PIV MEASUREMENTS WITHIN A WATER ANALOG ENGINE

by

Daniel Karl Fetter

A thesis submitted in conformity with the requirements for the degree of Masters of Applied Science
Graduate Department of Mechanical and Industrial Engineering
University of Toronto

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Abstract

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2000

An optical water analog engine with two symmetric valves has been designed and constructed to study the mixing characteristics within a piston driven reciprocating flow field. A timing system has been designed for use with Particle Image Velocimetry (PIV) to capture single exposure PIV images at pre-determined crank angles. Velocity fields were found at in the centre of the flow cell and near both walls at 3.5 valve diameters and 7.5 jet diameters away from the top of the flow cell. Two large recirculation zones were found to exist early on in the intake stroke and started to break down by 180 Crank Angle Degrees. Significant large scale mixing was seen to exist at the 3.5 valve diameter location in the centre of the flow cell. At the 7.5 valve diameter location in the centre there was relatively little large or small scale mixing. Near the walls, a boundary layer was found. The boundary layer was seen to break down near the end of the intake stroke.
Dedication

To my parents, Karl and Rita Fetter.
Acknowledgements

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# Nomenclature

<table>
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<th>Definition</th>
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<tbody>
<tr>
<td>CAD</td>
<td>Crank Angle Degrees</td>
</tr>
<tr>
<td>$U_E$</td>
<td>Ensemble averaged velocity</td>
</tr>
<tr>
<td>N</td>
<td>Number of measurements</td>
</tr>
<tr>
<td>k</td>
<td>Sample number</td>
</tr>
<tr>
<td>$\bar{u}$</td>
<td>Instantaneous velocity</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Crank angle</td>
</tr>
<tr>
<td>$\bar{U}_i$</td>
<td>Lancaster mean velocity</td>
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<tr>
<td>$\bar{U}_i$</td>
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<tr>
<td>$\bar{U}_E$</td>
<td>Time averaged ensemble velocity</td>
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<tr>
<td>PR</td>
<td>Particle response time</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
</tr>
<tr>
<td>D</td>
<td>Diameter</td>
</tr>
<tr>
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<td>Viscosity</td>
</tr>
<tr>
<td>V</td>
<td>Velocity</td>
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<tr>
<td>d</td>
<td>Fringe distance</td>
</tr>
<tr>
<td>t</td>
<td>Time</td>
</tr>
<tr>
<td>U</td>
<td>X component of velocity</td>
</tr>
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<td>$R_{aa}$</td>
<td>Reynolds stress component aa</td>
</tr>
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<td>Nusselt number</td>
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<tr>
<td>Pr</td>
<td>Prandlt number</td>
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<tr>
<td>P</td>
<td>Pressure</td>
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<tr>
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<td>Definition</td>
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</tr>
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<td>Mach number</td>
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<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>St</td>
<td>Strouhal number</td>
</tr>
<tr>
<td>$U_P$</td>
<td>Piston velocity</td>
</tr>
<tr>
<td>$M$</td>
<td>Magnification factor</td>
</tr>
<tr>
<td>CCD</td>
<td>Charge coupled device</td>
</tr>
<tr>
<td>MP</td>
<td>Mass percentage of seed particles</td>
</tr>
<tr>
<td>$P_a$</td>
<td>Number of pixels in direction a</td>
</tr>
<tr>
<td>MI</td>
<td>Measurement interval</td>
</tr>
<tr>
<td>$R(s)$</td>
<td>Cross correlation co-efficient</td>
</tr>
<tr>
<td>$I_1(s)$</td>
<td>Gray scale matrix of image one at position s</td>
</tr>
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<td>$I_2(s)$</td>
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<td>$R_D(s)$</td>
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<td>$R_F(s)$</td>
<td>Fluctuation of the background noise correlation</td>
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<tr>
<td>dev</td>
<td>Vector deviation</td>
</tr>
<tr>
<td>$u'$</td>
<td>Fluctuating velocity</td>
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<tr>
<td>$d_i$</td>
<td>Particle displacement in direction i</td>
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<tr>
<td>$e_a$</td>
<td>Error of a</td>
</tr>
<tr>
<td>$\sigma_a$</td>
<td>Standard deviation of a</td>
</tr>
<tr>
<td>$O(a)$</td>
<td>Order of a</td>
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</table>
Chapter 1

Introduction

1.1 Background

Due to stricter emissions standards, lean burn operating conditions in engines are more common as there is a reduction of nitric oxide, hydrocarbon and carbon monoxide concentrations in the exhaust gas mixture. Unfortunately, as air/fuel ratio decreases, combustion instabilities and misfire occur. Both the combustion duration and cycle to cycle variations increase (Heywood, 1988).

It has been noted that increasing the level of turbulence in engines leading up to the time of combustion decreases the burn duration and cyclic variation and increases the fuel/air mixing (Heywood, 1988). Modern multivalve Spark Ignition (SI) engines with 4 valves per cylinder have good characteristics in terms of power output and exhaust gas constituents and are flexible in terms of creating various flow patterns within the cylinder. The valve configurations typically create two large scale vortex motions: one which is perpendicular to the cylinder axis called tumble and another which is parallel to the cylinder axis called swirl (see Figures 1.1 and 1.2).

These large scale vortex motions speed up due to the conservation of angular mo-
**Figure 1.1**: Tumbling flow in a Cylinder

**Figure 1.2**: Top View of Swirling flow in a Cylinder
momentum during the compression stroke (Reeves et al., 1994). As the piston approaches Top Dead Center (TDC) in the compression stroke, the organized tumble and swirl break down into small scale turbulence at the time of ignition. The increased turbulence level at TDC (when the tumble and swirl have broken down) increases the burn rate and the mixing of fuel/air, betters the flame propagation and decreases the cyclic variability of the mean flow. As a result, a leaner fuel can be used. Although tumble and swirl decrease the cyclic variability, the in-cylinder flow motion has been shown to be non-stationary (Lancaster, 1976; Catania & Mittica, 1985).

A significant amount of research has been done to quantify and qualify the in-cylinder flow field, since it is so strongly linked with power output and exhaust gas concentration. Researchers have mainly used techniques such as Hot-wire Anemometry (HWA), Laser Doppler Velocimetry (LDV), Flow Visualization and Particle Image Velocimetry (PIV).

HWA is an intrusive measurement technique that allows continuous time velocity information at a single point in space to be collected. Hotwires are heated cylinders and the fluid passing over it in a windtunnel or engine cylinder acts to cool the hotwire (see Figure 1.3).

![Figure 1.3: A Typical Hotwire Probe](image)

HWA calibration relates temperature change to fluid velocity. There is no di-
rectional dependence (e.g., a flow moving at $10\frac{m}{sec}$ or $-10\frac{m}{sec}$ will cause the same temperature change). The thermal time constant of the wire (the time lag between the change in fluid velocity and the change in wire temperature) must be small and is related to the thickness of the wire and also to the maximum measurable frequency; the smaller the thermal time constant, the higher the maximum measurable frequency.

Lancaster (1976) used Hot Wires in a production engine and investigated the use of ensemble averaging a series of time resolved data in a non-stationary flow field. Ensemble averaging is defined as,

$$U_E = \frac{1}{N} \sum_{k=1}^{N} (\bar{u}(\theta, k))$$  \hspace{1cm} (1.1)

where $U_E$ is the ensemble averaged velocity and $\bar{u}(\theta, k)$ is the instantaneous velocity at crank angle $\theta$ and measurement $k$. Lancaster (1976) compared the ensemble averaged velocities to instantaneous velocities and found that they were significantly different. Lancaster (1976) concluded that although the ensemble averaged instantaneous velocity accurately defines the mean velocity in stationary flows, it did not accurately define the mean velocity in non-stationary flows (such as in-cylinder flows).

Lancaster (1976) suggested that the mean velocity could be better represented by separating the engine cycle into 45 Crank Angle Degree (CAD) intervals. Lancaster (1976) added a constant quantity to the ensemble average in each interval; the constant in each 45 CAD interval was different. The constant quantity was defined as the difference between the time average of the instantaneous velocity and the time average of the ensemble average:

$$\bar{U}_i(t) = U_E(t) + \bar{U}_i - \bar{U}_E$$  \hspace{1cm} (1.2)
\( \bar{U}_i \) and \( \bar{U}_E \) were found by integrating over 45 CAD:

\[
\bar{U}_i = \frac{1}{45^\circ} \int_{\theta_i}^{\theta_1 + 45^\circ} \tilde{u}(\theta, k) d\theta
\]  \hspace{1cm} (1.3)

\[
\bar{U}_E = \frac{1}{45^\circ} \int_{\theta_i}^{\theta_1 + 45^\circ} U_E(t) d\theta
\]  \hspace{1cm} (1.4)

Typically, HWA has a maximum cutoff frequency of about 5000 Hz (Sullivan, 1999) which limits the measurable turbulent frequencies. Because HWA is intrusive, measurements are limited to regions near the top of the piston chamber where the piston will not destroy the hotwire. As mentioned above, stationary HWA is directionally ambiguous and cannot extract flow direction in reversing flows.

LDV is a non-intrusive technique that can be used to obtain instantaneous velocities at a single point in space at significantly higher frequencies (e.g., on the order of MHz) and without directional ambiguity. Small highly reflective seed particles are used to follow the flow motion. The ability of the seed particles to follow the flow motion is determined by the particle slip velocity,

\[
|v - u| = \frac{\rho_P D_P^2 |\hat{v}|}{36 \rho v}
\]  \hspace{1cm} (1.5)

where \( \rho_P \) is the seed particle density, \( D_P \) is the particle diameter and \( v \) is the kinematic viscosity of the measurement fluid. The slip velocity proportional to the density of the particle, but also to the diameter squared. This is important because metallic particles with a large density reflect a significant amount of light. They will also follow the flow motion well if their diameter is small (typical LDV seed particle diameters are between 0.2 \( \mu m \) and 15 \( \mu m \)).
The properties of the seed particles vary depending on the flow environment. In fired production engines, Titanium Dioxide is typically used because it has a high melting point (Sullivan, 1999). Because Titanium Dioxide is dense, a diameter of about 0.2\( \mu m \) is used in air flows (Sullivan et al., 1999).

As the seed particles pass through an interference pattern created by the crossing of two laser beams, a velocity measurement is recorded. The resultant velocity is determined by,

\[ V = \frac{d}{t} \]  

(1.6)

where \( V \) is the instantaneous fluid velocity, \( d \) is the distance between fringes and \( t \) is the period of the signal (defined as the average time between fringe crossings).

The time between velocity measurements is random. It occurs when a seed particle passes through the measurement volume; the higher the seed density, the more closely spaced in time the measurements are. An upper limit exists on the number of particles that can be added to the fluid as high seed particle concentrations alter the fluid motion. The upper limit depends on the fluid and the seed particle properties.

Thus, LDV can be used to extract flow velocity and direction, as well as operate at higher frequencies than HWA.

Liou and Santavicca (1985) performed LDV measurements in a motored production engine and offered another definition of mean and turbulent velocity within an engine. They Fourier transformed the instantaneous velocity for an entire engine cycle, using the highest frequency of the ensemble average as a cut-off, took the inverse Fourier transform. The low frequency components were defined as the mean velocity and the high frequency components were defined as the turbulence. Because there
was one mean velocity and one turbulent velocity per cycle, their averaging technique was termed cyclic averaging.

Sullivan et al. (1999) recognized that there was a problem with both ensemble and cyclic averaging in non-stationary flows. Sullivan et al. (1999) compared ensemble, cyclic and wavelet-based averaging of crank angle resolved LDV velocity measurements within SI engines. They found that wavelet analysis identifies turbulence more clearly than ensemble or cyclic averaging within non-stationary flows. Because engine flows are non-stationary and quasi-periodic, ensemble averaging results in an over-prediction of turbulence levels when compared to cyclic and wavelet based averaging. Cyclic averaging implies a constant convection velocity because of the fixed cut-off frequency that separates the mean and turbulent portions of the flow. Cyclic averaging is not appropriate in non-stationary reciprocating flows since the time and length scales change with crank angle and between cycles (Sullivan et al., 1999). Sullivan et al. (1999) showed that the wavelet transformed instantaneous velocities were repeatable on a crank angle basis and had a wide range of scales which were important to the mean flow.

Figure 1.4 shows the effect of ensemble averaging a flow field that varies from cycle to cycle. Two cycles are shown on Figure 1.4. The two cycles clearly have a different mean velocity. When the two cycles are ensemble averaged and the turbulence is calculated, a large portion of the mean velocity is added to the turbulent quantity. The effect of ensemble averaging is to overpredict the amount of turbulence in non-stationary flows.

Figure 1.5 shows the effect of cyclic averaging a flow field. Cyclic averaging calculates a mean velocity for each cycle as shown in Figure 1.5. The ensemble average of all of the cyclic means equals the ensemble mean. However, because there is a different mean quantity determined for each cycle, the total turbulence is reduced
Figure 1.4: Ensemble Averaging A Flow that Varies between Cycles
Figure 1.5: Cyclic Averaging A Flow
with respect to ensemble averaging.

Wavelet averaging uses the energy of the flow field to determine a criteria for defining the mean velocity. Sullivan et al. (1999) determined that high energy events are repeatable within engines between cycles. Using the high energy events as a foundation for separating the mean and turbulent quantities, Sullivan et al. (1999) were able to clearly separate the mean and turbulent portions of the flow field.

LDV only captures information at discrete times and does not give a continuous time velocity measurement. Thus, the calculation of Power Spectral Densities becomes computationally intense and is limited to the largest time separation within the data set.

Flow visualization studies in a production engine by Arcoumanis et al. (1987) showed that there were significant variations in the phase and amplitude of large scale vorticity fields (tumble and swirl). Catania and Mittica (1989) indicated that the variations in large scale vorticity could contribute substantially to the cycle to cycle variations of the mean flow. In order to determine the effect and existence of flow structures such as tumble and swirl within an engine, spatially resolved measurements were necessary. In order to obtain spatially resolved data with HWA, a series of hotwires spaced very closely together (a rake) is used. However, the HWA rake is still limited by the maximum resolvable frequency and is intrusive. In order to obtain spatially resolved data with LDV, a number of LDV systems are needed; one system is needed for each point in space and each LDV system is expensive.

PIV is a technique that can determine a two component velocity field over a two dimensional region in space. A typical PIV velocity field can be seen in Figure 1.6 taken from the present work. X and Y are the dimensions in mm of the measurement area, and each arrow represents a velocity vector. Seed particles, similar to those used in LDV are used to track the flow. A thin laser sheet is used to illuminate
It is determined by one of two methods: cross-correlation or autocorrelation. If both lasers are separated into interrogation areas, the average seed particle displacement within the interrogation areas are determined by the displacement of a group of five particles in an interrogation region. The average seed particle displacement within the interrogation areas are separated into interrogation areas or interrogation regions with several particles imaged onto either photographic paper or onto a CCD camera array. The images of the seed particles at two instances in time. The particle positions are recorded by the images.
are pulsed during the same frame, autocorrelation is used. If there is one laser pulse on image frame one and another laser pulse on image frame two, cross-correlation is used. In Figure 1.7, a cross-correlation would be used to find \( \Delta X \) and \( \Delta Y \) since there are two images of the same group of five particles.

Flow direction is resolved with cross-correlation, but autocorrelation results in a 180° directional ambiguity. Tang and Sullivan (2000) compared a standard cross-correlation algorithm to an autocorrelation algorithm in a numerically simulated reversing flow field. Tang and Sullivan (2000) found that the cross-correlation algorithm resulted in 22 times fewer rejected vectors than the autocorrelation algorithm.

The lasers are pulsed at predetermined instances in time, therefore the velocities at each interrogation area can be found by,

\[
U = \frac{\Delta X}{\Delta t} \tag{1.7}
\]

\[
V = \frac{\Delta Y}{\Delta t} \tag{1.8}
\]

where \( U \) and \( V \) are the average particle velocities in the X and Y direction, \( \Delta X \) and \( \Delta Y \) are the average particle displacement in the X and Y direction (see Figure 1.7) and \( \Delta t \) is the time between laser pulses.

By changing the camera's field of view, it is possible to obtain information about large scale structures such as tumble and swirl (Trigui et al., 1994) as well as small scale flow characteristics such as turbulent production or dissipation (Davis, 1999).

The accuracy of PIV with respect to LDV and HWA is dependent on the spatial
resolution of PIV as well as to the flow field in question and the correlation algorithm. Westerweel et al. (1996) found that cross-correlated PIV and LDV results agreed to within 1% of the true mean velocity defined by a direct numerical simulation of a fully developed turbulent pipe flow. However, in highly reversing flows, if an autocorrelation method is used, the difference between LDV and PIV can be as high as 40% (Diodati et al., 1993). Diodati et al. (1993) found that HWA and LDV agreed to within 10% in a turbulent jet. The disagreement was attributed to the inability of the HWA to resolve the high turbulence level found within the turbulent jet.

