particle moved in the direction of the moving fringe pattern with the same speed, the Doppler frequency would decrease by an amount $v_B/\Delta x$. Thus, the Bragg cell introduced directionality to the speed of a particle in the probe volume.

In a hypothetical flow in which the highest speed in either direction perpendicular to the fringes is $v_h$, the corresponding range in Doppler frequency is $(40 \text{ MHz} - v_h/\Delta x)$ to $(40 \text{ MHz} + v_h/\Delta x)$. In order to ease the demand on the sampling rate of the acquisition system recording the photomultiplier signals, the photomultiplier output could be added to an artificial signal whose frequency is $(40 \text{ MHz} - v_h/\Delta x)$. This is called ‘downmixing’ and the highest velocity in the direction of fringe movement corresponds to a downmixed frequency of 0 Hz. With this downmixing, the highest velocity against the fringe movement corresponds to a downmixed frequency of $2 v_h/\Delta x$. A downmixer was used in the LDV experiments for this research.

Figure 3.28 shows a schematic overview of the LDV system in relation to the UBCRICM.
3.3.1 The LDV Laser

The light source for the LDV system used was a 5W Spectra-Physics Argon ion laser. The optimum power setting of 0.8 W was used during the experiments. Power levels beyond this setting introduced more noise to the photomultiplier signal, while lower power levels weakened the signal strength so that it was more difficult to discern Doppler bursts.

The argon ion laser produces linearly polarised light composed of a number of frequencies. After leaving the laser, these frequencies were separated into distinctly coloured beams in an enclosure comprising a series of prisms and mirrors. The two most dominant frequencies had wavelengths of 488 nm (blue) and 514.5 nm (green). The beams with these frequencies were separated from the remaining beams by placing a black panel in front of the different coloured beams and allowing the two desired beams to pass through carefully positioned holes in the panel. The green beam was designated for vertical velocity measurements while the blue one was used for horizontal measurements.

Figure 3.29 shows a photograph of the interior of the colour-separating module during operation. Smoke was introduced to make the different coloured beams conspicuous.

3.3.2 Beam Delivery

After leaving the colour separator the green and blue beams entered a cylindrical optical assembly that performed a number of optical functions including splitting, frequency shifting, steering and masking. As a result, the green beam was transformed into two parallel beams in a vertical plane and the blue beam turned into two parallel beams in a horizontal plane. The parallel beams for each beam pair were 2" (50.8mm) apart. Furthermore, both beam pairs shared the same midline. As shown in figure 3.28, a 45 degree mirror was placed at the exit of the optical assembly so that it reflected the beams onto another 45 degree mirror mounted on a horizontal traverse that moved parallel to the cylinder head. This enabled one to move the laser beams horizontally to a desired location in the UBCRICM combustion chamber. Vertical
movement of the beams was achieved by raising or lowering the optical bench on which the laser and the optics described were mounted. The final stage of beam manoeuvring was performed by the focusing lens, which converged the beams to form the probe volume inside the combustion chamber. The fringe separation for the blue probe volume was 4.32 microns and it was 4.61 microns for the green volume. The focusing lens had a focal length of 480 mm and it was mounted on the same 3-D traverse the PIV camera was attached to. The mirror and lens traverse combination was used to move the probe volume to the desired location.

The LDV system had only one Bragg cell and one working photomultiplier; thus, only one velocity direction could be measured during each experiment. In order to avoid confusion during alignment, the blue beams were masked during vertical velocity measurements. Similarly, the green beams were masked during horizontal velocity measurements.

One of the most important procedures when setting up the LD Velocimeter was to ensure that the beams crossed with maximum overlap. This ensured the formation of many fringes that provided a strong photodetector signal when a particle traversed them. The maximum overlap was achieved by finely adjusting the direction of one of the beams per pair with the beam steering component in the optics assembly. As the beam width was extremely small, a microscope lens was placed in the vicinity of the probe volume to aid the alignment. The image projected from the lens onto a black board in the background was used to monitor the degree of overlap. Figure 3.30 shows the magnified image of the green probe volume after the overlap was optimised.

A flat white paper surface with a dot marked on it was placed inside the combustion chamber so that the dot coincided with the desired measurement point. The probe volume was moved to cover the dot and the focusing lens was then gradually moved toward or away from the cylinder head with the lens traverse until a strong sinusoidal signal was observed from the photomultiplier. This ensured that the probe volume was placed at the measurement point.

3.3.3 Particles and Flow Seeding

The seed particles used for LDV measurements were silicone resin microspheres from ‘GE’. The microspheres had a nominal diameter of 2 microns and a specific gravity of 1.3. Figure 3.31
shows a microscope image of the particles after being deposited on a slide with the seeding system used during the experiments. The particles were small enough to track the turbulent flow, yet just large enough to scatter a noticeable amount of light for the photomultiplier. Except for the mass of seed used per test, the seeding technique was identical to that used for the PIV measurements. The mass (0.2 mg) was less than 12% of the mass used for PIV tests because the LDV particles were less than half the diameter of the PIV particles. Both particle types had a similar density. Since the mass of the air inside the cylinder was around 1080 mg, the 0.2 mg of seed particles made a negligible change to the overall density inside the cylinder.

3.3.4 Photomultiplier and Data Acquisition

The sparse light backscattered from the probe volume in the combustion chamber went back through the focusing lens and was reflected into the photomultiplier unit with a small 45 degree mirror. Before the light entered the photomultiplier tube it was filtered to only allow the light frequency of the probe volume. The scattered light was then focussed with a lens onto a pinhole designed to allow only light scattered from the probe volume. When the light finally entered the photomultiplier tube, each photon was converted into an avalanche of electrons in negligible time. The result was a greatly amplified electrical signal representing the scattered light intensity.

Spurious light that was reflected from the cylinder head or piston was minimised by masking regions of the focusing lens with black electrical tape. This measure greatly increased the signal to noise ratio of the photomultiplier.

The signal output from the photomultiplier was fed into the downmixer. The signal from the downmixer was then input to the TSI model 1984B input conditioner which applied a low-limit and high-limit filter. The filtered signal was finally fed into a Tektronix TDS 420A digital storage oscilloscope. The processed photomultiplier signal was recorded when the oscilloscope received the crank angle trigger. The recorded signal was then transferred to a PC for subsequent analysis. The maximum record length of the oscilloscope was 30,000 samples and at a sampling frequency of 5MHz and a crank speed of 800 rev/min, the photomultiplier signal was recorded between 27 degrees BTDC and 0 degrees BTDC. A larger crank angle range was preferred,
however, this would have necessitated a sample frequency that would have been too low to
detect the whole velocity range in the probe volume.

3.3.5 LDV Recording Procedure

After the initial alignment and positioning of the probe volume, a systematic procedure for LDV
measurements was adhered to each time the UBCRICM was fired. The procedure was necessary
in order to minimise the chance of time-consuming mistakes that cause equipment damage
and/or null experiments.

The procedure is now presented:

- Ensure Argon-ion laser is warmed up.
- Turn UBCRICM compressor on.
- Ensure that crank trigger is connected to oscilloscope.
- Ensure oscilloscope is on.
- Ensure photomultiplier power supply is on.
- Ensure PC is on.
- Retract piston to BDC.
- Clean and reassemble cylinder.
- Load 2 micron silicone resin microspheres (~0.25 mg) in mixing flask
- Ensure that air pressure in compressor has reached level needed for 800 rev/min.
- Do mock trigger and signal capture by flicking the crank trigger’s power supply on and off.
- Check crank position.
- Check UBCRICM valve positions are set for firing.
- Check that the UBCRICM solenoid switch is powered.
- Check UBCRICM head tightness.
- Turn out lights and set laser power to 0.8W.
- Squirt particles into cylinder and wait for flow to subside.
- Fire UBCRICM.
- Transfer photomultiplier signal to PC.
Each time the UBCRICM was run for LDV tests the data recorded by the storage oscilloscope contained a 6 ms time history of the processed photomultiplier voltage. The recorded signal required digital processing in order to extract velocity data as a function of crank angle.

Subsequently, Pascal computer code (called ‘Envelope’) was created that searched through a given series to identify times where Doppler bursts existed. This was achieved by first extracting the envelope of the series. The envelope was another series containing only the local maxima (and their times) of the input series. The envelope was then smoothed by applying a moving average and if the local maxima in the smoothed envelope exceeded a characteristic threshold voltage they were identified as the centres of Doppler bursts. A small file containing the list of times at which Doppler bursts were detected was then created.

A Matlab program called ‘LdvProcessor’ was also written to apply an FFT (Fast Fourier Transform) on the photomultiplier data series in the vicinity of the Doppler burst occurrence times. The dominant frequency observed in each Doppler burst was then converted into a velocity. At the same time, the Doppler burst time was converted into the corresponding crank angle. At the end of the data file analysis, LdvProcessor generated an output file containing a list of crank angles and their associated particle velocities. Since approximately forty UBCRICM tests were done for each velocity direction, the same number of output files were produced for each direction. On average, each file contained three velocity measurements for every five crank angle degrees. Mean velocities and rms velocity fluctuations were extracted from these files for comparison against KIVA-3V predictions.

Figures 3.32 and 3.33 illustrate the processes used in Envelope and LdvProcessor to derive velocity versus crank angle histories from photomultiplier records.

**Determination of Mean Velocity and rms Velocity Fluctuation**

Before determining how to calculate the mean velocity and the rms velocity fluctuation at a given position and crank angle it is worthwhile becoming familiar with the various statistical terms used to describe velocities inside combustion chambers. The following is a list of velocity...
terms adapted from Fraser and Bracco (1989). Unless otherwise qualified in the list, the term ‘velocity’ will refer to the instantaneous velocity of gas in the combustion chamber at a given point and crank angle. For this research the cycle is the test number of the UBCRICM for a given experimental case.

<table>
<thead>
<tr>
<th>Term</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ensemble-Averaged Velocity</td>
<td>Ensemble average of the velocity (for all cycles).</td>
</tr>
<tr>
<td>Bulk Velocity:</td>
<td>Low frequency large scale component of velocity in given cycle, evaluated for example, by low-pass cycle-resolved filtering.</td>
</tr>
<tr>
<td>Ensemble-Average Bulk Velocity:</td>
<td>Ensemble average of the bulk velocity (for all cycles).</td>
</tr>
<tr>
<td>Rms of the Bulk Velocity:</td>
<td>Ensemble average rms of the difference between the bulk velocity and the ensemble-average bulk velocity (for all cycles).</td>
</tr>
<tr>
<td>Turbulence Velocity:</td>
<td>Difference between the velocity and the bulk velocity in given cycle.</td>
</tr>
<tr>
<td>Fluctuation Velocity:</td>
<td>Difference between the velocity and the ensemble-averaged velocity in given cycle.</td>
</tr>
<tr>
<td>Turbulence Intensity:</td>
<td>Ensemble average rms of the turbulence velocity (for all cycles).</td>
</tr>
<tr>
<td>Fluctuation Intensity:</td>
<td>Ensemble average rms of the fluctuation velocity (for all cycles).</td>
</tr>
</tbody>
</table>

The list arises from the hypothesis that there are two fluctuating components of velocity in the cylinder:

a) Bulk velocity - This is the mean velocity in a particular cycle. This velocity is associated with large scale flow in the combustion chamber which can vary from cycle to cycle. These cycle-to-cycle variations have been attributed by a number of researchers (Arcoumanis et al., 1987; Miles et al., 2002) to variations in the flow boundary conditions (e.g., velocity fluctuations within the intake manifold). The ensemble averaged bulk velocity is the datum about which the bulk velocity fluctuates. As mentioned in section 3.2.7, the cyclic variations in bulk velocity were negligible for the UBCRICM.
b) Turbulence velocity - This is associated with random small scale flow that is superimposed on the bulk velocity.

As shown in the list, there are three different mean velocities that result from this hypothesis: the bulk velocity, the ensemble averaged bulk velocity and the ensemble averaged velocity. There are also two different types of velocity fluctuations: the turbulence velocity and the fluctuation velocity. The choice regarding which mean quantity and which fluctuating one should be used to compare with KIVA-3V depends on the counterpart values used in the code.

Based on specified initial and boundary conditions, KIVA-3V finds (for each solution point in the combustion chamber) one mean velocity and one encompassing measure of velocity fluctuations. KIVA-3V does not distinguish between types of fluctuations and it does not model the effect of uncertainty in the initial and boundary conditions. Despite the simplistic assumptions of the $k$-$\epsilon$ turbulence model in KIVA-3V (e.g., turbulence isotropy), the turbulent kinetic energy it evaluates covers all scales of motion. Thus for a given point and crank angle, KIVA-3V essentially determines the ensemble averaged velocity and the fluctuation intensity. For a rigorous validation study these two quantities should therefore be used to compare with KIVA, on the condition that the initial and boundary conditions are experimentally duplicated.

In situations unlike the UBCRICM, where cycle-to-cycle variations in initial and boundary conditions cannot be avoided (e.g., motored engines), it may be useful to subtract the bulk velocity fluctuations from the total fluctuations. This subtraction may compensate for the cycle-to-cycle variation, thus enabling a less biased comparison with the output KIVA-3V produces from fixed initial and boundary conditions. Nevertheless, this compensation method is inexact because the cut-off frequency used to extract the bulk velocity often overlaps with turbulence frequencies (Tabaczynski, 1983).