1.2 PIV measurements in cyclic flows

This section summarizes previous PIV and Flow Visualization work in cyclic flows.

With water instead of air-fuel as the working fluid, it is possible to greatly reduce the velocities involved in a piston chamber experiment. This is done with Reynolds and Strouhal number matching described in Section 2.2. Furthermore, there are many tracer particles such as polyamide seeds ($\phi \approx 5 - 50\mu m$), hollow glass spheres ($\phi \approx 10\mu m$) and silver coated hollow glass spheres ($\phi \approx 10\mu m$) that are available to track the flow. Flow visualization was used as a first attempt at identifying some of the flow structures present in reciprocating piston driven flow.

Arcoumanis et al. (1987) visualized the flow field of an optical plexiglass engine with fluid being injected at the center of the valve. The flow structure with injection resulted in a second tumbling vortex between the piston and cylinder wall. This was due to the jet interacting with the piston face.

Ekchian and Hoult (1979) investigated pathlines by following the motion of tracer particles that were injected into a cylindrical, plexiglass piston chamber at different crank angles and at different speeds. A qualitative comparison of flow features
suggested that the flow was more repeatable when tumbling vortices were present.

Khalighi and Huebler (1988) visualized pathlines with a continuous 2-Watt argon ion laser in a dual intake valve optical water analog engine operating at 65 RPM. They noted that the in-cylinder flow field during the induction process was dominated by the generation of strong jet flows which precipitated into organized vortical motions along the engine cylinder late in the induction stroke. They found that the jet initially moved along the cylinder wall, deflected off of the wall and piston face and formed two large recirculation regions between the valves and piston face.

Khalighi (1990) later determined that the inlet valve configuration strongly affected the in-cylinder flow. The valve lift and orientation created a complex fluid motion combining both tumble and swirl. When one valve was opened more than the other, a stronger tumbling motion would occur. The increase in tumble was due to the increased fluid velocity through one non-centered valve.

PIV was later used as a means to extract velocity fields from production engines. A limitation of extracting PIV measurements in an engine is the need for high resolution and large field of view. Until recently, the only method to achieve this was with 35 mm cameras. Thus, in order to operate in a high speed environment, it is necessary to expose both images on the same frame. Because both images are exposed on one frame, the direction of flow cannot be determined unless image shifting (Reuss, 1993) or 2 different colored lasers are used (Nino et al., 1992). With these two methods, the flow direction can only be resolved in flows that are not strongly reversing. Image shifting is used to add a constant displacement to the second exposure so that all of the particles in the second exposure have moved in the same direction. In this way, the particle displacements can be found with an autocorrelation algorithm and the direction of particle motion is known. The determine the true particle displacement, the constant displacement is removed from the calculated one and the velocity
direction can be resolved unambiguously.

Nino et al. (1992) compared results between LDV and a 2 color PIV system in a production engine. Even though both lasers were pulsed on the same frame, the direction of the particle motion was discernable because the exposures could be differentiated based on color. They found that the ensemble averaged tangential velocities measured with the LDV system and the 2 color PIV system were within 10% for 0 and 20 Crank Angle Degrees (CAD), but were within 30% at 340 CAD. They did not indicate why the disagreement increases at 340 CAD. The increased disagreement was likely due to the inability of 2 color PIV to effectively track the large flow reversals that were found at 340 CAD.

Reeves et al. (1996) took PIV measurements in a production engine to determine the breakdown of tumble and swirl during compression (180 to 360 CAD). Their work also used double exposed images and was subject to a directional ambiguity because they did not use 2 different colored lasers (Nino et al., 1992) or image shifting (Reuss, 1993). They found that there was good correlation between the broad features of the PIV results and the LDV and Hotwire measurements performed by other researchers in a similar engine geometry, although the similar work was not cited specifically. They also found that the flow was characterized by the formation of large scale vortices (tumble and swirl) which were shown to persist through the majority of the compression stroke. Reeves et al. (1996) noted that a cross-correlation algorithm was necessary for significant measurements in cyclic flows, as the directional ambiguity is removed even in strongly reversing flows.

Rouland et al. (1997) used PIV in a four valve four stroke single cylinder research engine with cross-correlation. Their field of view was approximately 55mm by 68mm in the tumble plane and 60mm by 60mm in the swirl plane. There was a spatial separation of approximately 0.62mm between velocity vectors which demonstrated
that PIV is effective in obtaining small scale flow characteristics. Their work showed that the cycle resolved bulk flow and the ensemble averaged flow are greatly different; the cyclic variations artificially increase the ensemble velocity fluctuations. This was particularly evident in the centres of large vortices that vary in position from cycle to cycle.

Trigui et al. (1994, 1996) used 3-D PTV in a 4 valve pent roof engine with water as the working fluid. Their results showed that if both intake valves were activated, a very strong tumbling motion and two weaker counter rotating cross tumble vortices were apparent. When one valve was deactivated, the flow field would exhibit a distorted tumbling vortex and a more organized (but weak) cross tumble vortex. PTV image collection rates limited the measurements to the end of the intake stroke at Bottom Dead Center (BDC) with operating speeds of 12.3 and 10.0 RPM for the water analog simulation.

Denlinger et al. (1998), performed 3-D PTV in the same facility at 10.7 RPM. They found that small scale motions exist during the early part of the intake stroke and only coalesce to form organized tumble and swirl motions late in the intake stroke (near BDC). The small scale motions early in the intake stroke are important to the turbulent mixing of air and fuel. The large-scale structures store a great deal of kinetic energy which is transformed into turbulence as they break down. Denlinger et al. (1998) did not take any measurements beyond BDC and did not show how the vortex motion would breakdown during the compression stroke.

Choi and Guezennec (1999) repeated the experiments for 12 RPM. They found that the average flow field does not evolve into the final form of organized tumble and swirl until late in the intake stroke. They found the appearance of small scale eddies at higher operating speeds and early in the intake stroke. They found that the flow was highly three dimensional and that 2-D PIV would be incorrect for engine
research.

Li and Sullivan (2000) compared previous PIV measurements in a square piston chamber by Davis (1999) with KIVA (Amsden et al., 1985) results and found good agreement between the two techniques at 20 RPM. This indicated that the flow was largely two dimensional at 20 RPM. There was some strong three-dimensionality that appeared at 40 and 60 RPM in the KIVA model that was not measured experimentally. The three-dimensionality can be attributed to the transition to turbulence at a lower Reynolds number in a square duct relative to a cylindrical chamber (Shames, 1992), and indicates that there is a range of fluid velocities whereby the flow can be considered two dimensional.

Current technology has limited the majority of PIV measurement within high speed production engines to double exposed images where flow direction is not resolved. A water analog engine was used by Denlinger et al. (1998) and Choi and Guezennec (1999) in order to sufficiently reduce the fluid velocities so that single exposed images could be captured and cross-correlation algorithms could be used.

It is clear that flow visualization and PIV has shown the existence, formation and breakdown of large scale vortices such as tumble and swirl. Tumble and swirl typically become organized late in the intake stroke (Denlinger et al., 1998; Choi & Guezennec, 1999) and do not breakdown into small scale turbulence until late in the compression stroke (Reeves et al., 1996). Denlinger et al. (1998) and Heywood (1988) have indicated the importance of these large scale vortices in the mixing of fuel/air as well as the decrease in cyclic variability of the fluid motion.

The unique data collection and reduction method used in this is well suited to other reciprocating flows; this includes the validation of modeled blood flow in the left side of the heart (Jacobsen, 1999) as well as the validation of a numerical simulation of airflow in the human nasal cavity (Keyhani et al., 1995). From abstracts obtained for
the 1999 ASME Bioengineering Conference, two current projects were found to be of interest. Zhao and Yeo (1999) used PIV to study the pulsatile flow through a bileaflet mechanical aortic heart valve under physiological conditions. The hinge positions of the valve is strongly related to the fluid motion and cause large scale flow separation and vortices during the opening and closing time. These fluid motions have been related to the instability of the bileaflet valve. Browne et al. (1999) compared the steady flow downstream of a St. Jude bileaflet heart valve using LDV and PIV. They found that PIV and LDV measurement had a maximum mean velocity difference of 40%. The disagreement is likely due to the cross-correlation algorithm. Many of the experimental issues facing Browne et al. (1999) and Zhao and Yeo (1999) have been dealt with in this study.

1.3 Terms of Interest

To characterize the mixing within a flow, the Reynolds stresses and vorticity are important terms to consider. For a 2 dimensional Cartesian (X,Y) velocity field, vorticity is defined as (Shames, 1992),

\[ \omega = \frac{\partial V}{\partial x} - \frac{\partial U}{\partial y} \]  

(1.9)

where \( V \) is the instantaneous velocity in the Y direction, and \( U \) is the instantaneous velocity in the X direction.

The three Reynolds stresses that can be characterized by 2 dimensional Cartesian (X,Y) velocity fields are \( R_{uu} \), \( R_{uv} \) and \( R_{vv} \) defined as (Shames, 1992),
where \( u' \) is the fluctuating portion of the instantaneous velocity in the X direction and \( v' \) is the fluctuating portion of the instantaneous velocity in the Y direction.

To date, no research has been done to investigate the terms which quantify large and small scale mixing.

1.4 Advantages and Disadvantages of the Three Techniques

Table 1.1 summarizes the advantages and disadvantages of the three techniques with respect to their basic limitations and their ability to measure terms in Equations 1.9 to 1.12.

With spatially resolved PIV data, all of the terms in Equations 1.9 to 1.12 can be found and both the small and large scale mixing within a flow can be quantified; it is clear that PIV offers the best opportunity to examine the mixing characteristics.
<table>
<thead>
<tr>
<th>Technique</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
</table>
| PIV       | • Spatial measurement technique  
          | • Data can be used to calculate spatial derivatives in non-stationary flows  
          | • Flow direction can be resolved  
          | • Non-intrusive  
          | • Can determine all terms in Equations 1.9 to 1.12 over a 2 dimensional region of space | • No time resolved information  
          | • Expensive relative to HWA |
| LDV       | • Has better frequency resolution than HWA (of the order MHz)  
          | • Non-intrusive  
          | • Flow direction can be determined  
          | • Can determine all terms in Equations 1.10 to 1.12 at one point in space | • Measurements are random in time  
          | • Spatial derivatives are difficult to obtain in non-stationary flows without the use of multiple LDV systems  
          | • Expensive relative to HWA |
| HWA       | • Inexpensive  
          | • Excellent time resolution  
          | • Can determine all terms in Equations 1.10 to 1.12 at one point in space | • Intrusive technique  
          | • Frequency resolution limited for cyclic flows of practical interest  
          | • Must be calibrated  
          | • Spatial derivatives are difficult to obtain in non-stationary flows  
          | • Flow direction is not resolved |

Table 1.1: Comparison of Measurement Techniques
within the flow.

1.5 The Present Study

The interest is on the development of a method to analyze statistically significant reciprocating flow properties. It is recognized that this work will not have direct relevance to production engines. However, the work will develop insight into cyclic flows, and also provide a database for validation of CFD codes. While the turbulent characteristics of production engines will not be matched, large scale motions should have some relation between the water analog engine and a production engine.

An optical water-analog engine has been designed and constructed for use with PIV and a triggering system has been constructed to gather velocity fields at specific crank angles within the engine cycle. To determine the spatial and time evolution of the flow fields, two dimensional velocity fields have been measured at 6 positions for varying crank angles. An Adaptive Cross Correlation algorithm implemented into a Matlab toolbox by Usera (1999) has been used to calculate the velocity fields from the raw images. Reynolds stresses, vorticity and instantaneous and mean velocities have been measured or calculated by ensemble averaging 200 velocity fields at each position at each crank angle. From these quantities, insight will be given into the mechanism of large and small scale mixing.

1.6 Objectives

The main objectives were as follows:

- Design and construction of an optical reciprocating piston chamber
- Construction of the PIV system to capture velocity fields at distinct crank angles
Investigate the flow fields near the walls and in the center of the valves with Particle Image Velocimetry.

Obtain a statistically significant database over an entire cycle for use in CFD validation.

Give insight into the mechanism of large and small scale mixing.
Chapter 2

Experimental Setup and Procedure

2.1 Optical Water Analog Engine

The water analog engine was square and made of stainless steel for structural stability. It had two circular valves and rectangular glass ports to allow the laser and camera to penetrate the flow cell.

2.2 Similarity

The flow cell was not designed to completely simulate the conditions of an automobile piston chamber. However, since it has been mentioned that some features of the flow cell will be similar to that of a motored production engine, the similarity conditions are outlined to categorize any assumptions necessary to compare the flow cell to an automotive production engine.

Ekchian and Hoult (1979) attempted to maintain dynamic similarity in an engine model using water as the working fluid. They only matched the Reynolds number based on the mean piston velocity and the diameter of the piston. Since water has
Table 2.1: Kinematic Viscosity of Water and Air at 20°C

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Kinematic Viscosity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>$1.02 \times 10^{-6} \text{ m}^2/\text{s}$</td>
</tr>
<tr>
<td>Air</td>
<td>$17.4 \times 10^{-6} \text{ m}^2/\text{s}$</td>
</tr>
</tbody>
</table>

a kinematic viscosity that is 17 times less than that of air (see Table 2.1), they were able to reduce the speed of the model by 17 times while maintaining similarity.

Khalighi and Huebler (1988) detailed a list of three criteria that must be met in order to retain similarity between a motored engine and a water analog engine. They noted that there are three criteria that must be retained for similitude,

- Thermal,
- Kinematic,
- Dynamic

Thermal similitude was retained if there was no heat transfer. Khalighi and Huebler (1988) note that during the intake stroke, the heat transfer within a motored production engine is negligible. It is possible to model the time averaged heat transfer within a cylinder using a Nusselt number relationship (Heywood, 1988):

$$Nu = aRe^m Pr^n$$  \hspace{1cm} (2.1)

Kinematic similarity was retained only when the compressible air-fuel mixture could be considered incompressible. From one dimensional incompressible flow theory (Equation 2.2), Ma et al. (1986) found that compressibility has only a moderate effect on the fluid motion during the intake stroke only,
\[ \frac{P_o - P}{0.5 \rho V^2} = 1 + \frac{M^2}{4} + \frac{M^4}{40} + \ldots \] (2.2)

where \( P_o \) is a reference pressure, \( P \) is the actual pressure, \( V \) is the fluid velocity and \( M \) is the Mach Number.

Heywood (1988) noted that through the valves, the fluid velocities are at levels where the gas can no longer be considered incompressible. Depending on the valve diameter, at high speeds, the flow can even become choked (where Mach number equals 1) (Heywood, 1988).

Dynamic similarity was shown to be retained by the non-dimensionalized momentum equation (Equation 2.3) for an incompressible fluid (Shames, 1992),

\[ St \frac{\partial V_j}{\partial t} + V_k \frac{\partial V_j}{\partial x_k} = -\frac{\partial P}{\partial x_j} + \frac{1}{Re} \frac{\partial^2 V_j}{\partial x_i \partial x_i} \] (2.3)

The Reynolds number

\[ Re = \frac{U_P D}{\nu} \] (2.4)

and the Strouhal number

\[ St = \frac{D}{t U_P} \] (2.5)

both appear in Equation 2.3 where \( D \) is the piston diameter, \( U_P \) is the piston velocity, \( t \) is a time scale that is associated with the oscillation of the flow, and \( \nu \) is the kinematic viscosity of the fluid. The Reynolds number is the ratio of the inertia of the fluid to the viscosity. The Strouhal number is the ratio of oscillation to mean flow velocity.
An applicable time scale that is associated with the oscillation of the flow for I.C. engines is the engine's angular velocity (Khalighi & Huebler, 1988); \( t \approx \frac{1}{\omega_p} \). Because of this, the Strouhal number for the model and engine will be matched since the angular velocity of the engine is directly related to the piston velocity.

Matching the Reynolds number between the model and the production engine leads to:

\[
\left( \frac{U_p D}{\nu} \right)_{\text{MODEL}} = \left( \frac{U_p D}{\nu} \right)_{\text{ENGINE}} \tag{2.6}
\]

or, alternatively

\[
\frac{U_{p,\text{MODEL}}}{U_{p,\text{ENGINE}}} = \frac{\nu_{\text{MODEL}}}{\nu_{\text{ENGINE}}} \tag{2.7}
\]

and because water has a kinematic viscosity that is 17 times less than that of air (see Table 2.1), Equation 2.7 demonstrates that it is possible to match Reynolds number while operating the model at a factor 17 reduction in speed so long as all other geometric conditions are kept constant.

The current water analog engine does not completely simulate a production engine for several reasons. The water analog engine could only simulate the intake stroke of a production engine where compressibility is typically small. However, Heywood (1988) mentioned that the flow at the valves can become choked which would imply that the flow is compressible in that area. Furthermore, the geometry of the water analog engine is square and does not simulate a cylindrical piston chamber. For these reasons, the water analog engine will not be compared to a production engine.
2.3 Engine Assembly

A square cross section was used to facilitate maintenance and optical access. Furthermore, the choice to use a square stainless steel engine with glass ports instead of a plexiglass engine was made to improve the structural stability of the flow cell which was a problem in the research conducted by Davis (1999).

2.4 Flow Cell Chamber

Figure 2.1: Flow Cell Assembly

In the present study, the flow cell was a 100 mm x 100 mm (inside area) stainless steel square cross section that was 245 mm long. A 29 mm by 180 mm rectangular slot was milled into the top and bottom and a 100 x 200 mm rectangular slot was milled into the two sides. Two pieces of crystal clear \textsuperscript{TM} glass were glued together
(the glue had the same index of refraction of the glass) and hand fitted flush ±0.1\text{mm} (Roosman, 2000) with the inside surfaces of all four slots, and silicone was used to bond and seal the glass to the stainless steel. Stainless steel slotted covers on the top and bottom and two sides were cross bolted to hold the glass ports firmly together. The end plates were fastened with bolts and sealed with a rubber gasket and silicone to minimize leakage and facilitate disassembly for maintenance and cleaning. The inlet plate (Figure 2.3) had two circular 40mm diameter holes into which the valves were placed and two 40mm inside diameter stainless steel connector tubes to draw water from the head tanks.
The back plate had a 40mm hole (see Figure 2.4) and steel connector tube that was connected it to a separate head tank. The back connection was needed to equalize the pressure on both sides of the piston when the piston was not in motion. It was also needed to ensure a relatively constant piston velocity profile when a constant torque motor was used to drive the flow.

2.5 Piston Assembly

The piston face was made of a 100 mm x 100 mm piece of stainless steel. 101 mm x 101 mm alternating layers of Teflon and 100 mm x 100 mm plastic were used to minimize the amount of fluid transfer across the piston face. The piston rod was screwed to the piston face assembly (see Figure 2.5 ) and fastened to Crank Arm A with a pin.
Figure 2.5: Piston Assembly

Figure 2.6: Crank Wheel Assembly
2.6 Crank Wheel and Engine

The aluminum crank wheel was 152 mm in diameter. It had an outer sliding sheath and was attached to a 1/2 HP variable speed direct current motor. Crank Arm A was fastened to the wheel 62.5mm away from the wheel's center point giving a stroke length of 125mm. The sliding sheath had a protrusion of metal that indicated the crank wheel position as shown on Figure 2.6. The sliding sheath could be moved 360° around the crank wheel and was fastened by set screws to the crank wheel surface.

2.7 Water Supply

A head tank with two clear plastic hoses was used to supply water to the stainless steel connector tubes of the engine (see Figure 2.7). A separate head tank was used to supply water to the back using a similar clear plastic hose. The two tanks were filled with approximately 50l of distilled water. Distilled water was used to minimize
the number of large particles that are found in tap water. Because there was some fluid transfer across the piston face, the head tanks would naturally equalize the pressure across the piston face if they were left undisturbed overnight. It was found that large bubbles would appear if the piston was set in motion immediately after the head tanks were filled with water. It was necessary to run the engine for 5 minutes and manually remove these bubbles by lifting the plastic tubes up and down. The seeding particles were injected with a syringe through the clear plastic tubing 2 inches away from the top intake connection. Some water sprayed out from the hole after the syringe was withdrawn, so a rubber sheath was clamped over the hole to eliminate any water loss. A similar clamp was placed over the bottom tube so that there was approximately equal flow resistance at the injection point in both tubes. After 2 days of experiments, agglomerates of seed particles settled in the plastic tubes. It was necessary to clean the plastic tubes whenever the agglomerates were large enough to detach from the tubing and float into the engine.

### 2.8 Coordinate System

A rectangular Cartesian coordinate system was used in this experiment (see Figure 2.8).

The origin of the coordinate system is in the bottom left hand corner of the flow cell in the geometric center of the spanwise direction as in Figure 2.8. X and U are the streamwise axis and velocity component respectively, Y and V are the vertical axis and velocity component respectively and Z is the spanwise axis.
CHAPTER 2. EXPERIMENTAL SETUP AND PROCEDURE

Figure 2.8: Coordinate System

Figure 2.9: Piston Displacement in Crank Angle Degrees (CAD)
2.9 Engine Cycles

When the piston is at top dead center, it is at zero crank angle degrees (see Figure 2.9). When the piston had moved half way through the piston in a positive X direction, it was at 90 CAD (62.5 mm from the top) and when the piston was furthest from the valves it is as 180 CAD (125 mm from the top). When the piston was moving towards the valves (a negative X direction) and was half way through its stroke, it was at 270 CAD (62.5 mm from the top) and when it returned to the TDC position it was at 360 CAD (125 mm from the top).

2.10 Location of Measurements

By changing the location of the metal obstruction on the crank wheel, the measurements were taken at different crank angles.

Six measurement positions were used (Figure 2.8); 35 mm and 75 mm away from the top of the cylinder along the X axis located in the center, top and bottom of the vertical axis. All measurements were made at the geometric center of the spanwise (Z) axis.

Table 2.2 lists the measurement positions. Labels for the measurement positions have been identified in the first column.

Table 2.3 lists the crank angles at which measurements were taken. 200 measurements were taken at each crank angle for all of the measurement locations.

2.11 Piston Velocity

The piston velocity was determined by following the motion of the piston face with the Pulnix TM-9701. The field of view of the camera was 11 mm by 7 mm. It was
CHAPTER 2. EXPERIMENTAL SETUP AND PROCEDURE

Table 2.2: Table of Measurement Positions

<table>
<thead>
<tr>
<th>Measurement Label</th>
<th>X Measurement Region</th>
<th>Y Measurement Region</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>35 mm to 46 mm</td>
<td>92 mm to 99 mm</td>
</tr>
<tr>
<td>(b)</td>
<td>75 mm to 86 mm</td>
<td>92 mm to 99 mm</td>
</tr>
<tr>
<td>(c)</td>
<td>35 mm to 46 mm</td>
<td>46.5 mm to 53.5 mm</td>
</tr>
<tr>
<td>(d)</td>
<td>75 mm to 86 mm</td>
<td>46.5 mm to 53.5 mm</td>
</tr>
<tr>
<td>(e)</td>
<td>35 mm to 46 mm</td>
<td>1 mm to 8 mm</td>
</tr>
<tr>
<td>(f)</td>
<td>75 mm to 86 mm</td>
<td>1 mm to 8 mm</td>
</tr>
</tbody>
</table>

Table 2.3: Crank Angles Measured

<table>
<thead>
<tr>
<th>Measurement Label</th>
<th>Crank Angles Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>60, 90, 120, 150, 180, 210, 240, 270, 300</td>
</tr>
<tr>
<td>(b)</td>
<td>90, 120, 150, 180, 210, 240, 270</td>
</tr>
<tr>
<td>(c)</td>
<td>60, 90, 120, 150, 180, 210, 240, 270, 300</td>
</tr>
<tr>
<td>(d)</td>
<td>90, 120, 150, 180, 210, 240, 270</td>
</tr>
<tr>
<td>(e)</td>
<td>60, 90, 120, 150, 180, 210, 240, 270, 300</td>
</tr>
<tr>
<td>(f)</td>
<td>90, 120, 150, 180, 210, 240, 270</td>
</tr>
</tbody>
</table>

Figure 2.10: Inverted Image of the Piston Position at Time A

Figure 2.11: Inverted Image of the Piston Position at Time B = Time A + 1/30 sec
positioned at 21 measurement locations with a 50 % overlap between each location. The measurement areas started at TDC and ended at BDC.

The metal protrusion on the crank wheel was set to trigger image collection 1 mm before the piston face reached the measurement area. The camera was set to collect 15 images per trigger at 30 Hz on shutter mode 6 (0.25 ms shutter duration) which was experimentally found to produce clear images. The difference in the position of the piston between images was divided by the time between images \( T = \frac{1}{30} \text{s} \) in order to determine the velocity,

\[
V_X = \frac{\Delta X}{(\frac{1}{30})M}
\]  

(2.8)

where \( \Delta X \) was the piston displacement between frames in pixels and \( M \) was the magnification factor ( \( M_{\text{average}} = 69.5 \text{ pixels/mm} \) ). The piston displacement was found by searching for the pixel location of the first peak in the gray level matrix of frame 1 (the peak occurs when the image goes from black to white which occurs at the piston face) and subtracting it from the pixel location of the first peak in the gray level matrix of frame 2.

A typical set of images can be seen in Figures 2.10 and 2.11. The piston motion is from right to left and the time between images is \( T = \frac{1}{30} \text{s} \).

The piston velocity curve can be seen in Figure 2.12. The displacement is measured in mm from the TDC position (20 mm from the top of the flow cell and 5 mm from the valves). The piston position as a function of CAD is shown in Figure 2.13.

During the intake stroke, the piston velocity curve accelerated smoothly from 0 mm (TDC) to 35 mm where it became level. The piston velocity remained level until about 100 mm when it sharply accelerated until approximately 125 mm (BDC).
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Figure 2.12: Piston Velocity Curve
Figure 2.13: Distance Travelled by the Piston
During the exhaust stroke, the piston velocity accelerated smoothly from 125 mm (BDC) to 100 mm where it became level. The piston velocity remained level until about 25 mm when it sharply accelerated until approximately 0 mm (TDC).

The average piston velocity was found to be $54.2 \frac{mm}{s}$ determined by integrating over the velocity curve and dividing by 250 mm.

### 2.12 Piston Chamber

![Piston Chamber Dimensions](image)

Figure 2.14: Piston Chamber Dimensions

A clearance of 20 mm was used in the experiment, and both valves were open 15 mm (See Figure 2.14). The piston stroke length was set to 125 mm. A variable speed 1/2 Horsepower direct current (DC) motor was used to drive the piston and was kept at a constant setting to give a mean piston speed of $54.2 \frac{mm}{s}$. The Reynolds number,

$$\Re = \frac{V D}{\nu} = 5366 \quad (2.9)$$

where $V$ is the mean piston velocity, $D$ is the width of the chamber (100 mm) and $\nu$ is
the kinematic viscosity of water \( 1.01 \times 10^{-6} \text{ m}^2/\text{s} \). Durst et al. (1989) suggested (from flow visualization and LDV measurements of piston-driven, unsteady separation at a sudden expansion in a tube) that the transition from laminar to turbulent flow occurs at a Reynolds number of 125. The Reynolds number was defined as in Equation 2.9 using the diameter of the tube for \( D \).

Thus, the flow was considered turbulent as defined by Durst et al. (1989).

### 2.13 Particle Image Velocimetry Theory

2D PIV is a non intrusive technique which allows the velocity of a fluid to be measured in a region of space that is illuminated by a thin sheet of light (see Figure 2.15). The fluid was seeded by small \( \approx 15 \mu m \) silver coated hollow glass spheres and the properties of the seeding particles can be found in Table 2.4. The seed particles had a relative density of 1.65 to water and the same particles used in the study done by Davis (1999) were used in this experiment. A pulsed Neodymium-Yttrium Aluminum Garnet (Nd:YAG) laser was used to illuminate the seed particles two instances in time on two separate CCD image frames. The time separation between the laser pulses was 162\( \mu \text{s} \) and the field of view of each image was 11 mm by 7 mm.

Each image was 768 pixels by 484 pixels. The images were separated into 32 by 32 pixel interrogation regions with approximately 7-10 particles per interrogation region as suggested by Keane and Adrian (1992). The interrogation regions were overlapped by 50\% to increase the number of vectors in the velocity field.

<table>
<thead>
<tr>
<th>Shape</th>
<th>Relative Density to Water</th>
<th>Mean Diameter</th>
<th>Diameter Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spherical</td>
<td>1.65</td>
<td>15 ( \mu m )</td>
<td>10 – 30 ( \mu m )</td>
</tr>
</tbody>
</table>

Table 2.4: Silver Coated Hollow Glass Core Seed Particle Properties

To determine the average particle displacements within each interrogation region
Figure 2.15: Particle Image Velocimetry in the Water Analog Engine
and to resolve flow direction, a cross–correlation algorithm was used (see Section 2.18 for details).

Once the average particle displacement in each of interrogation regions was known, velocity vectors were found by dividing the average particle displacement by the known laser pulse separation time ($\Delta t = 162 \mu\text{sec}$),

$$V = \frac{\text{Average Displacement}}{\Delta t}$$  \hspace{1cm} (2.10)

which resulted in one velocity vector for each interrogation region.

For a 768 by 484 pixel image with 32 by 32 pixel interrogation areas that are overlapped by 50%, the velocity field had 45 by 27 vectors. This resulted in a physical distance of 0.23mm between neighboring velocity vectors.

### 2.14 Data Acquisition Components

Table 2.5 summarizes the equipment used to obtain the PIV images.

A 24 Bit BITFLOW Road Runner 44 framegrabber was connected using the RS-422 digital protocol to obtain images from the 8 Bit Pulnix TM-9701 CCD camera. VideoSavant software V 3.0 for Windows NT (IO industries, London, ON) was used to store the images. A camera file was created for use with asynchronously reset Pulnix TM–9701 cameras (an asynchronous reset starts a new frame after an external trigger is received. It is necessary to asynchronously reset the camera if the images are to be collected at specific crank angles so that the PIV collection system can be sequenced to the beginning of the camera frame). The camera file would send a software trigger from the framegrabber to asynchronously reset the camera. Two images were stored
**Table 2.5: PIV Components Summary**

<table>
<thead>
<tr>
<th>Component</th>
<th>Type</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>CCD Camera</td>
<td>Pulnix TM 9701 2/3 inch Progressive Scan</td>
<td>768 (H) x 484 (V) Imager Size: 8.9 (H) x 6.6 (V) mm Frame rate: 30 Hz Video Output: 8 bit</td>
</tr>
<tr>
<td>Optical Sensor</td>
<td>Miro Circuit</td>
<td>TTL Pulse Generating Sensor</td>
</tr>
<tr>
<td>Trigger</td>
<td>Miro Circuit</td>
<td>TTL Pulse Generating Sensor Synchronizes Delay Generator and PC Image Acquisition</td>
</tr>
<tr>
<td>Frame Grabber</td>
<td>Bitflow Road Runner 44</td>
<td>RS 422 Interface 24 bit</td>
</tr>
<tr>
<td>Software</td>
<td>IO Industries Video Savant V 3.0</td>
<td></td>
</tr>
<tr>
<td>Laser</td>
<td>Continuum Minilite ND:Yag</td>
<td>Wavelength: 532 nm (Green) Peak Intensity: 4.0MW</td>
</tr>
<tr>
<td>Delay Generator</td>
<td>Stanford Research Model DG535 Digital Pulse generator</td>
<td></td>
</tr>
</tbody>
</table>

in one image buffer and a set of 24 image pairs were stored in the computer's RAM.

A CONTINUUM MINILITE dual cavity Neodymium-Yttrium Aluminum Garnet (Nd:YAG) laser was used to produce a vertically polarized green light source of 532 nm wavelength that illuminated the seed particles in the flow. The beam size was roughly 3 mm in diameter and has a power of 4.0 MW peak intensity for 5 ± 2 ns duration (Continuum, 1999). To create a 1 mm thick light sheet used to illuminate particles in the flow, a cylindrical lens of -25.4 mm focal length and a spherical lens of 250 mm focal length were used. The cylindrical lens was used to diverge the beam and create a laser sheet. The spherical lens was used to converge the sheet and create a 1 mm thick plane of light at the measurement locations.
2.15 Particle Seeding

The seed particles were silver coated spherical particles with a hollow glass core (see Table 2.4 for properties). With 50\(\ell\) of distilled water used in the experiment, approximately 6.4 g of seed particles were needed to seed the flow. Adding 6.4 g of particles to both tanks proved inadequate as the seed particles tended to be trapped by the plastic tubing or settled in cracks and crevices in the plastic tanks and connecting elbows. This reduced the measured seed concentration to under 5 particles per 32 by 32 pixel area. Furthermore, during an experimental run, the seed density would be reduced because the Teflon seals on the piston face would allow the passage of water, but would trap seed particles. To increase the seed density to between approximately 7 and 10 particles per 32 by 32 pixel area during an experimental run, an additional 15 grams (total) was added to both tanks and 3 grams were injected directly into the piston chamber. This increased the seed particle densities to a level where data reduction algorithms could be used.