Although LDV is very accurate and has very high spatial and temporal resolutions, it does suffer from one drawback: it cannot be used to make continuous measurements, due to the finite distances between seed particles. Increasing the particle density does not help because the chance of more than one particle traversing the probe volume at the same time increases. This is problematic because the photomultiplier signal becomes too noisy. The inability to make continuous velocity measurements is not important in stationary flows, because velocity statistics
are by definition independent of time—measurements at random times can be used to find quantities like mean and rms velocity fluctuations. In non-stationary cases, the flow may be repeated (cycled) to obtain a measurement at an identical time in each cycle. The measurements are then averaged to yield the ensemble average. This is clearly unachievable with LDV. Nevertheless, the problem is typically circumvented by taking measurements in a window of time centred at the moment of interest.

For example, in the case of reciprocating engines, the ensemble-averaged velocity at a given crank angle, \( \theta \), is approximated as:

\[
U_{EA}(\theta) = \frac{1}{N_c} \sum_{i=1}^{N_c} \sum_{j=1}^{N_i} U_{i,j}(\theta \pm \frac{\Delta \theta}{2})
\]

Equation 3.16

where \( U_{i,j}(\theta \pm \Delta \theta/2) \) is the jth velocity measurement of cycle i in the measurement window, \( \Delta \theta \) is the measurement window period, \( N_i \) is the number of velocity measurements recorded in cycle i, \( N_c \) is the number of measurement cycles, and \( N_i \) is the total number of measurements.

Accordingly, the ensemble-averaged rms velocity fluctuation at centred at crank angle \( \theta \) is:

\[
u_{rms}(\theta) = \sqrt{\frac{1}{N_c} \sum_{i=1}^{N_c} \sum_{j=1}^{N_i} \left[ u_{i,j}(\theta \pm \frac{\Delta \theta}{2}) \right]^2}
\]

Equation 3.17

where

\[ u_{i,j}(\theta \pm \frac{\Delta \theta}{2}) \]

is the fluctuation velocity for the jth measurement in cycle i, and is found by:

\[
u_{i,j}(\theta \pm \frac{\Delta \theta}{2}) = U_{i,j}(\theta \pm \frac{\Delta \theta}{2}) - U_{EA}(\theta)
\]

Equation 3.18
For brevity, the ensemble-averaged rms velocity fluctuation is simply called the rms velocity from here on in this document.

**Turbulence Scales**

Another parameter that can be used to validate KIVA-3V is the integral length scale, $L_i$. The integral scale characterises the size of large eddies responsible for most of the turbulence production. If velocities are measured at two points that are separated by a distance much smaller than $L_i$, the measurements will strongly correlate with each other. As the separation, $x$, increases the autocorrelation coefficient dwindles. Therefore, the integral length scale is defined as the integral of the autocorrelation coefficient with respect to $x$ (for all $x>0$). The integral length scale could not be directly measured because the LDV system did not have the ability to simultaneously measure velocities at two separate points.

In some flows the integral length time, $\tau_i$, can be used to find $L_i$. The integral length time is the temporal counterpart to $L_i$. It was possible to estimate $\tau_i$ from the LDV measurements because only single point velocity measurements over time are needed. If the large scale eddies in the flow do not distort while flowing past the measurement point, $\tau_i$ represents the time taken for a large scale eddy to flow past the point. The distortion is considered insignificant if the velocity fluctuations are less than 20% of the mean velocity (Anetor, 1994). In these undistorted or ‘frozen field’ conditions, Taylor’s hypothesis applies so that:

$$L_i = U \cdot \tau_i$$

Where $U$ is the mean flow velocity at the measurement point.

Unfortunately, the high levels of turbulence generally observed inside combustion chambers tends to invalidate Taylor’s hypothesis so that the integral length time cannot be used to find the integral length scale.
3.3.7 LDV Error Sources

Individual *particle* velocity measurements made by LDV are among the most accurate of velocimeters. Examination of the parameters (found in equations 3.14 and 3.15) used to determine the velocity from the frequency detected by the photomultiplier provides quick confirmation. For example, the laser beam wavelength ($\lambda$) is specified to within 4 significant figures, giving a negligible fractional error of less than 0.1%. The resolution of the frequency determined by FFT analysis is the reciprocal of the Doppler burst period, giving a typical fractional error in frequency of 3%. The uncertainty in the beam convergence half angle was estimated to be around 2%. Thus, the total uncertainty in the velocity of a particle was at most 5%.

**Particle Response**

Unfortunately, the uncertainty in rms or mean *gas velocities* found with LDV has more error sources than for the simple particle velocity measurement. As was already discussed, the flow velocity is indirectly determined by measuring the particle velocity. Therefore, an important LDV requirement when measuring the fluid flow is for the seed particles to follow the flow. If this requirement is not met, errors result depending on the size and density of the particle. As indicated in figure 3.18 the particles chosen for this research were expected to deviate from the air motion by 1% at the very most.

**Velocity Bias**

A further potential problem attributed to the intermittent velocity measurements in LDV is known as ‘velocity bias’. During a measurement window, more particles will pass through the probe volume if they have higher velocities. Thus, these more frequent fast particle measurements artificially bias the computed average towards higher values. It follows that this velocity bias can be compensated for, by weighting each sample by the inverse of its velocity (McLaughlin and Tiederman 1973). Nevertheless, the correction is entirely unnecessary when the ideal situation occurs in which a single measurement is taken at a desired crank angle for a
large number of cycles. This ideal condition is approached if the number of measurement cycles is high enough and the number of measurements per cycle window is small. After observing negligible correction values, this favourable situation was confirmed for the LDV data recorded in this research.

**Fringe Bias**

Another potential source of error with LDV is known as fringe bias or directional bias. This bias tends to favor measurements associated with particles flowing perpendicularly to the probe volume fringes. The opposite extreme occurs when a particle flows almost parallel to the fringes so that only one fringe is crossed. In this case the velocity cannot be measured due to the absence of a Doppler frequency. Several fringe crossings are needed in order to make a valid measurement, so particles passing through the probe volume at large angles may not be registered. Fringe bias is reduced by Bragg shifting one of the beams, so the fringes are always moving relative to the flow particles. In order to essentially eliminate fringe bias, Whiffen (1975) recommends a frequency shift of at least twice the Doppler frequency that a particle would produce with stationary fringes. As the 40 MHz Bragg shift used in this research far exceeded the 2.2 MHz expected from the fastest particles crossing stationary fringes, directional bias was deemed eliminated.

**Statistical Uncertainties**

The 90% confidence interval for the mean velocity was calculated using equation 3.13. For N values greater than 30, $t_{0.05}$ is considered independent of N and is approximated as 1.65. This condition applied to the LDV data, because in general more than 100 velocity measurements were made per crank angle window. Since the standard deviation of these velocity measurements was at most 1.7 m/s, the maximum statistical uncertainty in the ensemble-averaged velocity was 0.25 m/s.
The statistical uncertainty for the rms velocity was found by using the $\chi^2$ distribution. This statistical distribution depends greatly on N and unlike the normal distribution, is asymmetrical. The 90% confidence interval for the rms velocity can be expressed symbolically as:

$$\Pr\left[ s \sqrt{\frac{N-1}{\chi^2_{0.05}}} < \sigma < s \sqrt{\frac{N-1}{\chi^2_{0.05}}} \right] = 90\%$$

Equation 3.19

where $\sigma$ is the true standard deviation in velocity,

$\chi^2_{0.95}$ is the $\chi^2$ value above which the area under the $\chi^2$-distribution is 0.95

$\chi^2_{0.05}$ is the $\chi^2$ value above which the area under the $\chi^2$-distribution is 0.05

$s$ is the measured standard deviation in velocity

As N for each crank angle window was at least 100 and $s$ was at most 1.5 m/s, the estimated maximum fractional uncertainty in the rms velocity was calculated to be 13%. It is worth noting that given the same standard deviation, the number of velocity measurements required in order for this uncertainty to drop to 5% is around 600. Furthermore, N must reach about 1500 for a 3% uncertainty.

**Window Broadening Error**

Variation of the mean velocity during each measurement window leads to an overestimation of the rms velocity that is termed window broadening (Rask, 1984). This source of error can become significant if the window is wide enough and the variation in mean velocity is on the order of the local turbulent fluctuations. Comparison of the standard deviation of the mean velocity inside each window with the overall rms velocity (radial and tangential) confirmed that, for this study, window broadening error was negligible.
Summary of Uncertainty analysis

For each direction measured, the significant uncertainties for the ensemble mean velocity were added to yield the overall uncertainties. The summation was also done for rms velocities. The statistical components of the overall uncertainties depend on rms velocities for individual crank angles. As the rms values are reported in the results section of the dissertation, the overall uncertainties are reported there too.

As a rough guide, however, the ensemble mean velocity in the radial direction had a maximum relative uncertainty of 13% (for a mean radial velocity of 3.05 m/s), while in the tangential direction the absolute uncertainty was 0.17 m/s. The relative uncertainty for the tangential direction is meaningless to report because of the near zero tangential speed.

The rms velocity in the radial and tangential directions had a maximum relative uncertainty of 19%.
3.4 Figures

Figure 3.1: Schematic Diagram of the UBCRICM.
Figure 3.2: The transparent acrylic cylinder. Dimensions in millimetres unless shown otherwise.
Figure 3.3: The clear acrylic piston base. Dimensions in millimetres unless stated otherwise.
Figure 3.4: The clear acrylic piston crown for the Bowl-in-Piston combustion chamber. Dimensions in millimetres unless specified otherwise.
Figure 3.5: The clear acrylic piston crown for the Squish-Jet 1 combustion chamber. Dimensions in millimetres unless specified otherwise.
Figure 3.6: The clear acrylic piston crown for the Squish-Jet 2 combustion chamber. Dimensions in millimetres unless specified otherwise.
Figure 3.7: The clear acrylic cylinder head. Dimensions in millimetres unless specified otherwise.

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Figure 3.18: The simulated displacement history for 3 differently sized particles following fluctuating squish flow inside the Bowl-in-Piston chamber. The motion of the air that the particles are responding to is also shown. The particles have a density of 1000\(\text{kg/m}^3\). The motion of the 2 micron particle coincides with that of the air.

Figure 3.19: The simulated displacement history for 3 differently sized particles following fluctuating squish flow inside the Bowl-in-Piston chamber. The motion of the air that the particles are responding to is also shown. The particles have a density of 4000\(\text{kg/m}^3\).
Figure 3.20: The particle seeding system. The conical flask doubles as a cyclonic separator.

Figure 3.21: Microscope image of nylon particles used for PIV experiments. The calibrated cross-hairs are 21 microns wide.
Cross-Correlation between R1 and R2 = C (x₁, y₁, dₓ, dᵧ)

= d₁₁·e₁₁ + d₁₂·e₁₂ + d₁₃·e₁₃ + ... + d₄₄·e₄₄

Figure 3.22: Computing the cross-correlation between regions R₁ and R₂. A pixel is selected in R₁ and multiplied with the corresponding pixel in R₂. This is done for all pixels in R₁. The sum of the products is the cross-correlation.
Figure 3.23: Cross-correlating region $R_1$ with a range of different $R_2$ regions (that are displaced $d_x$, $d_y$ relative to $R_1$) is represented by a surface plot. The distinct peak in the cross-correlation surface indicates the displacement of the particle group in $R_1$ during the time between image 1 and image 2.

Figure 3.24: Orientation of the universal reference frame adopted for PIV processing.
Figure 3.26: The interference pattern produced by two crossing laser beams. Lines represent light wave maxima. The fringes in the interference pattern are small planes that are parallel to the midline of the laser beams and perpendicular to the plane connecting the two laser beams.
Figure 3.27: Photomultiplier signal recorded when a particle crossed the LDV probe volume. The signal was recorded on an oscilloscope during preliminary tests. Upper signal is raw signal from photomultiplier. Lower signal is the bandpass filtered raw signal and it more clearly shows the Doppler frequency.
Figure 3.28: Schematic overview of the LDV system in relation to the UBCRICM.
Figure 3.29: Photograph inside the colour separation module during operation. Smoke was introduced to make the different coloured beams conspicuous.

Figure 3.30: Magnified image of the green probe volume after the beam overlap was optimised. The fringe separation is around 4.6 microns.
Figure 3.31: Microscope image of the GE microspheres (nominal diameter of 2 microns) after being deposited on a slide with the seeding system used during the experiments. The dark calibration line is 21 microns wide.
Figure 3.32: The processes used in 'Envelope' to identify the sample number (hence time) of a Doppler burst in an LDV signal. The 'envelope' of the filtered photomultiplier signal is first formed by identifying local maxima (top graph). Local maxima are identified in the smoothed envelope (bottom graph) to determine Doppler burst times (e.g., $N_{\text{max}}$).
Figure 3.33: Sample output from 'LdvProcessor': the dominant frequency is found from FFT analysis of each Doppler burst. The burst times are determined by 'envelope'. The times and the dominant frequencies are subsequently converted into crank angles and velocities respectively.
### 3.5 Tables

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>100 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>100 mm</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>175 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>9.1:1</td>
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<tr>
<td>Initial Cylinder Pressure</td>
<td>Ambient</td>
</tr>
<tr>
<td>Squish Area Ratio</td>
<td>75%</td>
</tr>
<tr>
<td>Driving Cylinder Pressure</td>
<td>120 psig (805 kPa gauge)</td>
</tr>
<tr>
<td>Simulated Engine Speed (Crank Speed)</td>
<td>800 rev/min</td>
</tr>
</tbody>
</table>

Table 3-1: Important UBCRiCM Specifications.

<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>LASER</td>
<td>Gemini PIV 120-15</td>
</tr>
<tr>
<td>Wavelength</td>
<td>532 nm</td>
</tr>
<tr>
<td>Max. pulse energy</td>
<td>120 mJ</td>
</tr>
<tr>
<td>Pulse width</td>
<td>3-5 ns</td>
</tr>
<tr>
<td>Beam diameter</td>
<td>4.5 mm</td>
</tr>
<tr>
<td>Beam divergence</td>
<td>&lt; 2 degrees</td>
</tr>
<tr>
<td>SIGNAL GENERATOR</td>
<td>Berkeley Nucleonics model 500D</td>
</tr>
<tr>
<td>Number of channels</td>
<td>8</td>
</tr>
<tr>
<td>External trigger</td>
<td>1</td>
</tr>
<tr>
<td>Time accuracy</td>
<td>25 ns</td>
</tr>
<tr>
<td>Time resolution</td>
<td>0.2 µs</td>
</tr>
<tr>
<td>FRAME GRABBER</td>
<td>Matrox Meteor II-Digital</td>
</tr>
<tr>
<td>CAMERA</td>
<td>Roper Scientific ES 1.0</td>
</tr>
<tr>
<td>PIV PROCESSING SOFTWARE</td>
<td>Dantec Flowmanager</td>
</tr>
</tbody>
</table>

Table 3-2: Hardware and software specifications of PIV system.
<table>
<thead>
<tr>
<th><strong>CAMERA DESIGNATION</strong></th>
<th><strong>Roper Scientific Megaplus ES 1.0</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Operation mode used</td>
<td>Double exposure</td>
</tr>
<tr>
<td>Lens used</td>
<td>50 mm focal length C mount</td>
</tr>
<tr>
<td>Pixel clock rate</td>
<td>20 MHz</td>
</tr>
<tr>
<td>Exposure time for first frame</td>
<td>1--999 μs</td>
</tr>
<tr>
<td>Exposure time for second frame</td>
<td>33 ms</td>
</tr>
<tr>
<td>Resolution</td>
<td>1008 H x 1018 V</td>
</tr>
<tr>
<td>Pixel size in CCD</td>
<td>9 μm (square format)</td>
</tr>
<tr>
<td>Bit Depth</td>
<td>10</td>
</tr>
<tr>
<td>Saturation illumination</td>
<td>0.037 μJ/cm² at 550 nm wavelength</td>
</tr>
</tbody>
</table>

Table 3-3: Specifications for the Roper Scientific ES 1.0 digital camera.
4 NUMERICAL MODELLING

Since a main objective of the present study is to test the validity of KIVA-3V flow predictions inside Squish-Jet combustion chambers, an overview of the numerical code is warranted. In this chapter the physical principles behind KIVA-3V and the numerical procedures it executes to predict flows are described. Spray dynamics and combustion solvers that KIVA-3V possesses are not explicitly described, because of their irrelevance to the present objectives. A more detailed description of KIVA-3V can be found in the manual written by Amsden et al. (1997).

4.1 General Background

KIVA-3V was designed to solve the unsteady equations of motion of a turbulent, chemically reactive mixture of ideal gases, coupled to the equations for a single-component vaporising fuel spray. The gas-phase solution procedure is based on a finite volume technique called the arbitrary Lagrangian-Eulerian (ALE) method. The method entails the formation of a computational mesh comprising a multitude of tessellating hexahedral cells that subdivide the combustion chamber geometry. The vertices of the cells may be arbitrarily specified functions of time, thereby allowing a Lagrangian or Eulerian (or combined) flow description. The arbitrary mesh can conform to curved boundaries whose shapes may also change in time. Mass, momentum and energy conservation laws are applied to each cell in the mesh to yield the relevant flow descriptors.

The original KIVA program was released to the public in 1984 and was superseded by an improved version, KIVA-II (Amsden, 1989). Although these early versions worked adequately for confined in-cylinder flow, they were considered inefficient for complex geometries having features such as diesel prechambers and long transfer ports. The inefficiencies were ascribed to excessive cell deactivation caused by the generation of a mesh structure that encompassed more space than the combustion chamber being modelled. This is because each cell in the mesh structure was identified by an ordered triplet of indices (i,j,k) and each index varied over a fixed range (i = 1, ..., NXP; j = 1, ..., NYP; k = 1, ..., NZP).
KIVA-3 removed this redundancy by changing the indexing structure of the mesh. In the new ‘block-structured mesh’, each cell is identified by a single index which is linked to information such as the cell vertices and the indices of neighbouring cells. KIVA-3 is also a more modular package than KIVA-II containing three separate programs: the mesh generator, the flow solver and the graphical output processor. This modular structure enables users to supply their own pre- and post-processors if they so desire.

KIVA-II had another shortcoming: the computational mesh was generated by combining geometrical blocks that were volumes of revolution about the cylinder’s longitudinal axis. The exception to this is the special function available to model square piston bowls. Even though a block’s longitudinal axis could be offset, this mesh generation method was clearly limited and was more amenable to two-dimensional cylinder shapes. With the block-structured mesh generation used in KIVA-3, an almost unlimited number of 3-dimensional combustion chamber shapes can be represented. It is worth noting that despite the different mesh structures between KIVA-II and KIVA-3, both codes share the same equations and solution algorithms.

The most significant modification that KIVA-3 underwent when its successor KIVA-3 V was created, was the addition of code to model the geometry and motion of valves. KIVA-3 V essentially solves the same equation sets and algorithms as do KIVA-3 and KIVA-II.

As with the rest of the KIVA series, KIVA-3 V is a Fortran program. For the current study, it was run on a Fortran compiler installed on an IBM compatible PC.

4.1.1 Solution Procedure

The solution for the time-dependent variables inside the cylinder is marched out in a series of time increments called cycles or timesteps. For each cycle, the values of the variables are calculated from those of the previous cycle. Two phases constitute the calculations in each cycle—a Lagrangian phase and a rezone phase. In the Lagrangian phase the vertices of each cell move with the fluid velocity and there are clearly no flux terms across cell boundaries. The Lagrangian phase is an operation in which Newton’s second law and the first law of thermodynamics are applied to the control mass in each cell. In the rezone phase the flow field is
frozen, the cell vertices are moved to occupy a new mesh (that accommodates piston and valve motion) and the flow field is remapped or rezoned into the new mesh. The thermodynamic quantities in each new cell are calculated by convecting material across the cell boundaries, which are regarded as moving relative to the flow field.

4.2 The Governing Equations

In this section, the equations of motion for the fluid phase and the corresponding boundary conditions are presented. The equations are similar to those shown in Amsden et al. (1989), with chemical and fuel spray terms omitted due to their redundancy in the current work. The two equations needed to attain closure for the turbulence model \((k-\varepsilon)\) are also shown. For succinctness, the cartesian vector notation was adopted. Vectors and tensors are represented by arrowed symbols, where \(\vec{i}\), \(\vec{j}\) and \(\vec{k}\) indicate the unit vectors in the x, y and z directions respectively. The position vector \(\vec{r}\) is defined by

\[
\vec{r} = x\vec{i} + y\vec{j} + z\vec{k}
\]

and the vector operator \(\nabla\) is defined as

\[
\nabla = \vec{i} \frac{\partial}{\partial x} + \vec{j} \frac{\partial}{\partial y} + \vec{k} \frac{\partial}{\partial z}
\]

The mean fluid velocity vector \(\vec{u}\) is given by

\[
\vec{u} = u(x, y, z, t)\vec{i} + v(x, y, z, t)\vec{j} + w(x, y, z, t)\vec{k}
\]

where \(t\) represents time.
4.2.1 The Fluid Phase

The fluid conservation equations used by the KIVA codes are presented in this section. The equations apply to the gas in the cylinder away from the wall boundary layers. The boundary layers are treated separately in the next section.

Continuity

The mass conservation equation at a given position vector and time is:

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0 \]  

Equation 4.1

where \( \rho \) denotes the air density.

The first term represents the mass accumulation rate per unit volume while the divergence term represents the mass flux per unit volume. The source term is zero because of the absence of chemical (and nuclear) reactions and species diffusion.

Momentum Conservation

The momentum equation for the air inside the cylinder is:

\[ \frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla P - \nabla (\rho \bar{\nabla} k) + \nabla \cdot \bar{\sigma} + \bar{\rho} \bar{g} \]  

Equation 4.2

where \( P \) denotes pressure,

\( k \) is specific turbulent kinetic energy,

\( \bar{g} \) is the specific gravitational force (constant),

\( \bar{\sigma} \) is the viscous stress tensor, which is Newtonian in form and is expressed as:
\[
\bar{\sigma} = \mu_{\text{eff}} \left[ \nabla \bar{u} + (\nabla \bar{u})^T \right] - \frac{2}{3} \mu \nabla \cdot \bar{u} \cdot \mathbf{I}
\]

Equation 4.3

The superscript T denotes the transpose and \( \mathbf{I} \) is the unit matrix.

\( \mu_{\text{eff}} \) is the coefficient of viscosity and it contains laminar and turbulent contributions.

Thus:

\[
\mu_{\text{eff}} = \mu_{\text{air}} + \mu_t = \mu_{\text{air}} + c_{\mu} \frac{k^2}{\epsilon}
\]

Equation 4.4

\( c_{\mu} \) is an empirical constant with a standard value of 0.09, while \( \epsilon \) is the rate of dissipation of specific turbulent kinetic energy.

The accumulation and flux of momentum are shown on the left side of equation 4.2. The last term of the equation is the gravitational body force per unit volume, while the first three terms on the right side are sources related to molecular and turbulent transport of momentum. It is worth noting that the pressure term is related to the normal stresses from molecular motion while the second term on the right is the turbulent analogy. Furthermore, it is important to acknowledge that the stress tensor is Newtonian in form because the assumption was made that the turbulence in the cylinder (and away from the walls) is isotropic. This concept, which was introduced by Boussinesq (1877), may be unrealistic for crank angles that are further advanced from TDC when there are large scale eddies with biased directionality.

The turbulent component in equation 4.4, \( \mu_t \), is based on the additional assumption that the turbulence is in equilibrium—i.e., the production of turbulence equals the dissipation. The assumption may be dubious in very unsteady flows such as those in reciprocating engines, and further potential for misrepresentation exists. Nevertheless, the specific dissipation of turbulent kinetic energy in these conditions is:

\[
\epsilon \propto \frac{k}{\tau}
\]

Equation 4.5
Since $\tau$ is the eddy turnover time, it can be approximated by:

$$\tau \approx \frac{l_i}{u_i} \approx \frac{l_i}{k^{1/2}}$$  \hspace{1cm} \text{Equation 4.6}

where $l_i$ is a characteristic eddy size and $u_i$ is the characteristic eddy speed.

Combining equations 4.5 and 4.6 shows that:

$$\varepsilon \approx \frac{k^{3/2}}{l_i}$$  \hspace{1cm} \text{Equation 4.7}

Revisiting the molecular analogy and applying equation 4.7 explains the form for the eddy viscosity, $\mu_i$, shown in equation 4.4.

$$\mu_i \approx \rho l_i u_i \approx \rho l_i k^{1/2} \approx \frac{k^2}{\varepsilon}$$

**Energy Conservation**

The internal energy equation is:

$$\frac{\partial (\rho l)}{\partial t} + \nabla \cdot (\rho l \vec{u}) = -P \nabla \cdot \vec{u} - \nabla \cdot \vec{J} + \rho \varepsilon$$  \hspace{1cm} \text{Equation 4.8}

where $I$ denotes the specific internal energy of the air and $\vec{J}$ stands for the heat conduction which is:

$$\vec{J} = -K \nabla T$$  \hspace{1cm} \text{Equation 4.9}

where $T$ is the temperature.
$K$ is the effective thermal conductivity and contains molecular and turbulent contributions. Thus

$$K = \frac{\mu_{eff} c_p}{Pr}$$

Equation 4.10

where Pr is the Prandtl number and $c_p$ is the specific heat at constant pressure.

The left side of equation 4.8 contains the accumulation and flux terms for the internal energy. The first term on the right describes the flow work, while the last term is the heat introduced from the dissipation of turbulence.

Since $\mu_{eff}$ is a function of $k$ and $\varepsilon$, two additional transport equations are introduced to achieve closure.