The mass percentage of particles,

\[ MP = \frac{\text{Seed Particle Mass}}{\text{Water Mass}} \times 100\% \quad (2.11) \]

where the seed particle mass was 6.4 g + 15 g + 3 g or 24.4 g. The mass of 50\(\ell\) of water was approximately 50000 g. The mass percentage (MP) was then 0.05%.

2.16 Magnification Factor Measurement

Before measurements were taken, the camera was calibrated in order to relate the number of pixels to the actual measurement area. The camera was focused on a
scale. The scale indicated that the measurement area was 11mm by 7mm. Before each measurement, the camera was targeted on that same scale to ensure that the measurement area had not changed. By determining the number of pixels in a measurement area, the magnification factor could be found using,

\[ M_x = \frac{P_x}{MI_x} \]  
(2.12)

\[ M_y = \frac{P_y}{MI_y} \]  
(2.13)

where \( P \) is the number of pixels and \( MI \) is the measurement interval in mm. The Pulnix TM–9701 is 768 by 484 pixels and therefore \( P_x = 768 \) and \( P_y = 484 \). The measurement intervals are \( MI_x = 11 \) mm and \( MI_y = 7 \) mm. The magnification factors can then be found; \( M_x = 69.8 \frac{\text{pixels}}{\text{mm}} \) and \( M_y = 69.1 \frac{\text{pixels}}{\text{mm}} \) yielding an average magnification factor of \( M_{\text{average}} = 69.5 \frac{\text{pixels}}{\text{mm}} \).

2.17 Image Acquisition

<table>
<thead>
<tr>
<th>RS-422 Monochrome Video (Digital Format)</th>
</tr>
</thead>
<tbody>
<tr>
<td>• 768 pixels × 484 lines</td>
</tr>
<tr>
<td>• Digital, Non-Interlaced</td>
</tr>
<tr>
<td>• 29.97 frames/s</td>
</tr>
<tr>
<td>• Shutter mode 4</td>
</tr>
<tr>
<td>• 8 Bits</td>
</tr>
</tbody>
</table>

Table 2.6: Pulnix TM–9701 CCD Camera Properties

To obtain the ensemble statistics at each of the measured crank angles, a triggering device was created to signal the PIV system at specific crank angles. The schematic
for the triggering mechanism can be seen on Figure 2.16. When the metal obstruction placed on the crank wheel passed through an optical sensor circuit designed by Miro Kalovsky, a 3 Volt Positive TTL signal was sent to the Miro Trigger Circuit (MTC).

![Diagram showing the triggering mechanism](image)

**Figure 2.16: Schematic of Triggering Mechanism**

![Diagram showing camera timing](image)

**Figure 2.17: Camera Timing**

The MTC sent one 3 Volt positive TTL signal to the parallel port of the data acquisition PC and one to the delay generator.

The PC sent one TTL signal to asynchronously reset the Pulnix TM-9701 CCD camera. The Pulnix TM-9701 is designed to obtain cross-correlated image pairs in
free run mode (which means that it is not asynchronously reset and cannot accept triggers). In this mode, the camera is run without any electronic shutters and the lasers are pulsed during consecutive frames. The camera is run without shutters so that the lasers can be pulsed with a minimum time separation (Kinney, 1999), rated at under 30μsec in free run mode (Luo, 1999).

The TM–9701 was not designed to obtain cross–correlated image pairs in asynchronous reset mode; the TM–9701 did not have an option to run in both asynchronous reset mode and unshuttered mode at the same time. Kinney (1999) suggested that if the asynchronous reset pulse was sent for a period of 5 frames (170 ms) that frames 4 and 5 would be unshuttered. This was implemented by Kinney (1999).

Figure 2.17 shows the actual timing of the laser pulses, the asynchronous reset signal sent by the framegrabber and the image transfer mechanism from the camera. The VINIT signal is the TTL signal supplied by the PC (170ms), the discharge pulse clears the CCD pixels and the transfer gate is when the camera physically sends the data from the CCD camera to the frame grabber. When VINIT is kept low for five frames, the discharge pulse in frame four and five is bypassed. This is important for image transfer, since the information that has been integrated on the right side of the transfer gate would have been cleared at the discharge pulse in frame five, were it not bypassed.

The lasers were pulsed on either side of the transfer gate in frame four. The particle position during laser pulse one were output to frame four and the particle positions during laser pulse two were output to frame five.

The mechanism used to acquire the image frames was similar to the free run mode collection method used by Luo (1999) except that the PIV system was triggered at specific crank angles and the fourth and fifth frame were stored after each trigger.

Frames four and five were 8 bit grey scale images and were stored on the RAM of
the data acquisition PC. Twenty four image pairs were stored on the RAM of data acquisition PC before they were exported in TIF format from Video Savant to the hard disk.

<table>
<thead>
<tr>
<th>Signal from Delay Generator</th>
<th>Timing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Signal A (Flashlamp of Laser A)</td>
<td>100.338ms after the Crank Wheel Trigger</td>
</tr>
<tr>
<td>Signal B (Q-switch of Laser A)</td>
<td>152μs after Signal A</td>
</tr>
<tr>
<td>Signal C (Flashlamp of Laser B)</td>
<td>162μs after Signal A</td>
</tr>
<tr>
<td>Signal D (Q-switch of Laser B)</td>
<td>152μs after Signal C</td>
</tr>
</tbody>
</table>

Table 2.7: Delay Generator Timing Chart

A Stanford Research Model DG535 digital/pulse generator was used to trigger the lasers with a time separation of 162μsec. Table 2.7 shows the timing of the signals sent from the delay generator. The delay generator sent a series of four TTL signals to the dual cavity laser. For each laser beam output, two signals were needed: one for the flashlamp (which provided the energy to the laser rod) and one to the Q-switch (which increased the laser's peak power and allowed the rod to fire). The transfer gate in frame 4 occurred 572μsec after the asynchronous signal was sent to the camera.

The first signal (signal A) was sent from the delay generator to laser A's flashlamp 100.338 msec after the crank wheel triggered the system. The second signal (signal B) was sent 152μsec after signal A to laser A's Q-switch; 152μsec is the time needed to charge the flashlamp. The third signal (signal C) was sent 162μsec after signal A, and the fourth signal (signal D) was sent 152μsec after that.

Thus, laser pulse separation time was signal D minus signal B or 162μsec.

This is the first known research using a Pulnix TM-9701 to obtain cross-correlated image pairs in asynchronous reset mode.
2.18 Cross Correlation Theory

The image pairs were separated into a grid of 32 by 32 pixel interrogation areas and were overlapped by 50%. Each 32 by 32 pixel area had approximately 7 to 10 particles in it (as recommended by Keane and Adrian (1992)). The average particle displacement between image one ($I_1$) and image two ($I_2$) was determined by the highest peak on the surface obtained by the cross–correlation formula (Keane & Adrian, 1992),

$$R(s) = \int I_1(X)I_2(X+s)dX$$  \hspace{1cm} (2.14)

where $X$ was the interrogation area, $I_1$ and $I_2$ were the entire grey scale images obtained from the Pulnix TM–9701 and $s$ was the separation between the two interrogation areas. The cross–correlation procedure used the interrogation area from the first image and translated it an amount $s$ on the second image until a correlation peak was found. If the particles translated an amount $G$ from image one to image two, the cross–correlation surface, $R(s)$, had a distinct peak when $s = G$.

$R(s)$ can be broken up into three components (Keane & Adrian, 1992),

$$R(s) = R_C(s) + R_D(s) + R_F(s)$$  \hspace{1cm} (2.15)

where $R_C(s)$ is the mean background correlation (the correlation of the mean background light with itself), $R_F(s)$ is the fluctuation of the background noise (the correlation of the fluctuating background light with itself) and $R_D(s)$ is the displacement correlation peak. The displacement correlation peak is the largest of all three peaks and represents the mean particle displacement.

A typical cross correlation surface can be seen in Figure 2.18. The distance from
the center of the interrogation area to the centroid of displacement correlation peak is the average displacement of the particles within the interrogation area.

2.19 Adaptive Cross Correlation Algorithm

A goal in PIV is to maximize the number of vectors, where one vector is output per interrogation area. However, a lower limit exists for the size of the interrogation area because the accuracy of the cross correlation algorithm depends on the number of particles per interrogation region. Keane and Adrian (1992) detailed a set of optimization criteria that were applied to the cross-correlated PIV measurements.

- The number of particles per interrogation region should be approximately 7 per image
- The magnitude of the displacement vectors should be less than 25% of the...
interrogation region size; this minimized the number of in-plane particle pair losses.

- The out of plane particle pair losses were minimized by keeping the velocity which is normal to the laser sheet less than 25% of the laser sheet thickness divided by the laser pulse separation time.

An adaptive cross correlation algorithm (ACC) implemented by Usera (1999) based on the work by Nogueira et al. (1997) was used to enhance the standard cross correlation (SCC) and validation process and increased the spatial density of velocity vectors relative to an SCC algorithm.

For the SCC, as the number of velocity vectors per unit area (spatial resolution) increases, the maximum measurable velocity (velocity resolution) decreases because the displacement of the particles can be no more than 25% of the interrogation area.

The ACC algorithm is iterative, and a SCC algorithm was used as a first approximation to the velocity field. In the first iteration the interrogation area was 64 by 64 pixels. The interrogation areas were overlapped by 50% and a cross-correlation was carried out. The velocity field was validated with a maximum displacement and an eye validation. These techniques are detailed in section 2.20.

This allowed a smaller interrogation region size (32 by 32 pixels) to be used (see Figure 2.19 ) in a second iteration, quadrupling the spatial resolution. The interrogation regions in the second iteration were overlapped by 50% and a cross-correlation and validation were carried out in a similar way to the first iteration. This yielded a separation between neighboring vector of 0.23 mm and a maximum resolvable velocity of $1.1 \frac{m}{sec}$.

The velocity resolution was determined by the first iteration and the spatial resolution was determined by the second iteration. The obvious benefit of the ACC
over the SCC method is that a high velocity flow can be described with good spatial resolution.

![Diagram showing the process of Adaptive Cross Correlation]

Figure 2.19: Adaptive Cross Correlation

2.20 Validation

After each iteration of calculating the cross correlation surface for the image pairs, $R(s)$, a validation was made to ensure that certain criteria have been met. Figure 2.19 is a process flow chart showing where the validation criteria are applied in the data reduction analysis.

The validation criteria used in this experiment were a Maximum Displacement Validation suggested by Keane and Adrian (1992) and an Eye Validation algorithm suggested by Nogueira et al. (1997). The ACC algorithm with these two validation criteria was implemented into a MatLab toolbox by Usera (1999).

The ACC algorithm used a maximum displacement (defined as the magnitude of
the average displacement of particles within an interrogation area as a percentage of the interrogation area size) of 40%. Because the ACC is iterative, and the second interrogation region size is half of the first (32 pixels instead of 64 pixels), the 40% maximum displacement criteria means that the particles can move no more than 12.8 pixels or 20% of the initial interrogation region size, meeting the criteria set by Keane and Adrian (1992).

The Eye validation procedure developed by Nogueira et al. (1997) is used to check if there is a group of spurious vectors. The algorithm is used to identify groups of locally coherent vectors where coherency is determined by calculating the deviation of each vector from its eight neighbors,

\[ dev = \frac{\sum_i |v_i - v_0|}{\sum_i |v_i|} \]  (2.16)

where \( v_i \) refers to the eight closest neighboring vectors of the grid node whose velocity is \( v_0 \) and \( | \cdot | \) is the absolute value. Nogueira et al. (1997) recommended that the deviation should be under 20%; this was used in this study. The zone grows by incorporating neighboring nodes into a coherence zone using the same criteria for each node. This procedure is repeated until the entire velocity field is separated into several distinct coherence zones with approximately 7 coherence zones per velocity field. Nogueira et al. (1997) recommended that there should be a minimum of 10% of the total number of vectors in a coherence region; this was used in this study.

Points within the velocity field were then interpolated by searching for points where velocity vectors had a zero magnitude and then averaging the eight surrounding values (Raffel et al., 1998).

For this work, a sub-pixel interpolation algorithm implemented by Usera (1999) was used. Without subpixel interpolation, integer pixel displacements would be the
smallest resolvable resolution. A Gaussian Peak Fit algorithm detailed by Westerweel (1997) was used to reshape the cross-correlation surface, \( R(s) \) with a sub-pixel accuracy of approximately 0.1 pixels (Marxen et al., 1998).

### 2.21 Turbulence Averaging Technique

In the present experiment, an ensemble average of 200 instantaneous velocity fields will be used at each measurement position. The ensemble average of the instantaneous velocity is defined as,

\[
U(\theta)_i = \frac{1}{200} \sum_{k=1}^{200} \bar{u}_i(\theta, k)_i
\]  

(2.17)

and the fluctuating component is determined by the Reynolds decomposition,

\[
u'(\theta, k)_i = \bar{u}(\theta, k)_i - U(\theta)_i \]

(2.18)

Ensemble averaging typically results in an over-estimation of fluctuating quantities due to variations in the mean flow caused by changes in the inlet and boundary conditions between cycles (Lancaster, 1976; Catania & Mittica, 1985; Sullivan et al., 1999).

### 2.22 Velocity Gradients

In order to determine velocity gradients important for the vorticity calculation, three different methods would be used depending on the location of the velocity vector
within the vector field. If the velocity gradient to be determined was on a boundary, either a forward or backward difference scheme was used,

\[
\left( \frac{\partial U_i}{\partial X_i} \right)_{\text{forward}} = \frac{U_i^{j+1} - U_i^j}{\Delta X}
\]

(2.19)

\[
\left( \frac{\partial U_i}{\partial X_i} \right)_{\text{backward}} = \frac{U_i^j - U_i^{j-1}}{\Delta X}
\]

(2.20)

where \( \Delta X \) was the distance between samples \( U_i^j, U_i^{j-1}, \) and \( U_i^{j+1} \). Any other velocity gradients were determined using a central difference scheme defined as,

\[
\left( \frac{\partial U_i}{\partial X_i} \right)_{\text{central}} = \frac{U_i^{j+1} - U_i^{j-1}}{2\Delta X}
\]

(2.21)

Because there are two types of methods (forward/backward and central) being used to define the vorticity, around the perimeter (where the forward/backward difference method is used), the error will be different than in the center of the measurement area (where the central difference method is used).

### 2.23 Error Analysis

Davis (1999) detailed the error analysis used in this study. The instantaneous velocity fields at specific crank angles was obtained by,

\[
\tilde{u}_i = \frac{d_i}{M \Delta t}
\]

(2.22)
To determine the accuracy of the velocity measurements, four categories were defined:

- Errors in the crank angle
- Errors in the displacement of the particles
- Errors in the time interval
- Errors in the magnification factor

The optical distortion error caused by the camera lense was not considered. The telecentric lens used in this study was nearly distortion free. This was qualitatively verified by viewing a scale near the perimeter of the field of view and ensuring that the quality of the image was similar to the center of the field of view.

### 2.23.1 Errors in the Crank Angle

The angles marked on the crank wheel had an estimated accuracy of ±0.5°. The stroke length was 125 mm, and thus the average error was found by,

\[
180° = 125 \text{mm} \tag{2.23}
\]

\[
1° = \frac{125 \text{mm}}{180} = \pm 0.7 \text{mm} \tag{2.24}
\]

which yielded a displacement error of ±0.35 mm for ±0.5°.
2.23.2 Displacement Error

Marxen et al. (1998) used a three-point Gaussian sub-pixel interpolation scheme (similar to the one used in the toolbox implemented by Usera (1999)) on a numerical simulation of an Oseen Vortex. They found that within in a 32 by 32 pixel area, a displacement error between the calculated displacement and the true displacement of 0.1 pixels was typical.

Tang and Sullivan (2000) determined the displacement error of the Matlab toolbox implemented by Usera (1999) by comparing calculated results with a numerically simulated flow with varying degrees of vorticity. Tang and Sullivan (2000) found that the displacement error was between 0.1 pixels to 0.36 pixels and depended on the degree of vorticity. The vorticity in this work was qualitatively similar to that of the numerically simulated flow which yielded a 0.1 pixel error.

A displacement error of 0.1 pixels will be used in this work.

2.23.3 Time Interval Error

The time interval is defined as the time between laser pulses that illuminate the particles in the flow. The lasers are triggered by the delay generator, and therefore the delay generator is a source of error. The Stanford Research Model DG535 digital/pulse generator had a delay accuracy of ±1.5nsec (Stanford Research Systems, 1994).