The $k$ Equation

Launder and Spalding (1972) modelled the evolution of turbulent kinetic energy with the following differential equation:

$$\frac{\partial \rho k}{\partial t} + \nabla \cdot (\rho k \vec{u}) = -\frac{2}{3} \rho k \nabla \cdot \vec{u} + \bar{\sigma} : \nabla \vec{u} + \nabla \left[ \left( \frac{\mu_{eff}}{Pr_k} \right) \nabla k \right] - \rho \varepsilon$$

Equation 4.11

$Pr_k$ is an empirical constant the value of which is shown in table 4.1. As with all the conservation equations, the left side of the equation comprises the accumulation and flux terms. As $\frac{2}{3} \rho k$ is considered the turbulent analogue to pressure, the first term on the right side represents the corresponding flow work rate (per unit volume). The second term on the right is related to the production of turbulence by shear forces. The second last term and the final term represent the self-diffusion of turbulent kinetic energy and the decay of turbulence respectively.
As shown by Reynolds (1980), the form of the $k$ equation can be derived by manipulation of the Navier-Stokes and the averaged Navier-Stokes equations. A similar manipulation is used to provide the form for the $\varepsilon$ equation.

The $\varepsilon$ Equation

A variation of Launder and Spalding’s equation used in KIVA to model the dissipation of turbulent kinetic energy follows:

$$\frac{\partial \rho \varepsilon}{\partial t} + \nabla \cdot (\rho \varepsilon \vec{u}) = \left( \frac{2}{3} c_{e1} - c_{e3} \right) \rho \varepsilon \nabla \cdot \vec{u} + \frac{c_{e1}}{k} \dot{\varepsilon} \nabla \cdot \vec{u} + \nabla \cdot \left[ \left( \frac{\mu_{\text{eff}}}{Pr_\varepsilon} \right) \nabla \varepsilon \right] - \frac{c_{e2}}{k} \rho \varepsilon$$

Equation 4.12

The quantities $c_{e1}$, $c_{e2}$, $c_{e3}$, $Pr_\varepsilon$ are empirical constants and are shown in table 4.1. The first term on the right side models length scale changes due to expansion. The effect of the second term on the right is that the dissipation rate increases with turbulence production from shear. The second last term represents the self-diffusion of energy dissipation, while the last term shows that the dissipation rate tends to diminish more rapidly with larger dissipation rates. Although the terms do not seem entirely intuitive, it is worth reiterating that they are mathematically based on the Navier-Stokes equations.

The original equations by Launder et al. were developed for incompressible flows. They were subsequently modified for compressible flow by Jones (1979) and El Tahry (1983).

4.2.2 Boundary Conditions

In the KIVA codes, three types of physical boundaries are classified: flow boundaries, rigid walls and periodic boundaries. Flow boundaries are not discussed, as there is no inflow or outflow in the current study.
Periodic boundaries can be imposed where longitudinal symmetry planes exist inside the cylinder. The symmetry planes chosen for the Squish-Jet combustion chambers used in this study longitudinally bisect each of the four squish-fence segments. As a result, instead of the whole combustion chamber only a quadrant of it was needed for flow modelling. The computational time was thus reduced fourfold. Figure 6.3 shows the computational mesh of a Squish-Jet chamber at a crank angle of 30 degrees BTDC. Even though only one longitudinal plane was adequate to model the Bowl-in-Piston chamber studied, a quadrant was also used for continuity with the other chambers.

Rigid wall boundary conditions are assigned with respect to velocity and temperature. For velocity, the non-slip condition was imposed. Thus, the velocity of the gas was set equal to the velocity of the wall. The $k$-$\varepsilon$ model was not used at rigid walls because the isotropy assumption is violated. Instead, the shear stress at the wall was computed by matching the velocity profile near the wall to the profile of the universal law of the wall. The wall stress needed to effect the match was subsequently used. The law of the wall is expressed in Amsden et al. (1989).

The temperature boundary conditions were introduced by setting a fixed wall temperature. The heat flux at the wall was determined by Reynolds' analogy of the wall stress.

There were two boundary conditions used for the $k$ and $\varepsilon$ equations. The first was that the gradient of $k$ normal to each wall was zero. The basis for the second boundary condition was that the distance, $y$, of the cell centroid to the neighbouring wall represents the characteristic macroscale in the cell. This distance was hence used to determine $\varepsilon$ near a wall using the relation:

$$
\varepsilon = c_{\mu\varepsilon} \frac{k^{3/2}}{y}
$$

Equation 4.13

where

$$
c_{\mu\varepsilon} = \sqrt{\frac{c_{\mu}}{\text{Pr}_\varepsilon (c_{\varepsilon 2} - c_{\varepsilon 1})}}
$$

Equation 4.14
4.2.3 Input Processing

The KIVA-3V hydro code (the flow solver code) requires two input files. The first is called ITAPE17 and contains the data representing the computational mesh. This file was created by the pre-processing code K3PREP, the input of which is one file (user-generated and called IPREP) that contains the general geometrical specifications of the cylinder.

The second input file to the hydro code is called ITAPE5 and it contains more than 200 parameters whose values are assigned by the user. The parameters include information such as initial conditions and constants that dictate the method of solution. Some of the more important ITAPE5 parameters and their functions and settings for the current study are shown in table 4.2.

4.2.4 Output Processing

As indicated in table 4.2, the output from KIVA was in GMV format. GMV stands for ‘General Mesh Viewer’ and it is a free computer program from Los Alamos National Laboratories designed for visualising vector and scalar field data typically produced by CFD codes. GMV required running on a separate computer platform to KIVA-3V, so another program called Tecplot was used to visualise the data instead. Code was written in Pascal to convert the GMV output format into a format that Tecplot accepts.
4.3 Tables

<table>
<thead>
<tr>
<th>Parameter Name</th>
<th>Value assigned</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lwall</td>
<td>1</td>
<td>Law-of-the-wall boundary condition activated</td>
</tr>
<tr>
<td>Ncaspec</td>
<td>8</td>
<td>Data is output at 8 different crank angles which are specified in the next line. -60, -30, -25, -20, -15, -10, -5, 0</td>
</tr>
<tr>
<td>Gmv</td>
<td>1</td>
<td>Output is produced in format for separate data viewer called ‘GMV’.</td>
</tr>
<tr>
<td>Cafin</td>
<td>180</td>
<td>Calculations end at 180 degrees ATDC</td>
</tr>
<tr>
<td>Bore</td>
<td>10</td>
<td>Bore is 10 cm</td>
</tr>
<tr>
<td>Stroke</td>
<td>10</td>
<td>Stroke is 10 cm</td>
</tr>
<tr>
<td>Squish</td>
<td>0.2</td>
<td>Squish clearance is 0.2 cm</td>
</tr>
<tr>
<td>Rpm</td>
<td>0.8e+3</td>
<td>Crank speed is 800 rev/min</td>
</tr>
<tr>
<td>Atdc</td>
<td>-180</td>
<td>Crank angle after top dead centre for time=0</td>
</tr>
<tr>
<td>Conrod</td>
<td>17.5</td>
<td>Connecting rod length is 17.5 cm</td>
</tr>
<tr>
<td>Swirl</td>
<td>0.0</td>
<td>There is no initial swirl in the cylinder</td>
</tr>
<tr>
<td>Tcylwl</td>
<td>293</td>
<td>The initial temperature for the cylinder wall is 293K</td>
</tr>
<tr>
<td>Thed</td>
<td>293</td>
<td>The initial temperature for the cylinder head is 293K</td>
</tr>
<tr>
<td>Tpistn</td>
<td>293</td>
<td>The initial temperature for the piston face is 293K</td>
</tr>
<tr>
<td>Turbsw</td>
<td>1.0</td>
<td>k-ε turbulence model is activated</td>
</tr>
<tr>
<td>tign</td>
<td>9.99e+9</td>
<td>No ignition occurs</td>
</tr>
<tr>
<td>Presi</td>
<td>1.01325e+6</td>
<td>Initial cylinder pressure in dynes/cm²</td>
</tr>
<tr>
<td>Tempi</td>
<td>293</td>
<td>Initial temperature in Kelvin</td>
</tr>
<tr>
<td>Tkei</td>
<td>0.0</td>
<td>No initial turbulent kinetic energy</td>
</tr>
<tr>
<td>Er</td>
<td>0.0</td>
<td>Initial equivalence ratio is zero</td>
</tr>
<tr>
<td>Mfracfu</td>
<td>0.0</td>
<td>Initial mass fraction of fuel is zero</td>
</tr>
</tbody>
</table>

Table 4-1: Standard k-ε turbulence model constants.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>c₁</td>
<td>1.44</td>
<td></td>
</tr>
<tr>
<td>c₂</td>
<td>1.92</td>
<td></td>
</tr>
<tr>
<td>c₃</td>
<td>-1.0</td>
<td></td>
</tr>
<tr>
<td>Prₖ</td>
<td>1.0</td>
<td></td>
</tr>
<tr>
<td>Prₑ</td>
<td>1.3</td>
<td></td>
</tr>
</tbody>
</table>

Table 4-2: Description of some input parameters set in ITAPE5.
5 EXPERIMENTAL RESULTS

The PIV and LDV measurements recorded by methods described in chapter 3 are documented in this chapter.

5.1 PIV Measurements

As stated in chapter 3, the PIV measurements for both Squish-Jet combustion chambers were always taken in the plane bisecting the squish fence. For the Bowl-in-Piston chamber, the PIV plane always bisected the squish volume. The PIV plane positions are shown in figures 5.1 to 5.3, while typical PIV images recorded in these planes are found in figures 5.4 and 5.5. Figure 5.1 also shows the polar coordinate system adopted in this study to describe directions and positions in a PIV plane. The velocity maps determined from the PIV images are in figures 5.6 to 5.37. Each velocity map is split in two—the right half shows a vector field, while the left half mirrors the right side and shows the contour lines of flow speed in the plane. The PIV images were recorded in 5 degree increments between 30 degrees BTDC and TDC. All experiments were done with a crank speed of 800 rev/min.

5.1.1 Mean Radial Velocity

The Bowl-in-Piston Chamber

From the velocity maps shown in figures 5.6 to 5.12 it is evident that the predominant velocity in the Bowl-in-Piston chamber is radially inward (squish) for all crank angles tested. As shown by the concentric contour rings, the velocity distribution is symmetrical about the longitudinal axis and the peak squish velocity tends to penetrate the bowl volume as TDC is approached. The squish velocity is zero at the centre of the chamber because of the flow symmetry. These trends are summarised in figure 5.13 (or figure 5.14) where it is also revealed that the peak mean radial velocity inside the PIV plane reaches a maximum at around 10 degrees BTDC. It is also worthy to note from figure 5.13 that at a radial position of 25 mm, a noticeable velocity gradient appears at crank angles in which the peak radial velocity has penetrated the bowl (10 degrees BTDC to
TDC). This is most likely because the radial velocity in the thin squish volume is much more responsive to the piston’s slow speed near TDC than it is for the larger bowl volume where recirculating flow is expected to have formed. The recirculating flow inside the bowl volume is inevitable based on continuity arguments. As shown in the following chapter, KIVA-3V also predicts this secondary flow.

Although figure 5.14 contains the same summary information as does figure 5.13, the mean radial velocities are shown from different perspectives. While figure 5.13 (‘type 1 summary’) shows a velocity ‘snapshots’ inside the chamber for a number of crank angles, figure 5.14 is a ‘type 2 summary’ that displays velocity histories for a range of positions. Figure 5.14 shows that for positions close to the bowl lip, the mean radial velocity moving inwards tends to peak at around 10 degrees BTDC. The peak is most prominent at the bowl lip (r=25 mm). Although the form of the curve at the lip resembles the ideal curve shown in figure 3.17, the velocity magnitudes differ significantly. For example, the peak radial velocity occurs at around 10 degrees BTDC for both graphs, but this peak is 35% lower in figure 5.14. The difference unquestionably exceeds the maximum measurement uncertainty of 0.2 m/s in the mean radial velocity. Measurement uncertainties are discussed later in this section.

The discrepancy between the ideal theory and observations may be attributed to the key simplifying assumption in the ideal theory—that the density inside the combustion chamber is uniform at any one time. Given that the peak Mach number inside the chamber is at most 0.03, this assumption may at first seem justifiable; however non-uniform heat transfer may exist, forming temperature (and density) gradients that reduce the squish velocity. Turbulence generation may also reduce the squish velocity due to transfer of kinetic energy.

Another possible reason for the discrepancy is that the pressure developed inside the cylinder may have caused enough strain on the piston and cylinder head to enlarge the squish clearance. The likelihood of this strain was increased as the gudgeon pin was made of nylon—a material used to prevent cylinder scuffing. Numerical tests run with the ideal squish model revealed that squish velocity is very sensitive to squish clearance. For example, a 0.5 mm increase in the 2 mm clearance reduced the peak squish velocity by 20%, whilst the compression ratio reduction was only 3%.
As explained in chapter 3, the O-rings used for the piston rings and head gasket provided a tight seal for each combustion chamber. Consequently, cylinder leakage was not considered a cause for the discrepancy in squish velocities.

The crank angle sensor was a US Digital optical encoder that emitted a 5V pulse each crank degree. By recording the pulsed encoder signal against time, the crank speed during UBCRiCM operation was determined. The maximum error observed at crank angles between 50 degrees BTDC and TDC was 25 rev/min. The corresponding relative error of 3% in crank speed was therefore dismissed as a cause for the velocity discrepancy.