2.23.4 Magnification Factor Error

The magnification factor has an error in both terms. The pixel measurement (P) and the measurement interval length (MI) had associated error of $\epsilon_P$ and $\epsilon_{MI}$. The magnification factor error, $\epsilon_M$ can be found as follows,
and by rearranging Equation 2.25, the magnification factor error in the X direction can be found,

\[ \epsilon_{M_x} = \frac{M_x \times MI_x \pm \epsilon_{Px}}{MI_x \pm \epsilon_{MI_x}} - M_x \]  

(2.26)

Similarly, the magnification factor error in the Y direction can be found from

\[ \epsilon_{M_y} = \frac{M_y \times MI_y \pm \epsilon_{Py}}{MI_y \pm \epsilon_{MI_y}} - M_y \]  

(2.27)

The scale used to determine the measurement interval length (MI) had an accuracy of ±0.05 mm which was half of the smallest scale measurement. The accuracy of the camera was ±0.5 pixels. For this study, a measurement interval of 11 mm was used in the X direction and a measurement interval of 7 mm was used in the Y direction. This gave a magnification factor error of \( \epsilon_{M_x} = 0.36 \frac{\text{pixels}}{\text{mm}} \) and \( \epsilon_{M_y} = 0.57 \frac{\text{pixels}}{\text{mm}} \) in the X and Y directions respectively. The average error was then \( \epsilon_{M_{\text{average}}} = 0.47 \frac{\text{pixels}}{\text{mm}} \).

### 2.23.5 Error Summary

The error associated with the four categories were estimated to be ±0.35 mm, ±0.1 pixels, ±1.5 ns, and ±0.47 \( \frac{\text{pixels}}{\text{mm}} \) for the position, displacement, time interval and magnification error respectively. Assuming that 99% of all of the measurements lie within
these error estimates and that the errors have a Gaussian distribution, then the standard deviation of the displacement error, time interval error and magnification factor error are $\sigma_{di} = 0.04$ pixels, $\sigma_{\Delta t} = 5.8 \times 10^{-10}$ s, $\sigma_M = 0.18\ \text{pixels/mm}$ respectively.

From the analysis done in the Appendix, a summary of the errors is shown in Table 2.8.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Uncertainty [$\sigma$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Instantaneous Velocity</td>
<td>3.6 mm/s</td>
</tr>
<tr>
<td>Ensemble Mean Velocity</td>
<td>0.06 mm/s</td>
</tr>
<tr>
<td>Fluctuating Velocity</td>
<td>3.6 mm/s</td>
</tr>
<tr>
<td>Reynolds Stress $\langle u'_i u'_j \rangle$</td>
<td>8%</td>
</tr>
<tr>
<td>Mean Vorticity [Forward/Backward] $\frac{\partial V}{\partial x} - \frac{\partial U}{\partial y}$</td>
<td>11%</td>
</tr>
<tr>
<td>Mean Vorticity [Central] $\frac{\partial V}{\partial x} - \frac{\partial U}{\partial y}$</td>
<td>5.5%</td>
</tr>
</tbody>
</table>

Table 2.8: Error Estimates

### 2.24 Summary

The PIV camera and laser system has been used by Luo (1999) and (Davis, 1999) and has been calibrated with a known flow field previous to this work. The PIV system was not re-calibrated during this experiment. The seed particles used in this experiment were the same particles used in the work of Davis (1999) and the mass percentage of particles was 0.05% indicating that the particles would not significantly affect the flow motion. In this experiment, a new adaptive cross-correlation algorithm was used with a new triggering system that collected two single exposure images at distinct crank angles.
• An optical water analog engine was built

• A triggering system was created to signal the PIV system at specific crank angles

• Two images were taken on frames 4 and 5 after the system was triggered. One laser was pulsed on frame 4 and one of frame 5

• The images were converted to velocity vector fields by an Adaptive Cross Correlation algorithm implemented into a Matlab toolbox by Usera (1999)

• 200 images were ensemble averaged at each measurement position

• The Reynolds stresses, mean velocities and average vorticity at each measurement position were calculated
Chapter 3

Results

To characterize the large and small scale mixing, the ensemble averaged U and V velocity fields, ensemble averaged Reynolds stress contours and ensemble averaged vorticity contours have been calculated. In this study, large scale characteristics are those that are larger than the measurement area. Small scale characteristics are those that are smaller than the measurement area. Large scale mixing is defined as a relatively uniformly distributed Reynolds stress or vorticity contour. Small scale mixing is defined as a non-uniformly distributed Reynolds stress or vorticity contour.

Ensemble averaging is used in KIVA models (Amsden et al., 1985). In order to create a database for comparison with KIVA models, ensemble averaging has been used in this study. It is recognized that ensemble averaging typically overpredicts the amount of velocity fluctuation, particularly in the centres of vortices (Catania & Mittica, 1989).

In each figure, the X and Y measurement area co-ordinate of the measurement locations are given. This measurement area coordinate is at the bottom left hand side of the mean velocity fields, Reynolds stress contours and vorticity contours. An X dimension of a ranging from 0 to 11 mm and a Y dimension of b ranging from 0
to 7 mm has been used to characterize the X and Y distance from the measurement coordinate.

A subfigure at the bottom of each page of figures indicates the location of each velocity field within the flow cell, where the piston face is with respect to the measurement positions and the direction and velocity of the piston motion.

3.1 Mean Velocity Fields

The mean velocity fields were obtained by ensemble averaging 200 instantaneous velocity fields at each measurement location. The average of the velocity fields are shown as a function of crank angle on Figure 3.11. The mean velocity vector fields are shown in Figures 3.12 to 3.20. The mean velocity fields are shown to characterize the large scale flow motion within the flow cell.

The clearance between the two valves is 10 mm. The middle 35 mm measurement position is then 3.5 jet diameters away from the top of the flow cell and in a location close to the core of the jet. The 75mm measurement position is 7.5 jet diameters away from the top of the flow cell. The two positions were chosen to see the evolution of the flow fields and mixing characteristics both near and away from the valves at different crank angles.

Flow visualization studies by Ekchian and Hoult (1979) indicated that the in-cylinder flow motion of a single valved piston chamber was dominated by two large scale counter rotating vortices. The centres of the large scale vortices were about half-way between the valve face and piston face but would change position between cycles. The size of the vortices were approximately 2/3 of the piston displacement. These vortices would break down near the end of the intake stroke and the flow would become more uniform. Near the piston surface, the flow motion was more uniform
and in the direction of the piston motion (Davis, 1999).

The mean velocity fields as a function of crank angle are shown in Figure 3.11. It is interesting to note that at all of the measurement positions, the mean U velocity component approaches the mean piston speed \( (54.2 \text{ mm}) \) at 270 CAD. The V velocity at 270 CAD approaches zero. This is in agreement with the findings of Ekchian and Hoult (1979) who found that the flow would become more uniform after the large scale vortices broke down near the end of the intake stroke. The U velocities at all of the measurement positions are typically an order of magnitude larger than the V velocities except at position (d). The U velocity components at the top and bottom positions (a), (b), (e) and (f) show similar trends. At position (a) there is a larger U velocity at 60 CAD that is not apparent at position (e). At approximately 160 to 180 CAD (BDC), the piston velocity was shown to accelerate; at the top and bottom positions (a), (b), (e) and (f), the U velocity is near its maximum value and then starts to decrease as the piston changes direction at 180 CAD. At position (c), the U and V velocity components change direction at approximately 210 CAD which is 30 CAD after BDC. At position (d), the U velocity changes sign at approximately 150 CAD which is 30 CAD before BDC.

![Figure 3.1: Flow Orientation at 60 CAD](image)

The mean velocity fields at 60 CAD are shown in Figure 3.12. Positions (a)
and (c) are the top and bottom position respectively. Position (b) is the middle position. Positions (a) and (c) are both directed towards the left and have some obvious curvature. Position (a) has counter-clockwise curvature and position (c) has clockwise curvature. The curvature of the velocity fields indicates that there is some large scale vorticity at these positions whose scale is larger than the measurement area. Position (b) is directed to the right with some clockwise curvature.

Figure 3.2: Flow Orientation at 90 CAD

Figure 3.13 shows the mean velocity fields at 90 CAD. At position (c), the mean flow is directed to the right and parallel with the piston motion. At position (d), the mean flow is directed on a angle to the piston motion. At 75 mm (7.5 jet diameters), the centre of the jet has interacted with both the top and bottom walls causing a waviness in the mean velocity field. This is shown by the change in direction of the mean velocity vectors from position (c) to position (d). At positions (a) and (e), the mean flow is directed to the left and is curved towards the center of the flow cell. Positions (b) and (e) show that the velocity vectors are directed towards the wall. There is a clear separation at both positions where the U velocity changes sign. The separation can be seen at a=6 mm at position (b) and a=5 mm at position (f). To the right of these points, the flow is directed towards the piston and to the left of these points the flow is directed towards the left. At position (f), there is a clearer
separation between the negative U velocity and positive U velocity. The flow going
towards the piston face (positive U velocity) is more uniform at position (f) than the
flow going towards the valves (negative U velocity). This suggests that defining the
Reynolds number with respect to piston velocity may be flawed as the flow does not
respond uniformly to piston.

![Figure 3.3: Flow Orientation at 120 CAD](image)

Figure 3.3: Flow Orientation at 120 CAD

Figure 3.14 shows the mean velocity fields at 120 CAD. Positions (a) and (e) show
the velocity vectors pointing towards the left and are curved towards the centre of the
flow cell. The vectors at position (d) are directed up and to the right (similar to Figure
3.13). It is apparent that at this position, there is still an oscillation of the mean flow.
At position (c), there is some obvious curvature of the mean flow. Positions (b) and
(e) clearly show curvature in their mean velocity fields. This indicates that there is
large scale vorticity at these positions.

Figure 3.15 shows the mean velocity fields at 150 CAD. Positions (a) and (e) are
directed to the left and curved towards the centre of the flow cell indicating that there
is still some large scale vorticity at these two positions. The curvature of the velocity
fields at positions (b) and (f) indicates large scale vorticity at these two positions.
At position (c), there is a clear separation between two different large scale rotation
patterns along the diagonal from (a=4 mm, b=7 mm) to (a=11 mm, b=0 mm). To
the right of this diagonal, there is counter-clockwise rotation and to the left there is clockwise rotation. Position (d) also shows an interesting mean flow motion. By comparing position (d) at 150 CAD and 120 CAD, it can be seen that the mean motion had changed direction and is now directed from right to left while the piston is still translating from left to right and clockwise curvature at this position.

Figure 3.5: Flow Orientation at 180 CAD

Figure 3.16 shows the mean velocity fields at 180 CAD (BDC). At 180 CAD, the piston is at BDC and the piston velocity is zero. The velocity fields at positions (a) and (e) are directed towards the left. From Figure 3.11, positions (a) and (e) both have a maximum velocity at 180 CAD of -320 mm/sec and the piston velocity was shown to accelerate from approximately 160 CAD to 180 CAD. The amount of
curvature in the velocity fields at 180 CAD at positions (b) and (e) has decreased since 150 CAD. At position (d), the velocity vectors are angled up and to the left and are fairly uniform in magnitude and direction. There is no obvious curvature at position (d). At position (c), there is a large amount of counter-clockwise curvature of the velocity vectors indicating that there is still some rotation at this position.

![Figure 3.6: Flow Orientation at 210 CAD](image)

Figure 3.6: Flow Orientation at 210 CAD

Figure 3.17 shows the mean velocity fields at 210 CAD. At 210 CAD, the piston has changed direction and it can be seen from Figure 3.17 that the majority of the velocity fields are directed to the left with a noticeably smaller degree of curvature with respect to crank angles 60 to 180. At the top and bottom positions (a), (b), (e) and (f) the average U velocities have decreased since their peak positions which occurred between 150 CAD and 180 CAD. Position (c) shows that there is are at least two large scale vortices remaining; the top right hand side of position (c) displays counter-clockwise curvature, and the bottom left hand side has some clockwise curvature. At positions (a), (b), (d), (e) and (f) the mean velocity fields are all directed towards the left.

Figure 3.18 shows the mean velocity fields at 240 CAD. The direction of flow is to the left. Positions (a) and (e) do not display any obvious curvature. Position (c) is directed up and to the left, and uniform in magnitude and direction. Positions (b) and (f) are curved towards the centre of the flow cell indicating that there is
some large scale vorticity at these positions. Position (d) has some counter-clockwise curvature. It can be seen from Figure 3.11 that at all of the measurement positions, the mean U velocity is approaching the mean piston velocity at 240 CAD.

At 270 CAD, Figure 3.19 shows that the flow is generally to the left. Position (b) indicates that there is a small amount of rotation at this measurement position. Position (d) shows that there is a small amount of clockwise curvature, similar to position (f). Positions (a), (b) and (e) have relatively uniform velocity vectors that are oriented towards the valves. At 270 CAD, the mean U velocity is approaching the mean piston velocity ($54.2 \text{ mm/s}$).

At 300 CAD, Figure 3.20 shows that the flow is directed towards the valves. At
position (a) the flow is directed downwards and to the left. At positions (b) and (c) the flow is directed upwards and to the left. There is a small amount of curvature of the velocity field at position (c).

Figure 3.10: Large Scale Vorticity within the Flow Cell

Ekchian and Hoult (1979) found that there were two large scale counter-rotating vortices that persisted until the end of the intake stroke. The velocity vectors at the top 35 mm position were oriented towards the left and curved towards the centre of the flow cell indicating that there is a large scale counter-clockwise rotating vortex at this position. The velocity vectors at the bottom 35mm position were oriented towards the left and curved towards the centre of the flow cell as well, indicating that there is a large scale clockwise rotating vortex at this position. The velocity vectors
at the centre 35 mm position were to the right, indicating that the two large scale vortices passed through the centre–line of the flow cell. A figure of the overall flow field during the early part of the intake stroke (up to 150 CAD) can be seen in Figure 3.10. At 85mm, the flow was seen to oscillate. This was likely due to interaction of the top and bottom vortices causing an unsteady flow pattern downstream of the 35 mm location. The velocity fields from 150 to 210 CAD show that there is a significant amount of rotation through the centre positions. At this point, the large scale vortices were expected to break down (Ekchian & Hoult, 1979; Khalighi & Huebler, 1988). After 210 CAD, the velocity vectors are generally to the left and at 270 CAD the mean $U$ velocities were shown to approach the mean piston velocity. This indicates that the two large scale vorticities have broken down and that the flow becomes more uniform near 270 CAD. It is uncertain that these two large scale motions are symmetric.
Figure 3.11: Mean Velocity Fields as a Function of Crank Angle
Figure 3.12: Mean Velocity Fields at 60 CAD
Figure 3.13: Mean Velocity Fields at 90 CAD
Figure 3.14: Mean Velocity Fields at 120 CAD
Figure 3.15: Mean Velocity Fields at 150 CAD
Figure 3.16: Mean Velocity Fields at 180 CAD
Figure 3.17: Mean Velocity Fields at 210 CAD
Figure 3.19: Mean Velocity Fields at 270 CAD
Figure 3.20: Mean Velocity Fields at 300 CAD
3.2 Mean Vorticity

The Reynolds stresses and vorticity were presented as grey scale contours. In order to discern the spatial features of the contours, each contour had a color ranging from white to black. There was a colorbar adjacent to the contour indicating the corresponding Reynolds stress or vorticity magnitude.

The degree of mixing due to vorticity is characterized by the intensity of the vorticity and also the spatial distribution of the different magnitudes of vorticity. Large scale mixing is characterized by uniform vorticity contours. Small scale mixing is characterized by non-uniform contours. A high degree of either large or small scale mixing is characterized by a high average vorticity.

Figure 3.21 shows the evolution of mean vorticity at each measurement position as a function of crank angle degrees. At all of the positions, the average vorticity approaches zero after 210 CAD. This is in agreement with the results of Ekchian and Hoult (1979) who found that the large scale vortices break down late in the intake stroke. This is also in agreement with the suggestion that the flow becomes more uniform as the mean U velocities approach the mean piston velocity at 270 CAD. At position (a), there is a large negative vorticity at 60 CAD. At position (a), the vorticity is typically negative, until it approaches zero at 270 CAD. This is in agreement with the counter-clockwise curvature found in the velocity fields at position (a). At position (e), the vorticity is typically positive and peaks at 180 CAD (18 1/2). This is in agreement with the clockwise curvature found in the mean velocity fields at position (e). At position (f), the vorticity is also typically positive which is in agreement with the curvature of the mean velocity fields found at position (f). At position (b), the vorticity is negative and peaks at 150 CAD (-48 1/2). The magnitude of the vorticity at positions (b) and (f) are approximately 3 times larger than the
vorticity found at positions (a) and (e) respectively. The magnitude of the vorticity peak at position (b) and position (f) are similar, and both occur at 150 CAD. This indicates that there is some correlation between the vorticity at these two positions. The vorticity at position (c) is greater than the vorticity at position (d) indicating that there is greater mixing at position (c). At the top and bottom positions (a), (b), (e) and (f) the vorticity peaks between 150 CAD and 180 CAD after which, the vorticity decreases and eventually approaches zero. This indicates that the mean vorticity found at these positions begins to break down after the piston reaches BDC and the flow becomes uniform at around 270 CAD.