Figure 5.15 shows the fractional uncertainty in a single PIV velocity measurement. The uncertainty is due to the finite spatial resolution in the PIV images and it applies to all combustion chambers tested. As a rule of thumb, this uncertainty is around 5% for velocities near 1 m/s and it sharply increases to about 10% for 0.5 m/s. The statistical uncertainty in the mean radial velocity for a range of positions in the Bowl-in-Piston chamber is shown in figure 5.16. As explained in chapter 3 this uncertainty depends on the variability in the data set used to find the mean. The statistical uncertainty in the mean is expressed as the 90% confidence interval and it is worth noting that at TDC this interval is about twice as large above the bowl as it is above the squish area. The difference is caused by larger variations in velocity detected above the bowl due to a suspected higher level of turbulence there. Poorer PIV image quality above the bowl cannot be blamed for the larger variations because the reverse was observed. In fact, the particles appeared less distinct above the squish area because the light sheet was very close to the head and piston at TDC and stray reflection from these surfaces was more apparent. More turbulence is expected in the piston bowl because of the shear gradient resulting from squish flow at the bowl lip. The high variability in measured velocity is also evident in the PIV velocity map for TDC shown in figure 5.12. In this figure, the velocity contour bands are broken and less defined than for other crank angles.

The Squish-Jet 1 Chamber

As noted in the PIV velocity maps in figures 5.19 to 5.25, the prominent feature for the Squish-Jet 1 chamber are the jets issuing from the squish grooves. The velocity is purely radial in the
centrelines of these jets and as with the squish in the Bowl-in-Piston chamber, the peak velocity in each jet approaches the bowl centre as the piston nears TDC. The figures also show that the jets are not entirely isolated, suggesting that air is squished over the top of the fence and into the PIV plane. The jets are more conspicuous as TDC is approached because the gap between the squish-fence and the cylinder head is narrower, thus restricting the over-flow. At the same time, the piston speed decreases and generates less squish.

Figure 5.26 is a type 1 summary of the radial velocities observed in the jet mid-line of the Squish-Jet 1 chamber. The progressive migration of the squish velocity peak is also seen in this graph. By the time TDC is reached, the velocity peak penetrates the bowl by 7mm. As with the Bowl-in-Piston chamber, the squish velocity is zero at the centre of the chamber because of the flow symmetry. The figure also shows that the velocity peak has a similar magnitude for all crank angles measured. An explanation for this is offered in the next section.

The type 2 summary shown in figure 5.27 indicates that the greatest variation (with respect to time) in squish velocity occurs at the jet opening (d=0 mm). At this location the squish velocity begins at 30 degrees BTDC with 2.7 m/s and decreases to 2.4 m/s at 10 degrees BTDC. After 10 degrees BTDC the piston is relatively stationary and produces less squish, thus explaining the sudden dip there in the curve. Conversely, at a location 5mm downstream from the jet opening the squish velocity increases gradually from 2.3 m/s to a peak of 3 m/s at 15 degrees BTDC. It is possible that the velocity peak at the jet opening occurs earlier than at the 5mm location because a finite amount of time is needed for the velocity peak generated at the jet opening to travel 5mm downstream. The sudden velocity dip does not occur at the 5mm location because there is no direct exposure to the cause of squish, but rather, to the consequences of its production (i.e., flow recirculation inside the bowl). At the 10 mm position, squish speed is generally lower because of its proximity to the chamber centre.

It is also worthy to note that the peak squish velocity at the jet opening is around 10 degrees advanced relative to the peak for the Bowl-in-Piston chamber. This may be related to the PIV measurement error due to stray light reflection there. The stray light was reflected from the bowl edge, which was stationary in the PIV plane; thus, the velocity tended to be underestimated. The band of light (which is visible in figure 5.5) tended to be brighter as TDC was approached. Despite painting the groove edge black, the light band could not be entirely eliminated.
The only remaining groove in the PIV images was not used for velocity measurements because it had a more extensive bias error. The remaining groove, which was the lowest one in the PIV images, had a large shadow zone that was most likely produced by unavoidable light refraction in the lower left squish fence. This groove also had a bright band at the edge. The shadow and the bright band can be seen in figure 5.5. As evidenced in most of the PIV results, the abnormal illumination tended to reduce the squish velocities in the lower groove.

At the piston bowl, the PIV image quality for the Squish-jet 1 chamber was better than for the Bowl-in-Piston at TDC because the laser sheet was further away from the cylinder head. Nevertheless, figure 5.28 and in particular figure 5.30 show that the 90% confidence interval for the mean velocity is relatively high approximately 5mm in from the jet opening. Since the image quality cannot be attributed to the increased variability, it is likely that the level of turbulence (to which the 4 micron particles partly respond) close to the jet opening increases as TDC is approached. The existence of this increased turbulence is indicated by the appearance (in PIV images like figure 5.31) of coherent particle patterns having seed densities that are less uniform than other flow regions. The abnormal particle distribution seems to have been caused by turbulence generated when circulating flow inside the bowl crossed paths with the squish flow. Furthermore, the out-of-plane motion of the particles caused by the cross-flow increases the chance of local velocity errors and variability.

The Squish-Jet 2 Chamber

The PIV velocity maps for the Squish-Jet 2 chamber are shown in figures 5.32 to 5.38. The velocity trends for this chamber resemble those for the Squish-Jet 1 chamber. One noticeable difference, however, is that the squish velocities are generally higher in the Squish-Jet 2 chamber. For example, as indicated in the type 1 summary of figure 5.39, the highest squish velocity is 4.2 m/s and it occurs at 10 degrees BTDC. This represents a 40% increase on the maximum observed for the Squish-Jet 1 chamber. As indicated in equation 3.5, the increase is attributed to the smaller squish volume and larger bowl volume. At the same time the narrower squish grooves in the Squish-Jet 2 chamber make the squish flow area smaller, thus further enhancing the squish velocity.
As evidenced in Fig. 5.39 another difference between the Squish-Jet chambers is that the spatial peak in squish velocity for the Squish-Jet 2 chamber is more dependent on crank angle. For example, the peak squish velocity is 3.8 m/s at 30 degrees BTDC. It then rises to 4.3 m/s at 10 degrees BTDC and drops off again to 3.4 m/s at TDC. This rise and fall of the squish peak is similar for the Bowl-in-Piston chamber and it suggests that there is greater dissipation of bulk kinetic energy for these high squish chambers once the maximum squish has formed. This may be attributed to the higher levels of turbulence generated by greater squish flow.

As expected, the type 2 summary shown in figure 5.40 contains similar trends in mean squish velocity to those for the Squish-Jet 1 chamber. Nevertheless, in keeping with the supposition that the turbulence intensity is higher in Squish-Jet 2, a more sudden decay in squish is seen after 10 degrees BTDC at a point 5mm away from the jet opening.

It is also worth noting from figure 5.41 that the 90% confidence interval for the mean radial velocity is generally higher up to 12 mm downstream from the jet opening. The peak confidence interval once again occurs at TDC, but for the Squish-Jet 2 chamber this peak is around 5mm further inward than for the Squish-Jet 1 chamber. This is reflected in the PIV images because the abnormal particle seeding is detected further downstream as well. Therefore, it is possible that the recirculating flow in the Squish-Jet 2 chamber impinges on the squish jet at a position closer to the chamber centre, thus generating more turbulence there as well.
5.1.2 Mean Tangential Velocity

The Bowl-in Piston Chamber

It is quite evident from the PIV velocity plots for the Bowl-in-Piston chamber that the mean tangential velocity component for all radii is insignificant with respect to the radial component. As shown in the summary in figure 5.17, the insignificance holds for all crank angles. Indeed, the uncertainty for the tangential components usually exceeds their measured values.

Once more, the statistical uncertainty of the mean tangential velocity was found to be larger above the bowl than above the squish area. As shown in figure 5.18, this trend is particularly noticeable at TDC when the turbulence intensity is expected to be higher.

The Squish-Jet 1 Chamber

The PIV velocity plots for the Squish-Jet 1 chamber show that the tangential component of the mean velocity is not trivial near the edges of the squish jets. Nevertheless, as shown by the velocity plots and the summary in figure 5.29, the mean tangential component along the midline of the squish jets is insignificant. Once more, the uncertainty of the tangential component has a magnitude similar or greater than the measured value.

As indicated in figure 5.30, the statistical uncertainty is noticeably larger close to the jet opening at TDC than at other locations and times. For example, the 90% confidence interval tends to be around 0.05 m/s for all jet midline locations at 20 degrees BTDC whereas it leaps to about 8 times this value close to the jet opening at TDC. This disparity exceeds the uncertainty in the standard deviation (between 60% and 100%) due to the limited number of PIV tests. The turbulence and increased out-of-plane particle flow seem responsible for this large uncertainty. Moreover, the confidence interval for the tangential velocity is about three times that for the radial velocity. This implies that at TDC and 5mm away from the jet opening, the rms tangential velocity is three times the radial rms velocity.
Although the limited number of PIV measurements per test condition (typically 10) caused a large 90% confidence interval for the rms velocities, the uncertainty was not large enough to invalidate the discrepancy between the radial and tangential components. The 90% confidence interval for the rms velocity was around 60% and it was calculated using equation 3.19.

The Squish-Jet 2 Chamber

The Squish-Jet 2 chamber featured very similar trends in mean tangential velocity to the Squish-Jet 1 chamber. As shown in figure 5.42 the tangential velocity was also insignificant relative to the mean radial component, while figure 5.43 also shows that the 90% confidence interval is pronounced near the jet opening at TDC.

From comparison of the 90% confidence intervals shown in figures 5.41 and 5.43 the tangential rms velocity at TDC near the squish groove appears larger than the radial component. The statistical uncertainty in the rms velocity, however, is too high for one to be adequately sure of this discrepancy.

5.2 LDV Measurements

Unlike the PIV measurements, the LDV measurements were made at one fixed point inside one combustion chamber – the Squish-Jet 2 chamber. Figure 5.44 shows the measurement location. At 20 degrees BTDC, the location lies along the jet midline and 5mm in from the jet opening. The crank angles corresponding to the window centres of the LDV measurements were 25 degrees to 5 degrees BTDC with 5-degree increments. This range of crank angles was smaller than the range used for PIV because of oscilloscope storage limitations.
5.2.1 Mean Radial Velocity

Even though the LDV measurement location doesn’t follow the piston as in PIV, the mean LDV radial velocity’s dependence on the crank angle is similar in form to PIV observations and to ideal squish theory.

Comparison of the mean velocity curve in figure 5.45 with the d=5mm curve in figure 5.40 shows that the two differ by less than the experimental uncertainties. This is not surprising because the radial positions are identical while the axial positions differ by approximately 4mm at most. The theory used to calculate the uncertainties is presented in Chapter 3 of this document.

5.2.2 Mean Tangential Velocity

The mean tangential velocity at the LDV point is shown as a function of crank angle in figure 5.46. Comparison between figures 5.46 and 5.42 reveals that at 20 degrees BTDC, the mean tangential velocity at the LDV measurement point is close to the corresponding PIV value of -0.2 m/s. Even though the difference between the two curves falls within the measurement uncertainties, the LDV tangential velocity differs somewhat from the PIV values at 15 degrees and 10 degrees BTDC. For example, at 10 degrees BTDC, the mean PIV tangential velocity is around -0.1m/s while the corresponding LDV value is -0.8 m/s. The discrepancy may have occurred because the LDV measurement point is deeper in the bowl where it is exposed to slight swirl. The swirl, which can be attributed to slight yet noticeable machining errors in the wedge shaped squish grooves or to a slightly slanted piston during compression, is less prominent in the PIV plane where there is direct exposure to the radial squish jet.

5.2.3 RMS Radial and Tangential Velocities

Figures 5.45 and 5.46 show the evolutions (at the LDV measurement point) of rms radial and tangential velocities respectively. It is evident from the graphs that both rms components increase as TDC is approached. Furthermore, for crank angles after 15 degrees BTDC the radial
(downstream direction of jet) rms velocity component is typically about 50% larger than the tangential (cross-stream direction of jet) rms component. The difference is larger still for crank angles prior to 15 degrees BTDC. Downstream rms velocity components have also been observed by others to be larger than the cross-stream components in steady flow round turbulent free jets. For example, Panchapakesan and Lumley (1993) have reported downstream to cross-stream rms velocity ratios of 1.37. Similar results have been reported by Fukushima, Aanen and Westerweel (2000). The observation means that the turbulence in the squish-jets is anisotropic. As discussed in chapter 6, this places suspicion on the general isotropy assumption made in KIVA-3V.
Figure 5.1: PIV measurement plane (plane A) for the Bowl-in-Piston combustion chamber. The PIV laser sheet coincided with this plane. Dimensions are in millimetres. The polar coordinate system in any plane parallel to the cylinder head is also described.
Figure 5.2: PIV measurement plane (plane B) for Squish-Jet 1 combustion chamber. The PIV laser sheet coincided with this plane. Dimensions are in millimetres.
Figure 5.3: PIV measurement plane (plane C) for Squish-Jet 2 combustion chamber. The PIV laser sheet coincided with this plane. Dimensions are in millimetres. LDV data is recorded in plane D.
Figure 5.4: Raw PIV image pair captured for Bowl-In-Piston chamber at 25° BTDC. The lower photograph was taken 170 µs after the upper one. Displacements of particle groups in the first image are determined by spatial cross correlation analysis with the second image.
Figure 5.5: Raw PIV image pair captured for Squish-Jet 1 chamber at 20° BTDC. The lower photograph was taken 220 μs after the upper one.
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Figure 5.6: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.7: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.8: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.9: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.10: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.11: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.12: Mean PIV velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.13: Type 1 summary of PIV measurements in plane A for the Bowl-in-Piston chamber. The radial distribution of the mean squish speed is shown for a set of crank angles.