The vorticity contours for 60 CAD can be seen in Figure 3.22. There is a large negative vorticity from b=7 mm until approximately b=1 mm from the wall at position (a) indicating a significant amount of large scale mixing at this position. Near the wall, position (a) has a large positive vorticity indicating that there is a boundary layer. Position (a) has an average vorticity of $-38 \frac{1}{2}$. Position (b) is characterized by a vorticity contour that is strongest at the bottom left hand side of the measurement area. The average vorticity at position (b) is $-10 \frac{1}{2}$ which is four times less than position (a). Position (c) has a positive vorticity from b=1 mm until b=7 mm away from the wall with little negative vorticity in this region. Near the wall, position (c) has a strong negative vorticity indicating a boundary layer near the wall. The average vorticity at position (c) is $3 \frac{1}{2}$ which is 13 times less than position (a). Thus, at 60 CAD, position (a) has the strongest average vorticity and the vorticity at position (c) is spread out the most within the measurement area.

At 90 CAD, the vorticity contours can be seen in Figure 3.23. Position (c) has the highest average vorticity ($-16 \frac{1}{2}$) at 90 CAD. Position (c) has a larger negative component of vorticity at 90 CAD than at 60 CAD. The vorticity field at position (c) at 90 CAD is more non-uniform than at 60 CAD indicating that there is a large
amount of localized mixing at this position. It is also more non-uniform than positions (a) and (e). This indicates that there is a larger amount of local mixing at position (c) than positions (a) and (e); positions (a) and (e) are more uniform and are associated with large scale mixing instead of local mixing. At position (a) has a negative vorticity away from the wall and a localized positive vorticity near the wall indicating that there is a boundary layer near the wall. Position (d) is more evenly spread out across the measurement area than at position (c) with a comparatively smaller average vorticity. Positions (b) and (f) show that there is significant positive vorticity in the bottom half of the flow cell and negative vorticity in the upper half. The vorticity has become less uniform at positions (b) and (f) than (a) and (e) respectively with a comparatively larger average vorticity. This indicates that there is a larger amount of mixing at positions (b) and (f) than (a) and (e). Near the wall at positions (a) and (b) there is a strong positive vorticity indicating that there is a boundary layer at these positions. Near the wall at positions (e) and (f) there is a strong negative vorticity indicating that there is a boundary layer at these positions as well. Positions (a) and (e) have a stronger boundary layer than positions (b) and (f) indicated by the magnitude and uniformity of the vorticity near the wall at these positions.

At 120 CAD, the vorticity contours can be seen in Figure 3.24. Positions (a) has a large negative vorticity with relatively little positive vorticity. The average vorticity at position (a) is 3 times larger than position (e). The vorticity at position (e) is less uniform than position (a), indicating that position (a) is more associated with large scale mixing than position (e). Position (f) indicates that there is a strong positive vorticity at this position. Position (b) is less uniform than positions (a), (e) and (f) with an average vorticity that is approximately the same as position (a). Position (c) shows strong negative vorticity at the bottom left of the figure and a weaker positive vorticity at the upper right corner of the figure. Position (c) has the highest
average vorticity at 120 CAD. Position (d) shows that the vorticity has become very non-uniform and is not as intense as at position (c). Near the wall at positions (a) and (b) there is a strong positive vorticity indicating that there is a boundary layer at these positions. Near the wall at positions (e) and (f) there is a strong negative vorticity indicating that there is a boundary layer at these positions as well. Positions (a) and (e) have a stronger boundary layer than positions (b) and (f) indicated by the magnitude and uniformity of the vorticity near the wall at these positions.

Figure 3.25 shows the average vorticity contours at 150 CAD. At 35 mm, the vorticity contours have become less uniform than at 120 CAD. At position (b), the average vorticity peaks at 150 CAD at -48 $\frac{1}{2}$ f. The contour at position (b) has become more uniform than at 120 CAD with a vorticity that is almost 5 times greater. At position (f), the average vorticity also peaks at 150 CAD at 54 $\frac{1}{2}$ which is similar in magnitude to position (b). There is a strong positive vorticity at this position with relatively little negative vorticity. At position (d), the average vorticity is near zero and is well spread out throughout the measurement area indicating that there is little mixing at 150 CAD at position (d). Near the wall at positions (a) and (b) there is a strong positive vorticity indicating that there is a boundary layer at these positions. Near the wall at position (e) there is a strong negative vorticity indicating that there is a boundary layer at this positions as well. Positions (a) and (e) have a stronger boundary layer than position (b) indicated by the magnitude and uniformity of the vorticity near the wall at these positions.

Figure 3.26 shows the average vorticity contours at BDC or 180 CAD. The average vorticity at position (b) has been reduced by 60 % between crank angles 150 and 180. There is still some strong negative vorticity in this area, but it is not as strong as it was at 150 CAD. Positions (a) and (e) have increased in average vorticity by 60 % from 120 to 150 CAD and have become more non-uniform indicating that there
is a significant amount of localized mixing at these positions. Position (c) shows a relatively uniform negative vorticity with comparatively little positive vorticity. Position (d) has an average vorticity that is near zero and is spread out throughout the measurement area indicating that there is little mixing at 180 CAD at position (d). Near the wall at positions (a) and (b) there is a positive vorticity indicating that there is a boundary layer at these positions. Near the wall at positions (e) and (f) there is a negative vorticity indicating that there is a boundary layer at these positions as well. The degree of vorticity near the wall has decreased since 150 CAD, indicating that the boundary layer is not as strong as it was at 150 CAD. This is because the piston has zero velocity at 180 CAD. The boundary layer is also less uniform than at 150 CAD.

Figure 3.27 shows the average vorticity contours at 210 CAD. Position (a) has a near zero average vorticity. Near the wall, there is still some strong positive vorticity. Away from the wall there is a mix of a weaker negative and positive vorticity. Position (b) has a small average negative vorticity that is relatively non-uniform. Position (c) has an average vorticity of $-9 \frac{1}{2}$. The top of position (c) has a strong negative vorticity, and the bottom has a strong positive vorticity. At positions (d), (e) and (f) the vorticity contours are very non-uniform and average to near zero indicating that there is little mixing in these positions. Near the wall at positions (a) and (b) there is a positive vorticity indicating that there is a boundary layer at these positions. Near the wall at positions (e) and (f) there is a negative vorticity indicating that there is a boundary layer at these positions. The boundary layers are less uniform than at 150 CAD.

Figure 3.28 shows the average vorticity contours at 240 CAD. It is apparent at 240 CAD that at all of the measurement positions, the vorticity has spread out non-uniformly throughout each region. The average vorticity at each measurement region
is near zero indicating that there are no large scale rotations passing through any of the measurement areas. This is in agreement with the velocity fields which had no obvious curvature at 240 CAD. Near the wall at positions (a) and (b) there is a strong positive vorticity indicating that there is a boundary layer at these positions. Near the wall at position (f) there is a strong negative vorticity indicating that there is a boundary layer at this positions as well. The boundary layers are more uniform than 210 CAD.

Figure 3.29 shows the average vorticity contours at 270 CAD. At all of the measurement locations, the vorticity is spread out non-uniformly throughout each region. The average vorticity in each region is near zero except at position (e) where there exists a small average positive vorticity. The range in vorticity has remained the same since 240 CAD. Near the wall at position (a) there is a strong positive vorticity indicating that there is a boundary layer at this positions. The boundary layers are less uniform and less intense than at 240 CAD.

At 300 CAD, Figure 3.30 shows the average vorticity contours at 300 CAD. At each of the measurement locations, the vorticity range has gone down since 270 CAD indicating a smaller range vorticity scales. The average vorticity at all of the positions are near zero and they are more non-uniformly distributed within each of the measurement areas than at 270 CAD. Near the wall at position (a) there is a strong positive vorticity indicating that there is a boundary layer at this positions. Near the wall at position (c) there is a strong negative vorticity indicating that there is a boundary layer at this positions as well. The boundary layers are very non-uniform at 300 CAD.

It is apparent that at the centreline, position (c) offers more of a potential than position (d) for mixing. At position (c), the average vorticity is larger and more uniform than position (d) indicating that there would be a larger amount of large
scale mixing at position (c) than position (d). Positions (b) and (f) have a larger average vorticity than positions (a) and (e) respectively indicating that there is larger amount of large scale mixing at positions (b) and (f). Positions (a) and (e) have a stronger and more uniform boundary layer during the intake stroke than position (b) and (f). During the exhausting stroke, the boundary layers at all of the positions are similar in magnitude and are more non-uniform than during the intake stroke. By comparing position (c) with positions (b) and (f), it is apparent that position (c) has a larger and more uniform average vorticity until 150 CAD indicating that position (c) would offer better mixing potential early in the intake stroke than positions (b) and (f).
Figure 3.21: Mean Vorticity as a Function of Crank Angle
Figure 3.22: Mean Vorticity at 60 CAD
Figure 3.23: Mean Vorticity at 90 CAD
Figure 3.24: Mean Vorticity at 120 CAD
Figure 3.25: Mean Vorticity at 150 CAD
Figure 3.26: Mean Vorticity at 180 CAD
Figure 3.27: Mean Vorticity at 210 CAD
Figure 3.28: Mean Vorticity at 240 CAD
Figure 3.29: Mean Vorticity at 270 CAD
Figure 3.30: Mean Vorticity at 300 CAD
3.3 Mean Reynolds Stress $\bar{u}'u'$

The mean Reynolds stresses are shown as a function of crank angle in Figure 3.31. The Reynolds stress $\bar{u}'u'$ component is larger than the Reynolds stress $\bar{v}'u'$ by a factor of approximately 2. The Reynolds stress $\bar{u}'v'$ component is the smallest in all of the cases and is near zero. At all of the measurement locations, the Reynolds stresses approach zero as the piston approaches 270 CAD. This is in agreement with the vorticity curves and velocity fields which showed that there was more uniform flow and little mixing at 270 CAD. It can be seen from Figure 3.31 that at position (c), the Reynolds stress curves peak between 90 and 120 CAD. At positions (a), (e) and (b), the Reynolds stress curves peak at 150 CAD. At position (e), the Reynolds stress curves peak at 210 CAD. At position (d), the Reynolds stress curves peak at 120 CAD, but are much lower in magnitude than at position (c). From Figure 3.31, it is apparent that there is a significantly higher Reynolds stress at position (c) early in the intake stroke from 60 CAD to 120 CAD. At positions (a), (b), (e) and (f) there is significantly higher Reynolds stress at crank angles 150 to 210 CAD. At positions (d), there is a relatively low Reynolds stress throughout the piston displacement indicating that there is little mixing at this position.

Figure 3.32 shows the Reynolds stress $\bar{u}'u'$ contours at 60 CAD for the 35 mm positions. The contours are fairly uniform, indicating that there is large scale mixing at these positions due to the fluctuating u velocity component. At position (c), the Reynolds stress range is smaller than at positions (a) and (b) indicating that there is less mixing at this position.

At 90 CAD, Figure 3.33 shows the Reynolds stress $\bar{u}'u'$ contours. Position (c) has the highest average Reynolds stress at 90 CAD and is spread out uniformly throughout the measurement area indicating that there is a significant amount of
large scale mixing at this location. Positions (a) and (e) are uniformly spread out, but have a lower average Reynolds stress magnitude indicating that there is less large scale mixing at these two locations than position (c). Positions (b), (d) and (f) are non-uniformly distributed within the measurement areas indicating a more localized or small scale mixing. The average Reynolds stress at these positions is lower than at position (c) indicating that there is less mixing at 85 mm than at position (c).

At 120 CAD, Figure 3.34 shows the Reynolds stress $\overline{u^2}$ contours. Position (c) has the highest average Reynolds stress at 120 CAD and is spread out uniformly throughout the measurement area indicating that there is a significant amount of large scale mixing at this location. Position (a) is uniformly spread throughout the measurement area and has the second highest average Reynolds stress indicating that there is a significant amount of large scale mixing at this location. Position (e) is non-uniformly distributed and has a low average Reynolds stress, indicating that there is less mixing here than at positions (a) and (c). Positions (b), (d) and (e) show that the average Reynolds stress is lower than at 35mm and the contours are more non-uniform than positions (a) and (c) indicating that there is comparatively little small scale mixing at these locations.

At 150 CAD, Figure 3.35 shows the Reynolds stress $\overline{u^2}$ contours. At all of the positions, the Reynolds stress contours have become more non-uniform than at 120 CAD indicating that there is more localized mixing at 150 CAD. This is in agreement with the vorticity contours and mean velocity fields which showed that the organized flow patterns have started to break down by 150 CAD. At positions (a), (b) and (f), the mean Reynolds stress peaks at 150 CAD. The average Reynolds stress at positions (a), (b), (e) and (f) are comparable in magnitude and are distributed non-uniformly. This indicates a large degree of localized mixing at 150 CAD at these positions. At positions (c) and (d), the average Reynolds stress is non-uniform and
lower in magnitude than at the other positions, indicating that there is little mixing at 150 CAD at these positions.

At 180 CAD, Figure 3.36 shows the Reynolds stress $\overline{u'u'}$ contours. The Reynolds stress contours positions (a), (b), (d) and (e) are non-uniform. At positions (a), (b) and (e) the average Reynolds stresses are the highest indicating that there is a large amount of localized mixing at these locations. At positions (c) and (f), the Reynolds stress contours are more uniform indicating that there is a comparatively smaller amount of large scale mixing at these positions. Position (d) has the smallest average Reynolds stress indicating that there is little mixing at this location.

At 210 CAD, Figure 3.37 shows the Reynolds stress $\overline{u'u'}$ contours. Positions (b), (d), (e) and (f) are non-uniform. Positions (b), (f) and (e) have the highest average Reynolds stress at 210 CAD indicating that there is a large amount of localized mixing at these locations. Positions (a) and (c) have a comparatively lower Reynolds stress and are spread out more uniformly indicating that there is large scale mixing at these two positions, but the mixing is less intense. At position (d), the average Reynolds stress is at a minimum indicating that there is little mixing at this location.

At 240 CAD, Figure 3.38 shows the Reynolds stress $\overline{u'u'}$ contours. At all of the measurement locations, the Reynolds stress is non-uniformly distributed indicating that there is localized mixing at all of the locations. The average Reynolds stress magnitude has decreased at all of the measurement locations since 210 CAD indicating that the mixing at 240 is less intense than at 210 CAD.

At 270 CAD, Figure 3.39 shows the Reynolds stress $\overline{u'u'}$ contours. At all of the measurement locations, the Reynolds stress is non-uniformly distributed. The average Reynolds stress has decreased in magnitude at all of the measurement locations. This indicates that there is significantly less mixing at 270 CAD than at 240 CAD.

At 300 CAD, Figure 3.40 shows the Reynolds stress $\overline{u'u'}$ contours. At all of the
measurement locations, the Reynolds stress is non-uniformly distributed and the average Reynolds stress has decreased since 270 CAD. This indicates that there is significantly less mixing at 300 CAD than at 270 CAD at 35 mm.

From the Reynolds stress $\overline{u'w'}$ contours it is apparent that at 90 CAD, there is a significant amount of large scale mixing early in the intake stroke (60 CAD to 120 CAD) at position (c). At around 150 CAD, there is a large amount of localized mixing at the top and bottom positions with little mixing at the centre. At position (d), the Reynolds stress is low and non-uniformly distributed throughout the entire crank angle range. This indicates that there is little mixing at 75 mm in the centre. The Reynolds stress $\overline{u'w'}$ contours show that there is less mixing at 270 which is in agreement with the vorticity contours and mean velocity fields. The large scale mixing at position (c) due to the Reynolds stress $\overline{u'w'}$ early in the intake stroke is highest. The vorticity contours indicated that at position (c), there was a significant of large scale mixing early in the intake stroke. It is apparent, then, that there is a significant amount of large scale mixing early in the intake stroke at position (c) and a significant amount of localized mixing at position (b) and (f) later in the intake stroke due to both the vorticity and the Reynolds stress $\overline{u'w'}$. 
Figure 3.31: Reynolds Stresses as a Function of Crank Angle
Figure 3.32: Reynolds Stress Contour ($u'u'$) at 60 CAD
Figure 3.33: Reynolds Stress Contour ($u'u'$) at 90 CAD
Figure 3.34: Reynolds Stress Contour ($u'u'$) at 120 CAD
Figure 3.35: Reynolds Stress Contour ($u'u'$) at 150 CAD
Figure 3.36: Reynolds Stress Contour ($u'u'$) at 180 CAD
Figure 3.37: Reynolds Stress Contour ($u'u'$) at 210 CAD
Figure 3.38: Reynolds Stress Contour ($u'u'$) at 240 CAD
Figure 3.39: Reynolds Stress Contour ($u'u'$) at 270 CAD
Figure 3.40: Reynolds Stress Contour ($u'u'$) at 300 CAD
3.4 Mean Reynolds Stress $\overline{u'v'}$

The Reynolds stress $\overline{u'v'}$ contours are shown in Figures 3.41 through 3.49. The average Reynolds stress $\overline{u'v'}$ magnitudes are typical 2 times smaller than the average Reynolds stress $\overline{u'u}$ magnitudes and therefore contribute less to the overall mixing process.