Figure 5.14: Type 2 summary of PIV measurements in plane A for the Bowl-in-Piston chamber. The evolution of the mean radial squish speed is shown for a set of radial positions.
Figure 5.15: The fractional error of a particle group's velocity (determined by PIV analysis) depends on the laser pulse separation time ($D_t$) and the particle group's actual speed.

Figure 5.16: The radial distribution (in plane A) of the 90% confidence interval for the mean radial velocity in the Bowl-in-Piston chamber. The distribution is shown for two different crank angles.
Figure 5.17: The mean tangential velocity (in plane A) for the Bowl-in-Piston chamber is insignificant relative to the mean radial velocity for all observed crank angles and positions. The tangential velocity is positive when its direction (viewed from the cylinder head) is clockwise.

Figure 5.18: The radial distribution (in plane A) of the 90% confidence interval for the mean tangential velocity in the Bowl-in-Piston chamber. The distribution is compared for two different crank angles.
Figure 5.19: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.20: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.21: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.22: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.23: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.24: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.25: Mean PIV velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.26: Type 1 summary of PIV measurements along midline of jet in plane B for the Squish-Jet 1 chamber. The radial distribution of the mean squish speed is shown for a set of crank angles.
Figure 5.27: Type 2 summary of PIV measurements in plane B for the Squish-Jet 1 chamber. The evolution of the mean radial squish velocity is shown for a set of radial positions.

Figure 5.28: The radial distribution (in plane B) of the 90% confidence interval for the mean radial velocity in the Squish-Jet 1 chamber. The distribution is shown for two different crank angles.
Figure 5.29: The mean tangential velocity for the Squish-Jet 1 chamber is insignificant relative to the mean radial velocity for all observed crank angles and positions. The tangential velocity is positive when its direction (viewed from the cylinder head) is clockwise.

Figure 5.30: The radial distribution (in plane B) of the 90% confidence interval for the mean tangential velocity in the Squish-Jet 1 chamber. The distribution is shown for two different crank angles.
Figure 5.31: Example of PIV image taken of plane B in Squish-Jet 1 chamber at TDC. The abnormal circular region of particles near the left jet opening suggests cross-flow of re-circulating bowl currents into the squish-jet flow.
Figure 5.32: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.33: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.34: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.35: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.36: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.37: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 5.38: Mean PIV velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 5.39: Type 1 summary of PIV measurements along midline of jet in plane C for the Squish-Jet 2 chamber. The radial distribution of the mean squish speed is shown for a set of crank angles.
Figure 5.40: Type 2 summary of PIV measurements in plane C for the Squish-Jet 2 chamber. Evolution of the mean radial squish velocity is shown for a set of radial positions.

Figure 5.41: The radial distribution (in plane C) of the 90% confidence interval for the mean radial velocity in the Squish-Jet 2 chamber. The distribution is shown for two different crank angles.
Figure 5.42: The mean tangential velocity for the Squish-Jet 2 chamber is insignificant relative to the mean radial velocity for all observed crank angles and positions. The tangential velocity is positive when its direction (viewed from the cylinder head) is clockwise.

Figure 5.43: The radial distribution (in plane C) of the 90% confidence interval for the mean tangential velocity in the Squish-Jet 2 chamber. The distribution is shown for two different crank angles.
Figure 5.44: The crosses indicate the LDV measurement location. The location is fixed relative to the cylinder head. Dimensions are in millimetres. The arrow for \( V_t \) defines the positive direction of the tangential velocity whilst the arrow for \( V_r \) defines the radial velocity's positive direction.
Figure 5.45: Evolution of mean and rms radial velocity (determined by LDV) with crank angle. The measurement location is specified in figure 5.44. A positive mean radial velocity is toward the centre of the combustion chamber.

Figure 5.46: Evolution of mean and rms tangential velocity (determined by LDV) with crank angle. The measurement location is specified in figure 5.44. A negative tangential velocity is counterclockwise when viewed from the cylinder head.
6 NUMERICAL RESULTS AND COMPARISON WITH EXPERIMENTS

In this chapter, the computational meshes representing the geometries of the three squish chambers tested are described. The flow predictions from the CFD code (KIVA-3V) that used the computational meshes are also presented. KIVA-3V and its operation are described in chapter 4. The KIVA-3V flow predictions are compared with the experimental observations, in order to assess the code’s validity.

6.1 Mesh Generation and refinement

The input file named ‘IPREP’ that was referred to in chapter 4 is used by ‘K3PREP’ (the separate mesh generating module of KIVA-3V) to produce the computational mesh for the relevant combustion chamber. IPREP is created by the user in a format acceptable to K3PREP and it contains general information about geometric blocks of which the chamber is composed. For example, the Bowl-in-Piston chamber is composed of three geometric blocks. The first block represents the bowl of the chamber, the second block is for the cylindrical volume above the bowl and the third block represents the squish volume. The outlining geometry of each block and its desired number of divisions (per coordinate axis) are specified in IPREP. Important common coordinates of each block are also specified to link the blocks.

A grid convergence exercise was conducted on the Squish-Jet 1 combustion chamber to determine the optimum number of mesh cells. The mean cell size was progressively reduced from coarse dimensions until a negligible change in peak squish velocity (approximately 1%) was calculated by KIVA-3V. Consequently, the final number of hexahedral cells representing one quadrant of the Squish-Jet 1 chamber was close to 40,000. A similar number of cells was used for the other two chambers tested. The computational mesh for each chamber is shown in figures 6.1 to 6.3.

The initial time step was set to 1.042 μs. Each subsequent time step was automatically adjusted by KIVA-3V to ensure a stable solution whilst keeping the computation time low.
6.2 Presentation format

In order to facilitate comparison with PIV results, most of the KIVA results are presented for planes (‘cutplanes’) which intersect the combustion chamber and are parallel to the cylinder head. The positions of the cutplanes matched the placement of corresponding PIV laser sheets. As with PIV results, each cutplane is divided by a mirror line to the right of which a velocity vector plot is shown and to the left of which a contour plot of squish speed is shown. These velocity cutplanes are found between figures 6.4 and 6.28. In addition, cutplanes that fall on the Squish-Jet 2 chamber’s longitudinal axis and bisect two opposed squish grooves are presented. As these longitudinal cutplanes contain the LDV measurement location, comparison of KIVA with LDV results is made possible. These cutplanes are shown in figures 6.31 to 6.37.

6.3 Mean Radial Velocity in Radial-Azimuthal Planes

6.3.1 The Bowl-in-Piston Chamber

The velocity maps that were predicted by KIVA-3V in plane A for the full range of crank angles are shown in figures 6.4 to 6.10. Each velocity map exhibits pure squish flow and as with the PIV results, the squish peak is seen to penetrate the bowl volume as TDC is approached. Comparison of the type 1 summaries shown in figures 6.11 and 5.13 reveals that the experimentally observed trends in mean squish velocity are very close to the KIVA-3V output. Despite the matching trends the PIV squish velocities tend to be lower than the predicted values, especially near 15 degrees BTDC when squish velocities are larger in general. For example, at this crank angle the peak squish measured by PIV at the bowl rim is approximately 17% lower than the KIVA-3V prediction. A similar discrepancy occurs at other locations in the Bowl-in-Piston chamber. Figure 6.40 shows a comparison in squish velocity histories between KIVA-3V and PIV results. The comparison is made for two radial positions in plane A of the Bowl-in-Piston chamber: 25 mm and 15 mm out from the centre.
In order to systematically determine the causes of this discrepancy the sources of error were grouped into the following three categories: (a) incorrect generation of flow being modelled, (b) PIV measurement errors, and (c) numerical modelling errors. The categories are shown in the tree diagram of figure 6.39. Parameters affecting category (a) can be identified by doing a mass balance in the squish volume. Using the same notation found in Appendix A, the mass balance can be expressed as follows:

$$\rho_2 v_s A_s = \rho_1 A_p V_p + V_1 \frac{d\rho_1}{dt} - \dot{m}_l$$

Equation 6.1

where the new term $\dot{m}_l$ denotes the mass loss rate due to leakage and crevice filling. The mass transfer by molecular diffusion is ignored because it is insignificant relative to the squish flow.

Acknowledging that the squish flow is turbulent, the appropriate velocities and densities used in equation 6.1 can be split into their mean and fluctuating components. If the result is then Reynolds averaged the following relation is produced:

$$\overline{v}_s = \frac{\overline{\rho}_1 A_p v_p - V_1 \frac{d\overline{\rho}_1}{dt} + \dot{m}_l - \overline{\rho}_2 v_s A_s}{\overline{\rho}_2 A_s}$$

Equation 6.2

where overbars represent means and the primes denote fluctuations from the means.

It is worth noting that the last term in the numerator of equation 6.2, which arose from the averaging operation, represents the turbulent transport of mass from the squish volume. It follows from equation 6.2 that the mean squish velocity $\overline{v}_s$ at a given crank angle depends on initial and boundary conditions along with time-dependent geometric parameters ($V_1, A_p, \rho_1$) and mass losses ($\dot{m}_l$). The density and turbulence terms are not included as separate variables because they too depend on the above parameters (through the energy equation and the ideal gas law). The geometric parameters depend on variables such as squish gap, squish area, bowl
diameter, stroke, crank speed and crank angle. Careful examination of these independent variables shows that squish gap and crank speed are more prone to error. These two variables along with the mass loss are shown in figure 6.39 under category A. They must have uncertainties that are small enough to generate a well-defined squish velocity history.

There are two relevant sources of PIV measurement uncertainty and as shown in category B of figure 6.39, they are seed particle errors and PIV processing. The seed errors are related to distribution, visibility and flow-following ability of the particles.

The number of possible errors that can arise in the numerical modelling is exhaustive. An example may be one misplaced negative sign. Nevertheless, the errors can be separated into two general groups: incorrect programming of the differential equations to be solved and a suspect physical basis of these differential equations. Typical culprits for the latter group are turbulence models. For example, the \( k-\varepsilon \) model contains the often-unrealistic assumptions of isotropic turbulence and turbulence equilibrium. Heat transfer models are also error prone because they often depend on turbulence models and generalised empirical constants. Category C of figure 6.39 shows the types of numerical errors. The peak pressure inside the Bowl-in-Piston chamber was measured at 1.48 MPa absolute with a compression tester. At 1.43 MPa, the predicted pressure was only 3.4% lower than measured and this may indicate that the heat transfer model is satisfactory. Nevertheless, this general test does not confirm local heat transfer predictions.

In the current study, a number of causes (of the disparity between numerical and experimental squish results) identified at the bottom level of figure 6.39 could be eliminated because of their relative insignificance. For example, the uncertainty in crank speed was neglected because of its low relative value of 3%.

Leakage was also considered negligible because of the rigorous yet successful static helium tests that were described in chapter 3. Further evidence that both leakage and crevice-flow were insignificant was the absence of PIV velocity vectors pointing toward the cylinder wall. In order to account for the 25% squish velocity discrepancy at 15 degrees BTDC, for example, the outward flow that had to be observed was about 0.75 m/s. No such flow was detected in the PIV results.
The possibility of particle flow infidelity did exist if the nylon PIV particles had a tendency to agglomerate. Nevertheless, this cause was also dismissed because a finer oil mist that was tried yielded similar velocity results to the nylon particles. The smaller size of the olive oil particles in the mist was readily confirmed by PIV image inspection. The oil mist was not regularly used for the PIV measurements because of the difficulty in repeating particle concentration and uniformity. In this study, a method was developed to adequately distribute the seed particles; however, occasionally the visibility of the particles was compromised due to either spurious light reflections or lack of light caused by refraction. For example, the PIV images of the Bowl-in-Piston chamber in figure 5.4 show light reflections at the left extreme of the bowl rim. The reflections clearly outshine the light scattered from the local particles. Therefore, although many error sources related to seed particles were made trivial, they could not be entirely discounted. Thus, these error sources appear bolded in figure 6.39.

PIV processing errors were also considered negligible causes for the discrepancy because manually measured particle displacements closely matched PIV results.

The $k$-$\varepsilon$ turbulence model is not only responsible for determining the evolution of the specific turbulent kinetic energy ($k$), but also of the specific turbulence dissipation rate ($\varepsilon$). Underestimation of the production of $k$ can lead to overestimation of the mean squish velocity. Furthermore, overestimation of $\varepsilon$ inside the bowl rim can lead to higher temperatures and specific volumes, thus causing overpredicted mean squish velocities.

The remaining causes for the discrepancy that could not be eliminated without further investigation are: an excessive squish gap, inadequate heat transfer modelling and erroneous programming of the differential equations. These possible causes are bolded in figure 6.39.

Although the squish clearance was accurately set when the piston was stationary (with a depth gauge through a duplicate cylinder head), there was no assurance that the clearance was maintained during UBCRICM operation. As a future exercise, the operative squish gap could be checked by photographing the side of the piston at TDC during UBCRICM motion. The resulting image can be compared with a reference image taken when the piston is stationary at TDC. The validity of the heat transfer model in KIVA-3V could also be checked by comparison with heat flux measurements in select wall locations.
Since the velocity discrepancy (between KIVA and PIV) for the Bowl-in-Piston is independent of position, the squish gap error seems a more likely cause. As explained in section 5.1.1, squish velocities are sensitive to squish gap error. The other possible causes depend more heavily on position in the combustion chamber. For example, production of turbulent kinetic energy is greater in high shear regions and PIV errors occurred in unique locations with abnormal lighting.

Fansler and French (1987) conducted squish and swirl tests in a motored engine and they too reported discrepancies from computed values of similar proportions. The authors cited causes such as heat transfer, blow-by, viscous dissipation, ring-land volumes and uncertainty in the actual TDC clearance height.

It is emphasised that despite the noticeable discrepancy between measured mean squish velocity and KIVA predictions, the squish velocity trends with respect to crank angle and position are very similar.

6.3.2 The Squish-Jet 1 Chamber

Figures 5.26 and 6.20 show the type 1 summaries of the Squish-Jet 1 chamber for PIV measurements and KIVA-3V predictions respectively. Both figures reveal a progressive migration of the peak squish velocity into the bowl centre as TDC is approached. This is expected because as TDC is approached, squish generation greatly diminishes while the recirculating flow inside the bowl persists. The recirculating flow is inevitable from continuity considerations. Examples of the recirculating flow appear in figures 6.31 to 6.37. Nevertheless, the magnitudes of the corresponding peaks in radial velocity differ between KIVA and PIV results. While the peak squish velocities for the PIV measurements stay at around 3 m/s for all crank angles tested, the peaks for KIVA-3V rise to a maximum of 3.5 m/s at 15 degrees BTDC and then suddenly drop to 2.6 m/s at TDC. This drop as TDC is approached may result from local overestimation by KIVA-3V of the turbulence production inside the bowl. The overestimation converts more of the kinetic energy of the bulk recirculating flow into turbulent kinetic energy. Another explanation for the drop predicted by KIVA-3V is the miscalculation of the axis of the recirculating flow near TDC. For example, if the axis of recirculation is predicted to be shallower in the bowl, the PIV plane is closer to the axis where radial velocities are smaller.
The latter explanation for the squish peak drop predicted by KIVA-3V at TDC is likely because it is consistent with the LDV observations discussed in section 6.5. As explained in this section, the miscalculation of the axis location could well be attributed to erroneous eddy viscosity predictions in the turbulence model.

Figure 6.41 shows a type 2 summary comparison between KIVA-3V predictions and PIV results for the squish velocity histories for a range of jet midline positions. The KIVA squish value at the jet opening peaks near 13 degrees BTDC. Ideal squish theory concurs with this peak angle prediction. It is also apparent that the PIV squish velocity at the jet opening (d=0 mm) deviates from the KIVA-3V prediction much more noticeably than at other radial positions. This disproportionate local discrepancy implies that an error in the squish clearance is not the sole cause. Further evidence that squish gap error is not primarily responsible is that the squish velocity reaches its peak 10 degrees before expected. A more likely cause is the undesirable band of light reflected at the bowl edge of the squish groove. Since the light band was bright and stationary (in the PIV plane), the velocity in its vicinity was undervalued. This band of unwanted light can be seen in both PIV images in figure 5.5 and it is more prominent as TDC is approached.

One less possible cause for the exceptional discrepancy in squish between PIV results and KIVA at the jet opening is the code’s underestimation of the local production of turbulent kinetic energy. This cause is less likely because a similar local discrepancy was not observed at the bowl rim of the Bowl-in-Piston chamber.

6.3.3 The Squish-Jet 2 Chamber

Figures 5.39 and 6.29 show the type 1 summaries of the Squish-Jet 2 chamber for PIV measurements and KIVA-3V predictions respectively. As expected, the smaller squish volume in the Squish-Jet 2 chamber yields accordingly higher squish velocities. For instance, the maximum measured squish velocity for the Squish-Jet 2 chamber was 42% higher than for the Squish-Jet 1 chamber. Comparison of the summaries is similar to the comparison for the Squish-Jet 1 chamber. For example, the maximum squish value from KIVA-3V was also 17% higher than
from PIV observations. The migration of the peak squish velocity toward the chamber centre is seen for both KIVA and PIV and the maximum penetration of the peak is also 7 mm for both chambers. The comparison of the type 2 summaries shown in figure 6.42 once again shows the exceptional local discrepancy between the measured and predicted squish velocity at the jet opening. For example, the PIV value for the jet opening (d=0mm) at 15 degrees BTDC was 34% lower than the KIVA value, while at d=10mm, the corresponding discrepancy was only 9%. The squish at the groove also reaches a peak about 10 degrees before the more credible KIVA prediction of 12 degrees BTDC. As was the case for the Squish-Jet 1 chamber, the local discrepancy can be attributed to the spurious light reflections at the edge of the jet opening.

6.4 Mean Tangential Velocity in the Radial-Azimuthal Planes

As expected from symmetry, the mean tangential velocity component predicted by KIVA-3V along jet midlines for the squish-jet chambers is zero. The same holds for all positions in the Bowl-in-Piston chamber. The generalisations are reflected in the PIV results. For this reason, no tangential velocity graphs for KIVA-3V are presented.

6.5 Mean Radial Velocity in the Radial-Axial Plane

Figures 6.31 to 6.37 show the KIVA velocity plots for the radial-axial plane (plane D) bisecting two opposing squish grooves in the Squish-Jet 2 chamber. The plots cover the full range of crank angles tested and each plot indicates the LDV measurement point by a cross. As dictated by symmetry, the velocities into the page (tangential velocities) are zero. The generation of squish and the resulting recirculating flow are evident in the vector plots. Despite the eventual dwindling of the squish velocity at the jet opening, the relative persistence of the recirculating flow is apparent.

Figure 6.38 shows a summary of KIVA-3V predictions for the LDV measurement point. For comparison, the LDV results are included on the same set of axes. The upper curves in the figure show the temporal evolution of the mean radial velocity while the lower ones are of the rms radial velocity. The rise and fall of the mean radial velocity so characteristic of squish flow is
demonstrated by both KIVA and LDV readings; however, contrary to the PIV cases the squish velocity from KIVA-3V is significantly lower than the measured LDV value as TDC is approached. It appears that the greater depth of the LDV measurement point (for crank angles after 20 degrees BTDC) is causing this contradiction. The term 'depth' refers to the axial distance inside the chamber from the cylinder head.

The influence of the depth on KIVA predictions can be appreciated by observing the mean radial velocity 5 mm in from the jet opening at two different depths: the PIV plane and the LDV point. At 10 degrees BTDC, the LDV point is 2.9 mm deeper than the PIV plane. Examination of figures 6.42 and 6.38 shows that the predicted radial velocity at the shallow point (PIV) is 4.8 m/s while it is only 2.9 m/s at the deep point (LDV). At the same time, the measured values were 4.1 m/s and 4.0 m/s for the shallow and deep points respectively. This means that while the measured radial velocity gradient in the axial direction is negligible, KIVA predicts a more profound gradient. The large discrepancy is seen for all crank angles after 10 degrees BTDC and it cannot be accounted for by measurement error.

The location predicted by KIVA-3V of the recirculating flow in the bowl may have a large role in causing the above discrepancy. It is possible for example, that KIVA may have predicted a shallower circulation region than in reality, thus causing more prominent squish velocity gradients in the axial direction near the squish groove. The sharp velocity gradients found in the plane D vector maps (figures 6.33 to 6.37) appear to support this explanation. The shallow axis of recirculation probably arises from underestimation of the turbulent viscosity ($\mu_t$) in the high shear region beneath the squish jet. This smaller viscosity causes less effective momentum transfer from the squish jet into deeper regions of the bowl.
6.6 RMS Velocity at LDV measurement point

The rms velocities from KIVA-3V that are shown in figure 6.38 were calculated using the specific turbulent kinetic energy \( (k) \) output and the assumption in the code that the turbulence is isotropic. The following equations illustrate the direct connection between the isotropic rms velocity and \( k \):

\[
k = \frac{u_{\text{rms}}^r}{2} + \frac{u_{\text{rms}}^t}{2} + \frac{u_{\text{rms}}^a}{2} = \frac{3}{2} u_{\text{rms}}^{\text{iso}}
\]

Equation 6.3

\[
\therefore u_{\text{rms}}^{\text{iso}} = \sqrt{\frac{2k}{3}}
\]

Equation 6.4

where

\( u_{\text{rms}} \) is the rms velocity fluctuation

and

the superscripts \( r \), \( t \) and \( a \) denote the radial, tangential (azimuthal) and axial (longitudinal) directions respectively. The superscript \( \text{iso} \) refers to the isotropic value.

As shown in the experimental results, the rms velocity measured at 5 degrees BTDC in the radial direction was about 50% larger than the tangential value. The difference was larger for early crank angles. The turbulence was therefore considered anisotropic at the LDV measurement point. This is consistent with observations made by Iijima and Bracco (1987) near the bowl rims of bowl-in-piston chambers. Nevertheless, an isotropically equivalent value was sought for each tested crank angle in order to compare it with the corresponding KIVA-3V value. This equivalent rms value for LDV was found after first estimating the total measured turbulent kinetic energy per unit mass, \( k_{\text{LDV}} \).
As shown in the following equation, the estimation was made by assuming that the sum of the turbulent energies in the radial and tangential directions is two thirds of the total.

\[
k_{LDV} = \frac{3}{2} \left( \frac{U_{rms}^2 + U_{t rms}^2}{2} \right)
\]

Equation 6.5

Substituting \( k_{LDV} \) into equation 6.4 yielded the equivalent isotropic rms velocity for LDV measurements.

The equivalent isotropic rms velocities for LDV are plotted in figure 6.38 along with the corresponding KIVA-3V values. Comparison of the two histories shows that while the rms velocity for KIVA is relatively steady at a high level (approximately 1 m/s), the measured value progressively increases from 0.26 m/s at 25 degrees BTDC to 1.3 m/s at 5 degrees BTDC.

The relative steadiness in the predicted rms velocity means that for the crank angles tested, the modelled dissipation of \( k \) (and its net transport away from the LDV point) erroneously balances the production of \( k \).

The general overprediction of the rms velocity (and hence \( k \)) at the LDV point also implies that \( \varepsilon \) is overestimated. This is because the turbulent viscosity \( (\mu_t) \), which is proportional to \( k^2/\varepsilon \), is suspected of being underpredicted by KIVA-3V. The reason for the suspicion was explained in section 6.5.

6.7 Turbulent Kinetic Energy

Despite large statistical uncertainties, the PIV results for each chamber tested exhibited larger mean velocity variations inside the bowl than in the squish volume. This was reflected in the confidence interval distributions shown in figures 5.18, 5.30 and 5.43 and the phenomenon was attributed to higher turbulence intensities in the bowls.
As shown in figures 6.43 to 6.45, KIVA-3V results also show this spatial trend. The trend is more pronounced in the Squish-Jet chambers than in the Bowl-in-Piston chamber even though the latter produces the highest squish velocities. The larger turbulent kinetic energy in the Squish-Jet chambers is attributed to more regions of shear flow created by the squish jets. Squish-Jet 2 chamber is believed to have more turbulent kinetic energy than Squish-Jet 1 chamber because the shear gradients near the jets are larger as a result of the faster squish flow.
6.8 Figures

Figure 6.1: Computational grid used with KIVA-3V for the Bowl-in-Piston combustion chamber. Crank angle is 30° BTDC. Grid dimensions are in millimetres. The z-axis is in the axial (or longitudinal) direction.
Figure 6.2: Computational grid used with KIVA-3V for the Squish-Jet 1 combustion chamber. Crank angle is 30° BTDC. Grid dimensions are in millimetres. The z-axis is in the axial (or longitudinal) direction.
Figure 6.3: Computational grid used with KIVA-3V for the Squish-Jet 2 combustion chamber. Crank angle is 30° BTDC. Grid dimensions are in millimetres. The z-axis is in the axial (or longitudinal) direction.
Figure 6.4: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.5: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.6: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.7: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.8: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.9: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.10: Mean KIVA-3V velocity plot in plane A for the Bowl-In-Piston chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.11: Type 1 summary of KIVA-3V predictions in plane A for the Bowl-in-Piston chamber. The radial distribution of the mean squish speed is shown for a set of crank angles.

Figure 6.12: Type 2 summary of KIVA-3V predictions in plane A for the Bowl-in-Piston chamber. The evolution of the mean radial squish speed is shown for a set of radial positions.
Figure 6.13: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.14: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.15: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.16: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.17: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.18: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.19: Mean KIVA-3V velocity plot in plane B for Squish-Jet 1 chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.20: Type 1 summary of KIVA-3V predictions along midline of jet in plane B for the Squish-Jet 1 chamber. The radial distribution of the mean squish velocity is shown for a set of crank angles.