It was shown in Figure 3.31 that all three components of the Reynolds stress had similar trends at each measurement location. The Reynolds stress $\overline{u'v'}$ component would approach zero near 270 CAD at all of the measurement locations. At position (c), the Reynolds stress $\overline{u'v'}$ peaked early on in the intake stroke (90 CAD to 120 CAD) at $8100 \text{ mm}^2/\text{s}^2$. At position (d), the average Reynolds stress $\overline{u'v'}$ was lower than all of the other positions at all crank angles. At positions (a), (b), (e) and (f) the average Reynolds stress $\overline{u'v'}$ peaked between 150 CAD and 210 CAD. The average Reynolds stress was greatest at position (b) at 150 CAD with a peak value of $7200 \text{ mm}^2/\text{s}^2$.

Figure 3.41 shows the Reynolds stress $\overline{u'v'}$ contours at 60 CAD. The contours show that the Reynolds stress $\overline{u'v'}$ components are uniform at 60 CAD. The average Reynolds stress is larger at positions (a) and (b) than at position (c). This indicates that there is a significant amount of large scale mixing at positions (a) and (b) and a smaller amount of large scale mixing at position (c).

Figure 3.42 shows the Reynolds stress $\overline{u'v'}$ contours at 90 CAD. Position (c) has the highest average Reynolds stress $\overline{u'v'}$ and is uniformly distributed indicating that there is a significant amount of large scale mixing at this location. Positions (a) and (e) are uniformly distributed with a smaller average Reynolds stress indicating that there is a comparatively smaller amount of large scale mixing at these locations. Position (d) is non-uniformly distributed with a small average Reynolds stress indicating that there is a small amount of local mixing at this position. Positions (b) and (f) both
have a low average Reynolds stress $\overline{v'v'}$. Position (b) is non-uniformly distributed indicating a small amount of local mixing at this position. Position (f) is more uniform indicating a small amount of large scale mixing at this position.

Figure 3.43 shows the Reynolds stress $\overline{v'v'}$ contours at 120 CAD. Position (c) has the highest average Reynolds stress at 120 CAD and is uniformly distributed indicating that there is a significant amount of large scale mixing at this position. Positions (a) and (e) have a smaller average Reynolds stress $\overline{v'v'}$ at 120 CAD and are more non-uniform indicating that there is a comparatively smaller amount of localized mixing at these two positions. Positions (b) and (d) are non-uniform with a small average Reynolds stress indicating that there is a small amount of localized mixing at these positions. Position (f) is more uniform with a low average Reynolds stress indicating that there is a comparatively low amount of large scale mixing at this position.

Figure 3.44 shows the Reynolds stress $\overline{v'u'}$ contours at 150 CAD. The Reynolds stress $\overline{v'u'}$ contours at positions (a), (b), (c), (e) and (f) are uniform. At position (d), the contour is non-uniform. Position (b) has the highest average Reynolds stress $\overline{v'u'}$ at 150 CAD. There is significant large scale mixing at this position. At positions (a), (c), (e) and (f) there is a comparatively smaller amount of large scale mixing. At position (b) there is a low amount of large scale mixing.

Figure 3.45 shows the Reynolds stress (vv) contours at 180 CAD. Position (b) has the highest amount of Reynolds stress $\overline{v'v'}$ at 180 CAD and is non-uniformly distributed indicating a significant amount of localized mixing at this position. At 35mm, positions (a), (c) and (e) at more uniformly distributed and have a lower average Reynolds stress $\overline{v'v'}$ indicating that there is a comparatively small amount of large scale mixing at 180 CAD. At positions (d) and (f) the contours are very non-uniform with a small average Reynolds stress indicating that there is a small amount of localized mixing at these locations.
Figure 3.46 shows the Reynolds stress $\overline{uv}$ contours at 210 CAD. The contours are uniform at positions (a), (c) and (e) and very non-uniform at position (b), (d) and (f). At positions (a) and (c) the average Reynolds stress $\overline{uv}$ has decreased since 180 CAD indicating that there is less large scale mixing at these two positions than at 180 CAD. At position (d), the contour is very non-uniform and the average Reynolds stress is low indicating that there is a small amount of localized mixing at this position. At positions (b) and (f), the contours are non-uniform with a larger average Reynolds stress $\overline{uv}$ indicating that there is a significant amount of small scale mixing at these two positions. At position (e), there is a large average Reynolds stress indicating that there is a significant amount of large scale mixing at this position.

At 240 CAD, 3.47 shows the Reynolds stress $\overline{uv}$ contours. At positions (b), (d) and (e) the contours are very non-uniform with a low average Reynolds stress indicating that there is a small amount of localized mixing at these positions. At positions (a), (c) and (f) the contours are more uniform with a low average Reynolds stress indicating that there is a small amount of large scale mixing at these positions.

Figure 3.48 shows the Reynolds stress $\overline{uv}$ contours at 270 CAD. Positions (a), (b) and (f) are relatively uniform and have low average Reynolds stresses indicating that there is a small amount of large scale mixing at these positions. In the middle, at positions (c), (d) and (e) the contours are non-uniform and have a low average Reynolds stress indicating that there is a small amount of localized mixing at these positions.

Figure 3.49 shows the Reynolds stress $\overline{uv}$ contours at 300 CAD. At the top and bottom positions (a) and (c) the contours are relatively uniform with a low average Reynolds stress $\overline{uv}$ indicating that there is a small amount of large scale mixing at these positions. At position (b), the contour is very non-uniform with a low average Reynolds stress indicating that there is a small amount of localized mixing at this
position.

From the Reynolds stress $\overline{u'v'}$ contours, it is apparent that at position (c) there is a significant amount of large scale mixing from 90 CAD to 120 CAD. At positions (b), (f), (a) and (e), there is a large amount of more localized mixing between 150 CAD and 210 CAD. At position (d), there is a small amount of localized mixing throughout all of the crank angles. At all of the measurement positions, the Reynolds stress $\overline{u'v'}$ component approaches zero indicating that there is little mixing at 270 CAD which is in agreement with the mean velocity fields, vorticity contours and Reynolds stress $\overline{u'v'}$ contours.
Figure 3.41: Reynolds Stress Contour \((v'v')\) at 60 CAD
Figure 3.42: Reynolds Stress Contour ($v'^2$) at 90 CAD
Figure 3.43: Reynolds Stress Contour ($v'v'$) at 120 CAD
Figure 3.44: Reynolds Stress Contour ($v'v'$) at 150 CAD
Figure 3.45: Reynolds Stress Contour \((u'v')\) at 180 CAD
Figure 3.46: Reynolds Stress Contour ($v'v'$) at 210 CAD
Figure 3.47: Reynolds Stress Contour ($v'v'$) at 240 CAD
Figure 3.48: Reynolds Stress Contour ($v'v'$) at 270 CAD
Figure 3.49: Reynolds Stress Contour ($v'v'$) at 300 CAD
3.5  Mean Reynolds Stress $u'v'$

The Reynolds stress $u'v'$ component is an order of magnitude less than the Reynolds stress $u'u'$ component and therefore contributes an order of magnitude less to the mixing process than the Reynolds stress $u'u'$ component. The Reynolds stress $u'v'$ contours are shown in Figures 3.50 through 3.58. The Reynolds stress $u'v'$ component approaches zero near 270 CAD at all of the measurement locations. This is in agreement with the mean velocity fields, vorticity contours, Reynolds stress $u'u'$ contours and Reynolds stress $v'v'$ contours. The Reynolds stress $u'v'$ at position (c) peaked at 120 CAD. At positions (b) and (f), the Reynolds stress $u'v'$ peaked at 150 CAD. At position (e), there was a noticeable peak at 180 CAD.

Figure 3.50 shows the Reynolds stress $u'v'$ contours at 60 CAD. The contours show that the Reynolds stress $u'v'$ components are non-uniform at 60 CAD. The average Reynolds stress is larger at positions (a) and (b) than at position (c). This indicates that there is some small scale mixing at 35 mm at 60 CAD at positions (a), (b) and (c). The mixing is larger at positions (a) and (b) than position (c).

Figure 3.51 shows the Reynolds stress $u'v'$ contours at 90 CAD. Position (c) has the highest average Reynolds stress $u'v'$ and is non-uniformly distributed indicating that there is small scale mixing at this location. Positions (a) and (f) are uniformly distributed with a smaller average Reynolds stress indicating that there is a comparatively smaller amount of large scale mixing at these locations. Positions (b), (d) and (e) are non-uniformly distributed with a small average Reynolds stress indicating that there is a small amount of local mixing at this position.

Figure 3.52 shows the Reynolds stress $u'v'$ contours at 120 CAD. Position (c) has the highest average Reynolds stress at 120 CAD and is non-uniformly distributed indicating that there is a significant amount of small scale mixing at this position.
relative to the other positions. Positions (a), (b) and (e) have a smaller average Reynolds stress $u'\bar{v}'$ at 120 CAD and are more non-uniform indicating that there is a comparatively smaller amount of localized mixing at these two positions. Positions (d) and (f) are more uniform with a small average Reynolds stress indicating that there is a small amount of large scale mixing at these positions.

Figure 3.53 shows the Reynolds stress $\bar{u}'\bar{v}'$ contours at 150 CAD. The Reynolds stress $\bar{u}'\bar{v}'$ contours at positions all of the positions are non-uniform. The maximum Reynolds stress $\bar{u}'\bar{v}'$ at 150 is at positions (b) and (e) which indicates that there is a significant amount of localized mixing at these positions relative to the other positions. At the other positions, the contours are non-uniformly distributed with a small average Reynolds stress $\bar{u}'\bar{v}'$ indicating that there is a comparatively smaller amount of localized mixing.

Figure 3.54 shows the Reynolds stress $\bar{u}'\bar{v}'$ contours at 180 CAD. Position (c) is relatively uniform with a small average Reynolds stress $\bar{u}'\bar{v}'$ indicating that there is a small amount of large scale mixing at this position. The remaining positions have a lower average reynolds stress and are non-uniformly distributed within the measurement area indicating that there is a comparatively small amount of localized mixing at these locations.

Figure 3.55 shows the Reynolds stress $\bar{u}'\bar{v}'$ contours at 210 CAD. The contours are non-uniform at all of the measurement positions. The maximum Reynolds stress $\bar{u}'\bar{v}'$ at 210 CAD is at positions (b) indicating that there is a significant amount of small scale mixing at this position relative to other positions. The rest of the positions have a smaller average Reynolds stress with a non-uniform distribution indicating that there is a small amount of localized mixing at these positions.

At 240 CAD, 3.56 shows the Reynolds stress $\bar{u}'\bar{v}'$ contours. At all of the measurement positions, the contours are non-uniform with a small average Reynolds stress
indicating that there is a small amount of localized mixing at these positions. At positions (a) and (c) the contours are more non-uniform that at 210 CAD.

Figure 3.57 shows the Reynolds stress $u'v'$ contours at 270 CAD. At all of the measurement positions, the contours are non-uniform with a small average Reynolds stress indicating that there is a small amount of localized mixing at these positions.

Figure 3.58 shows the Reynolds stress $u'v'$ contours at 300 CAD. At all of the measurement positions, the contours are non-uniform with a small average Reynolds stress indicating that there is a small amount of localized mixing at these positions. The average Reynolds stresses at 300 CAD are lower than at 270 CAD indicating that the mixing is less intense at 300 CAD.

From the Reynolds stress $u'v'$ contours, it is apparent that at position (c) there is a large amount of large scale mixing from 90 CAD to 120 CAD relative to other positions. At positions (b), (f), (a) and (e), there is a large amount of more localized mixing at 150 CAD to 210 CAD. At position (d), there is a small amount of localized mixing throughout all of the crank angles. At position (c), there is a significant amount of large scale mixing due to the vorticity and all three Reynolds stress terms early on in the intake stroke. Near the wall, there is a significant amount of localized mixing later on in the intake stroke due to vorticity and all three Reynolds stress terms from 150 to 210 CAD.
Figure 3.50: Reynolds Stress Contour ($u'v'$) at 60 CAD
Figure 3.51: Reynolds Stress Contour ($u'v'$) at 90 CAD
Figure 3.52: Reynolds Stress Contour ($u'v'$) at 120 CAD
Figure 3.53: Reynolds Stress Contour ($u'u'$) at 150 CAD
Figure 3.54: Reynolds Stress Contour ($u'v'$) at 180 CAD
Figure 3.55: Reynolds Stress Contour $(u'v')$ at 210 CAD
Figure 3.56: Reynolds Stress Contour \((u'u')\) at 240 CAD
Figure 3.57: Reynolds Stress Contour \((u'u')\) at 270 CAD
Figure 3.58: Reynolds Stress Contour ($u'v'$) at 300 CAD
Chapter 4

Conclusions

This thesis has investigated the mixing in a dual valve water analog engine at six different measurement positions throughout the intake and exhaust stroke. The mean velocity fields, Reynolds stresses and vorticity were presented and insight was given into the large and small scale mixing found at the measurement positions.

4.1 Discussion

PIV measurements were performed within the water analog engine at six different measurement positions. Measurements were taken 35 mm and 75 mm from the face of the inlet plate of the piston chamber at 3 different positions: the centreline, the top of the piston chamber and the bottom of the piston chamber. At the 35 mm position, 200 image pairs were collected at 60, 90, 120, 150, 180, 210, 240, 270 and 300 CAD. At the 75 mm position, 200 image pairs were collected at 90, 120, 150, 180, 210, 240 and 270 CAD. An adaptive cross correlation algorithm implemented into a Matlab toolbox by Usera (1999) was used to calculate and validate the velocity fields. Each velocity field was 11 mm by 7 mm. There were 45 by 27 velocity vectors in each fields
with a 0.23 mm separation between vectors. The 200 velocity fields were ensemble averaged, and the Reynolds stresses and vorticity were calculated at each position at each crank angle.

The mean velocity fields showed that there was a significant large scale rotation from 60 CAD to 150 CAD. There were two counter-rotating vortices. The top vortex was rotating counter-clockwise and the bottom vortex was rotating clockwise. At 150 to 180 CAD, there was significant rotation in the centreline of the piston chamber with less average vorticity at the top and bottom position indicating that the two counter-rotating vortices had distorted. At 270 CAD, the mean U velocity approached the mean piston velocity and the vorticity approached zero at all of the measurement positions indicating that the flow became more uniform and the vortices had broken down. Since the mean velocity fields did not follow the piston motion, the definition of Reynolds number as a function of the average piston velocity may be flawed.

The vorticity at all of the measurement positions converged to near zero at 270 CAD with a non-uniform distribution within the measurement areas indicating that there is relatively uniform flow and little mixing due to vorticity at 270 CAD. The vorticity contours indicated that there was a significant amount of large scale mixing at 35 mm in the centre from 60 CAD until 120 CAD. From 150 to 210 CAD, the top and bottom positions had a large average vorticity that was non-uniform within the measurement area. This indicated that at 150 to 210 CAD, there was a large local mixing due to vorticity at the top and bottom positions. At 75 mm in the centre, there was a low average vorticity that was non-uniformly distributed within the measurement area indicating that there was little local mixing due to vorticity at this position. There was a boundary layer found near the wall at the top and bottom positions. The boundary layer was stronger and more uniform at the 35 mm position than at the 75 mm position during the intake stroke. During the exhaust stroke, the
boundary layer was less uniform and less intense than during the intake stroke.