Figure 6.21: Type 2 summary of KIVA-3V predictions in plane B for the Squish-Jet 1 chamber. The evolution of the mean radial squish velocity is shown for a set of radial positions.
Figure 6.22: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.23: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.24: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.25: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.26: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.

Figure 6.27: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.28: Mean KIVA-3V velocity plot in plane C for Squish-Jet 2 chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. Out-of-plane velocities are not shown.
Figure 6.29: Type 1 summary of KIVA-3V predictions along midline of jet in plane C for the Squish-Jet 2 chamber. The radial distribution of the mean squish velocity is shown for a set of crank angles.

Figure 6.30: Type 2 summary of KIVA-3V predictions in plane B for the Squish-Jet 2 chamber. Evolution of the mean radial squish velocity is shown for a set of radial positions.
Figure 6.31: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 30° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber’s squish fence and a cross indicates the LDV measurement point.

Figure 6.32: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 25° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber’s squish fence and a cross indicates the LDV measurement point.
Figure 6.33: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 20° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber’s squish fence and a cross indicates the LDV measurement point.

Figure 6.34: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 15° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber’s squish fence and a cross indicates the LDV measurement point.
Figure 6.35: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 10° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber's squish fence and a cross indicates the LDV measurement point.

Figure 6.36: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 5° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber’s squish fence and a cross indicates the LDV measurement point.
Figure 6.37: Mean KIVA-3V velocity plot in plane D for Squish-Jet 2 chamber. Crank angle is 0° BTDC. Crank speed is 800 rev/min. The contour plot only shows the radial velocity component. Out-of-plane mean velocities are zero. The hidden lines (dashed) represent the chamber's squish fence and a cross indicates the LDV measurement point.
Figure 6.38: Temporal evolution of mean and isotropic rms radial velocity at the LDV measurement point. The measurement location is shown by the crosses in the sketch beside the graph. The solid curves are KIVA-3V predictions and the dashed curves are from LDV measurements.
Figure 6.39: Classification and identification of possible causes for discrepancies in squish velocity between CFD predictions and PIV results.
Figure 6.40: Temporal evolution of mean radial velocity at two radial locations in Bowl-in-Piston chamber. Results from KIVA-3V and from PIV measurements are shown for comparison. Crank speed is 800 rev/min.

Figure 6.41: Temporal evolution of mean radial velocity at three radial locations in Squish-Jet 1 chamber. Results from KIVA-3V and from PIV measurements are shown for comparison. Crank speed is 800 rev/min.
Figure 6.42: Temporal evolution of mean radial velocity at three radial locations in Squish-Jet 2 chamber. Results from KIVA-3V and from PIV measurements are shown for comparison. Crank speed is 800 rev/min.
Figure 6.43: Specific turbulent kinetic energy (k) contour plot coupled with mean velocity map for plane A in Bowl-in-Piston chamber. Results are from KIVA-3V. Crank angle is TDC and crank speed is 800 rev/min.

Figure 6.44: Specific turbulent kinetic energy (k) contour plot coupled with mean velocity map for plane B in Squish-Jet 1 chamber. Results are from KIVA-3V. Crank angle is TDC and crank speed is 800 rev/min.
Figure 6.45: Specific turbulent kinetic energy (k) contour plot coupled with mean velocity map for plane C in Squish-Jet 2 chamber. Results are from KIVA-3V. Crank angle is TDC and crank speed is 800 rev/min.
7 CONCLUSIONS AND RECOMMENDATIONS

In keeping with the objectives outlined in section 1.3, numerical and experimental flow observations were made inside three different IC engine combustion chambers that generate squish. As a result, a greater understanding of the complex flow processes inside plain squish and Squish-Jet chambers was gained. The validity of flow predictions from the CFD code, KIVA-3V, was also assessed for the three chambers. The assessment was made by comparing mean and rms velocity measurements with corresponding values from KIVA-3V. Measurements were made by PIV and LDV and in order to reproduce the initial and boundary conditions set in KIVA-3V, the UBCRICM (University of British Columbia Rapid Intake and Compression Machine) was used. The combustion chambers did not contain fuel and they were transparent to allow optical access for the laser measurements. A compression ratio of 9.1:1, a squish clearance of 2 mm and a crank speed of 800 rev/min were used for all tests. One of the combustion chambers was a plain Bowl-in-Piston type while the remaining two were Squish-Jet chambers. Although all the combustion chambers had the same bore (100 mm) and bowl diameter (50 mm), the Squish-Jet chambers featured a 10 mm wide fence that surrounded the bowl rim. Each fence also had four equi-spaced grooves from which air jets squirted inwards as TDC was approached. Squish-Jet 2 chamber had a shorter fence with narrower squish grooves than Squish-Jet 1 chamber. For the two Squish-Jet chambers PIV measurements were made in a radial-azimuthal plane that bisected the squish fence. The PIV plane for the Bowl-in-Piston chamber was also radial-azimuthal and it bisected the squish volume. The LDV measurements were made in the Squish-Jet 2 chamber at a fixed point that was 5mm in from a squish groove. The fixed point was centred in front of the squish groove when the piston was 20 degrees BTDC.

7.1 Conclusions

As a result of the study described, the following major conclusions were drawn:

1) Among factors such as a particle’s flow following ability and conspicuity, PIV image resolution should be considered when determining the size of seed particles used. Fine particles used to follow turbulent scales that the PIV system cannot resolve are redundant. As
a result of these considerations, the PIV seed particles chosen for this study were 4 micron nylon microspheres.

2) The spatial resolution of typical LDV systems is high enough that it does not require consideration when determining the size of seed particles. As a result, the LDV seed particles chosen for this study were 2 micron silicone resin microspheres. These particles were able to follow the finest of turbulence scales.

3) Root mean square (rms) velocities determined by PIV were used as a rough measure of turbulence intensity because of the technique’s poor spatial resolution and large statistical uncertainty. Conversely, rms velocities determined by LDV were most accurate because of this technique’s high resolution and the low statistical uncertainty that resulted from abundant velocity measurements. Mean velocity measurements made by PIV were superior to LDV measurements because global measurements were possible whilst maintaining similar accuracy to LDV.

4) Measurements and simulations in planes above the bowls of the three test chambers generally showed an inward mean flow that peaked at approximately 10 degrees BTDC. After this crank angle the flow rapidly subsided so that at TDC only a residual inward flow was observed above the bowl region. By continuity arguments, this indicated a persisting recirculating flow inside each bowl—a phenomenon also predicted by KIVA-3V. In the Squish-Jet chambers the inward flow was predominantly in the form of jets, while the flow was axisymmetric in the Bowl-in-Piston chamber.

5) For each combustion chamber tested, PIV tests showed that the statistical error in the mean velocity for both radial and tangential directions near TDC are generally greatest near the bowl rim. This implied that the turbulence level was higher at these locations. This is expected because high levels of shear that generate turbulence exist at the rims. The results are consistent with observations from other researchers (Fansler and French, 1987).

6) With a few local exceptions, the peak mean radial velocities measured in the PIV planes were generally about 17% lower than the KIVA-3V predictions, regardless of chamber type. A more suspected cause for this general discrepancy is a slight expansion of the squish
clearance. The expansion may have resulted from strain on the cylinder head retainer and the
nylon gudgeon pin when high in-cylinder pressures were developed close to TDC.

7) The exceptional discrepancies between KIVA-3V predictions and measurements of the mean
radial velocity occurred only at the jet openings. At these locations, measured values were
less than KIVA-3V values by as much as 42%. These discrepancies were attributed mainly to
unwanted light reflections from the bowl rim. Excessive production of turbulent kinetic
energy that KIVA-3V could not model was a less likely cause because the velocity gradients
in the jet midlines were not considered significant. A further reason for placing less emphasis
on this cause was the absence of the local discrepancy in the Bowl-in-Piston chamber. For
the Bowl-in-Piston chamber, the mean velocity at the bowl rim was calculated from velocity
samples taken at points all around the bowl rim. Consequently, the error from the local light
reflection was mitigated by this symmetrical averaging process.

8) Experimental results from the Squish-Jet 2 chamber indicated a milder squish velocity
gradient near the jet (in the axial direction) than KIVA-3V predictions. For example, at 10
degrees BTDC and 5 mm away from the jet opening, the measured mean squish gradient was
one twentieth the predicted value of 0.66 (m/s)/mm. The gradient was calculated between the
PIV plane and the LDV point (which was 1.4 mm below the squish groove). This significant
discrepancy implies that KIVA-3V predicts a shallow axis for the toroidal recirculating flow
inside the piston bowl. This misrepresentation most probably arises from underestimation of
the turbulent viscosity ($\mu_t$) in the high-shear region beneath the squish jet. The smaller
viscosity causes less effective momentum transfer from the squish jet into deeper regions in
the bowl.

9) Although the greatest squish velocities were observed in the Bowl-in-Piston chamber, this
chamber was predicted to have the lowest turbulence intensity. Some indication of this trend
was also apparent in the PIV measurements. The largest turbulence intensity was predicted in
the short fenced squish-jet chamber (Squish-Jet 2 chamber). This was also indicated by the
measurements.

10) Experimental measurements showed that the turbulence at the LDV point was anisotropic
near the bowl because the radial rms velocity was generally about 50% larger than the
tangential component. Although higher degrees of isotropy are expected in most other regions inside the combustion chamber, this finding invalidates the isotropic assumption made in KIVA-3V.

11) The specific turbulent kinetic energy ($k$) measured at the LDV test point was generally higher than the KIVA value. As turbulent viscosity is assumed in KIVA-3V to be proportional to $k^2/\varepsilon$ (where $\varepsilon$ is the dissipation rate of $k$), the hypothesis that $\mu_t$ is underpredicted means that KIVA-3V overestimates $\varepsilon$. Despite the continual rise in the measured value of $k$ as TDC was approached, the predicted value remained relatively steady. This is consistent with the proposition that KIVA-3V overestimates $\varepsilon$ near the jet.

To summarise, numerical and experimental observations showed the potential for squish-jet chambers to accelerate combustion and to improve thermal efficiency and emissions in reciprocating internal combustion engines. Moreover, the general flow trends are well predicted by KIVA-3V. For example, the symmetrical inward migration of the squish velocity peaks, the expected recirculation inside the piston bowl and the spatial variation in turbulence intensity were all present in the KIVA-3V results. The characteristic temporal trends for squish velocity were also present. When optimising combustion chamber shape for faster burning, the relevant changes in these trends (as different chambers are simulated) are more important than a consistent prediction error.

### 7.2 Recommendations for future work

Although the general flow trends predicted follow the experimental observations, there is potential for KIVA-3V to more closely match measurements by improving the code and the experiments. In addition, further testing is useful to check the validity of other important quantities generated by KIVA-3V. The improvements and testing methods are proposed in the following summary:

1) The squish clearance during UBCRICM operation warrants inspection because it was suspected of generally lowering the peak radial velocity. The inspection could be done by photographing the side of the piston at TDC during the compression stroke. The resulting image may then be compared with a reference image taken when the piston is stationary at
TDC. If an incorrect squish gap is confirmed, the gudgeon pin should be replaced by a steel one that has teflon caps to prevent cylinder scuffing. If this measure is not corrective, the cylinder head retainer plate should be replaced with one that is suitably thicker.

2) More antireflective coatings need to be tested in order to reduce unwanted light coming off the bowl rims of all the pistons tested. The successful coating would eliminate the local error in PIV velocity.

3) The storage oscilloscope should have extra active memory installed in order to hold LDV samples covering a crank angle range that matches the larger PIV test range.

4) Although a comprehensive understanding of the mean flow inside the combustion chambers was gained from the extensive PIV work done in this study, similar detail is useful for rms velocities and turbulence scales. These quantities are important because combustion rate is strongly related to them. Moreover, these quantities are crucial validity indicators of the turbulence model in KIVA-3V. Although very time-consuming, they can be measured with special two-point LDV measurement systems. The extra detail can also reveal the extent of validity of the KIVA code’s isotropic turbulence assumption.

5) Simultaneous measurements of three orthogonal velocity components inside the piston bowls could also be made to reveal axial-radial cross-sections of the recirculating flow structure. The cross-sections may then be compared with corresponding velocity maps from KIVA-3V; thus enabling validity checks of other important predicted flow parameters (e.g., vortex core locations). The 3-D velocity measurements could be made with the more experimental stereoscopic PIV technique.

6) The validity of the heat transfer model in KIVA-3V could also be checked by comparison with heat flux measurements in select wall locations. Observation of the temperature field in the combustion chamber can also be very useful in this validation exercise. This may be achieved by seeding the flow with temperature sensitive particles. A fast-response pressure transducer could also be installed so that comparison of the measured pressure histories with predictions can be used to check the general validity of the code’s heat transfer model.
7) Since the degree of in-cylinder turbulent anisotropy that has been measured by others is not pronounced enough to deem the $k$-$\varepsilon$ model invalid (Fraser and Bracco, 1987), refining the model for squish jet chambers seems more justifiable than replacing it. This option is also preferred as the $k$-$\varepsilon$ turbulence model is relatively lenient on computer resources. To begin with, the constants in the $\varepsilon$ equation can be calibrated to lower the turbulent dissipation rate. The constant in the $k$ equation ($Pr_k$) may also be calibrated so that predicted $k$ histories may approach the measured trends.
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