The Reynolds stress contours showed that at all positions, the Reynolds stress $u'u'$ component was approximately twice as large as the Reynolds stress $v'v'$ component. The Reynolds stress $u'u'$ component was an order of magnitude less than the Reynolds stress $u'u'$ component indicating that the Reynolds stress $u'u'$ component contributed significantly less to the overall mixing process. At all positions, the three Reynolds stress components converged to near zero at 270 CAD indicating that there is little mixing at this position. At 35 mm in the centre, the Reynolds stress $u'u'$ and $v'v'$ components were larger than the other positions from 90 CAD until 120 CAD and were spread out uniformly indicating that there is a large amount of large scale mixing at this position from 90 until 120 CAD. From 150 until 210 CAD, the top and bottom positions at 35 mm and 75 mm had a large average Reynolds stress and were spread out non-uniformly within the measurement areas indicating that there is a large amount of localized mixing at these positions from 150 until 210 CAD. At 75 mm in the centre, there was a low average Reynolds stress with a non-uniform distribution within the measurement area indicating that there is a small amount of localized mixing within this measurement area.

It is apparent that at the 35 mm location in the centre, there is a significant amount of large scale mixing from 60 CAD to 120 CAD by both the vorticity and Reynolds stresses. From 150 to 210 CAD, the top and bottom positions at 35 mm and 75 mm contribute significantly to the small scale mixing. The 75 mm location in the centre does not contribute significant to either small or large scale mixing. Both the average Reynolds stresses and vorticity are small at this location.

Trigui et al. (1994) found that there was a significant amount of small scale motion during the early part of the intake stroke which organized into large scale vortices later on (near BDC). This study shows that there is clearly large scale rotation early
on in the intake stroke. However, the ensemble averaging used in this study has likely blurred some of the small scale turbulent motions. The oscillation of the mean vortices is interpreted as turbulent fluctuations with the ensemble averaging process. As a result, the turbulent fluctuations have a higher magnitude and are larger in scale than cyclic averaging.

This work has also shown that the Reynolds stresses have distinct peaks that correlated with the mean vorticity peaks. This may indicate that the oscillation of the mean vorticity is contributing substantially to the turbulent fluctuations. This should be investigated by comparing the ensemble averaged results found in this study to cyclic averaged and wavelet averaged results. In the cyclic and wavelet averaged results, the overall turbulence levels will decrease and the correlation between vorticity and Reynolds stresses will be reduced.

This work has also shown that the Reynolds stresses at position (c) early on in the intake stroke are strongest in the corner of the measurement location. This indicates that there is a gradient in both the X and Y direction causing both a stretching of the vortices and indicating that the turbulent vorticity is large scale. The Reynolds stresses near the walls later on in the intake stroke are more non-uniform indicating that there is local stretching due to the Reynolds stress gradients as well as more small scale turbulent vorticity.

The Reynolds number based on mean piston velocity does not make sense based on the results of this study. The turbulence is clearly a function of both the piston location, the measurement location and the piston velocity. The turbulence level was seen to increase as the piston moved towards BDC and was seen to decrease as the piston approached TDC.

This work has contributed to the understanding of the mixing process within a piston driven reciprocating flow field. Insight was given into the large and small scale
mixing process and a database for validation with CFD codes has been outlined.

4.2 Recommendations

It was found that within a set of measurements, the 1/2 HP motor would become very hot after 30 minutes of data acquisition. It was then necessary to let the motor cool for over 2 hours. To compare flow field results with other researchers, it is necessary to match the piston velocity profile. Typical piston velocity profiles are sinusoidal (Heywood, 1988). The piston velocity profile in this study was a significant improvement over the work done by Davis (1999), but the profile is still has a sharp acceleration region at 180 CAD and 360 CAD. To improve this, a higher powered motor should be used.

The ensemble averaging procedure overestimates the turbulence found within non-stationary flows (Catania & Mittica, 1989). Future work should include the use of a Wavelet transform to more clearly separation the mean and turbulent quantities, as Sullivan et al. (1999) did with crank angle resolved LDV data.

Although the 162\(\mu\)sec time separation between laser pulses allowed a maximum out of plane velocity of 6.17 m/s which was much greater than the maximum U component of velocity, the three dimensional velocity field is important to consider. The third component of velocity would allow the validation of velocity fields by using the continuity equation for an incompressible fluid,

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} + \frac{\partial W}{\partial Z} = 0$$  (4.1)

where W is the out of plane velocity in the Z direction. A third validation criteria, after the eye validation and maximum displacement criteria could be set by determin-
ing the deviation of the above equation from zero. If the left hand side of Equation 4.1 did not sum to under \( \pm A \), where \( A \) is some pre-defined threshold, the velocity vector would be rejected.

This engine was not intended to completely simulate a production engine. However, the mean velocity fields and vorticity information did demonstrate features that were similar to the results of Ekchian and Hoult (1979). Experimental and computational studies should determine the differences between the flow fields in a square and cylindrical piston chamber. The assumptions that Khalighi and Huebler (1988) made in order to detail the list of similarity parameters used to compare a water analog engine and a production engine should be further investigated experimentally or computationally.

The techniques used to collect and analyze the data in this study are well suited for lower speed cyclic flows, e.g. biological flows such as air flow through the nasal cavity and blood flow through the heart. PIV has been used in these areas, but the unique triggered data collection method could be used to correlate the blood flow with the position of the heart valve at a fraction of the cost of traditional complete PIV system packages. The adaptive cross-correlation algorithm could be used to increase the spatial resolution of the PIV measurements while keeping a high maximum resolvable velocity. The adaptive cross-correlation algorithm is also better than the standard cross-correlation algorithm in areas where there is a high velocity gradient (Usera, 1999). This is important in areas near the wall, where high shear stresses and Reynolds stresses are typically found.

A time resolved full field study of flow process within the engine should be done. Larger seed particles with a continuous laser and a digital camera on free run mode should be used. Alternatively, dye can be injected before the valves and the flow can be visualized near the valves. Since significant local mixing was found to take
place near the valves, it would be expected that the dye would not track the flow motion downstream of the valves as it would be mixed and dispersed within the water. This type of study would show the large scale motions within the flow and some additional insight would be given into the cyclic variability of the flow structures, their breakdown and areas of recirculation.
Appendix A

Error Analysis

A.1 Methodology

The error analysis in this study follows the error analysis done by Davis (1999). The uncertainty in \( \sigma_i^2 \) of \( \zeta = f(\gamma_1, \gamma_2, \gamma_3, \ldots, \gamma_N) \) can be approximated using the following method (Shames, 1992) where,

\[
\sigma_\zeta^2 \approx \sum_{i=1}^{N} \left( \sigma_{\gamma_i}^2 \left( \frac{\partial \zeta}{\partial \gamma_i} \right)^2 \right)
\]  

(A.1)

A.2 Instantaneous Velocity Error

The instantaneous velocity of a particle, \( \bar{u}_i \), is,

\[
\bar{u}_i = \frac{d_i}{M \Delta t}
\]  

(A.2)
where \( d_i \) the particle displacement in pixels, \( M \) is the magnification factor in \( \text{pixels/mm} \), and \( \Delta t \) is the time interval between laser pulses in seconds.

From equation A.1, the uncertainty in the instantaneous velocity is,

\[
\sigma_{\ddot{u}_i}^2 = \sigma_{d_i}^2 \left( \frac{\partial \ddot{u}_i}{\partial d_i} \right)^2 + \sigma_M^2 \left( \frac{\partial \ddot{u}_i}{\partial M} \right)^2 + \sigma_{\Delta t}^2 \left( \frac{\partial \ddot{u}_i}{\partial \Delta t} \right)^2 \tag{A.3}
\]

where \( \sigma_{d_i}^2 \) is the variance of the particle displacement, \( \sigma_M^2 \) is the variance of the magnification factor and \( \sigma_{\Delta t}^2 \) is the variance in the time interval.

It can be found that,

\[
\frac{\partial \ddot{u}_i}{\partial d_i} = \frac{1}{M \Delta t} \tag{A.4}
\]

\[
\frac{\partial \ddot{u}_i}{\partial M} = \frac{-d_i}{M^2 \Delta t} \tag{A.5}
\]

\[
\frac{\partial \ddot{u}_i}{\partial \Delta t} = \frac{-d_i}{M \Delta t^2} \tag{A.6}
\]

Inserting A.4, A.5 and A.6 into equation A.3 yields,

\[
\sigma_{\ddot{u}_i} = \ddot{u}_i \sqrt{\left( \frac{\sigma_{d_i}}{d_i} \right)^2 + \left( \frac{\sigma_M}{M} \right)^2 + \left( \frac{\sigma_{\Delta t}}{\Delta t} \right)^2} \tag{A.7}
\]
Substituting $M = 69.5 \text{ pixel/mm}$, $\sigma_M = 0.18 \text{ pixel/mm}$, $\Delta t = 162 \times 10^{-6} \text{ s}$, $\sigma_{\Delta t} = 5.8 \times 10^{-10} \text{ s}$, $\sigma_{\Delta t} = 0.04 \text{ pixel}$, and noting that $d_i = \bar{u}_i M \Delta t$ equation A.7 becomes,

$$
\sigma_{\bar{u}_i} = \bar{u}_i \sqrt{\left( \frac{3.6}{\bar{u}_i} \right)^2 + 6.7 \times 10^{-6}} \tag{A.8}
$$

For the instantaneous velocity range of $5 - 350 \text{ mm/s}$ it can be found from A.8 that the change in variance is $0.11 \text{ mm/s}$. Recognizing that the first term will have the greatest influence on uncertainty, a constant uncertainty will be taken,

$$
\sigma_{\bar{u}_i} = 3.6 \text{ mm/s} \tag{A.9}
$$

### A.3 Mean Velocity Error

The mean velocity $\langle U_i \rangle$ is defined as,

$$
\langle U_i \rangle = \frac{1}{N} \sum_{k=1}^{N} (\bar{u}_i)_k \tag{A.10}
$$

where $k$ is the sample number.

The variance in the mean velocity, $\sigma_{\langle U_i \rangle}$, is found by adding the instantaneous variances $\sigma_{\bar{u}_i}$,
Using an average sample size of $N = 200$, and using Equation A.9,

$$\sigma_{(u_i)} = 0.06 \text{ mm/s} \quad (A.12)$$

### A.4 Fluctuating Velocity Error

The fluctuating velocity $u'_i$ is defined as,

$$u'_i = \bar{u}_i - \langle U_i \rangle \quad (A.13)$$

The fluctuating variance $\sigma_{u'_i}$ is,

$$\sigma_{u'_i}^2 = \sigma_{u_i}^2 \left( \frac{\partial u'_i}{\partial \bar{u}_i} \right)^2 + \sigma_{(u_i)}^2 \left( \frac{\partial u'_i}{\partial \langle U_i \rangle} \right)^2 \quad (A.14)$$

which can be simplified to,

$$\sigma_{u'_i} = \sqrt{\sigma_{u_i}^2 + \sigma_{(u_i)}^2} \quad (A.15)$$

and from A.9 and A.12,
\[ \sigma_{u_i} = 3.6 \text{ mm/s} \quad (A.16) \]

### A.5 Velocity Gradient Error

In order to determine the gradient of a function, \( \zeta \), a central, forward, and backward difference method was used. The error produced in these schemes is different. It is caused by the accuracy of the velocity measurements and the truncation error resulting from the each scheme.

The central difference scheme for the first order gradient is defined as,

\[
\left( \frac{\partial \zeta_i}{\partial x_j} \right)_k = \frac{(\zeta_i)_{k+1} - (\zeta_i)_{k-1}}{2\Delta x} + O(\Delta x^2) \quad (A.17)
\]

The forward difference scheme for the first order gradient is defined as,

\[
\left( \frac{\partial \zeta_i}{\partial x_j} \right)_k = \frac{(\zeta_i)_{k+1} - (\zeta_i)_k}{\Delta x} + O(\Delta x) \quad (A.18)
\]

The backward difference scheme for the first order gradient is defined as,

\[
\left( \frac{\partial \zeta_i}{\partial x_j} \right)_k = \frac{(\zeta_i)_k - (\zeta_i)_{k-1}}{\Delta x} + O(\Delta x) \quad (A.19)
\]

The \( O(\Delta x^n) \) represents the truncation error. For each scheme the truncation error
is found as,

\[ O(\Delta x^2) = \frac{(\Delta x)^2}{3!} \frac{\partial^3 \zeta_i}{\partial x_j^3} + \frac{(\Delta x)^4}{5!} \frac{\partial^5 \zeta_i}{\partial x_j^5} + \ldots \]  (A.20)

\[ O(\Delta x) = \frac{(\Delta x)}{2!} \frac{\partial^2 \zeta_i}{\partial x_j^2} + \frac{(\Delta x)^2}{3!} \frac{\partial^3 \zeta_i}{\partial x_j^3} + \ldots \]  (A.21)

Using \( \Delta x = 0.23 \text{mm} \) and assuming that all gradients are approximately equal, the truncation errors can be written as,

\[ O(\Delta x^2) \approx 0.01 \frac{\partial \zeta_i}{\partial x_j} \]  (A.22)

\[ O(\Delta x) \approx 0.12 \frac{\partial \zeta_i}{\partial x_j} \]  (A.23)

Gradient calculations neglect the truncation errors, therefore equations A.17, A.18 and A.19 reduce to,

\[ \left( \frac{\partial \zeta_i}{\partial x_j} \right)_k = \frac{(\zeta_i)_{k+1} - (\zeta_i)_{k-1}}{2\Delta x} \]  (A.24)
\[
\left( \frac{\partial \zeta_i}{\partial x_j} \right)_k = \frac{(\zeta_i)_{k+1} - (\zeta_i)_k}{\Delta x} \quad (A.25)
\]

\[
\left( \frac{\partial \zeta_i}{\partial x_j} \right)_k = \frac{(\zeta_i)_k - (\zeta_i)_{k-1}}{\Delta x} \quad (A.26)
\]

The variance \( \sigma_{\zeta_i}^2 \) (without the truncation error) can be calculated for each scheme using equation A.1, which for the central difference scheme, reduces to,

\[
\sigma_{\zeta_i}^2 = \frac{2}{\sigma_{\zeta_i}^2} \quad (A.27)
\]

The variance for the forward and backward scheme can be reduced to,

\[
\sigma_{\zeta_i}^2 = 2(\Delta x)^2 \sigma_{\zeta_i}^2 \quad (A.28)
\]

If the truncation error is assumed to have a gaussian distribution then the variance for this error can be written as,

\[
\sigma_e^2 = \left( \frac{0.01 \frac{\partial \zeta_i}{\partial x_j}}{0.11} \right)^2 = 0.003 \left[ \frac{\partial \zeta_i}{\partial x_j} \right]^2 \quad (A.29)
\]
\[ \sigma_i^2 = 0.012 \left[ \frac{\partial \xi_i}{\partial x_j} \right]^2 \]  
(A.30)

for the central (A.29), forward and backward schemes (A.30) respectively. The complete estimate for the variance in the gradient calculation is found by adding the first gradient estimate with the truncation estimate. For the central difference scheme the complete estimate is,

\[ \sigma_\varphi^2 = \sigma_{\nabla \xi_i}^2 + \sigma_i^2 = 0.03 \sigma_i^2 + 0.003 \left[ \frac{\partial \xi_i}{\partial x_j} \right]^2 \]  
(A.31)

and for the forward and backward schemes,

\[ \sigma_\varphi^2 = 0.1 \sigma_i^2 + 0.012 \left[ \frac{\partial \xi_i}{\partial x_j} \right]^2 \]  
(A.32)

The variance for the mean velocity gradients can now be determined by inserting equation A.12 into the error from each scheme. For the central difference scheme,

\[ \sigma_{\varphi\langle U_i \rangle}^2 = 0.0001 + 0.003 \left[ \frac{\partial \langle U_i \rangle}{\partial x_j} \right]^2 \]  
(A.33)

And for the forward and backward difference scheme,

\[ \sigma_{\varphi\langle U_i \rangle}^2 = 0.0004 + 0.012 \left[ \frac{\partial \langle U_i \rangle}{\partial x_j} \right]^2 \]  
(A.34)
For the central difference scheme, the $\sigma_{\nabla(u_i)}$ ranges from 2.73 \( \frac{1}{\xi} \) to 27.4 \( \frac{1}{\xi} \) with respect to a minimum vorticity of 50 \( \frac{1}{\xi} \) and a maximum vorticity of 500 \( \frac{1}{\xi} \) where the relative error is 5.5%.

For the forward and backward difference schemes, the $\sigma_{\nabla(u_i)}$ ranges from 5.5 \( \frac{1}{\xi} \) to 54.8 \( \frac{1}{\xi} \) with respect to a minimum vorticity of 50 \( \frac{1}{\xi} \) and a maximum vorticity of 500 \( \frac{1}{\xi} \) where the relative error is 11%.

### A.6 Mean Vorticity Error

The variance, $\sigma_\omega$, in the mean vorticity is found as,

$$
\sigma_\omega = \sigma_{\frac{\partial \omega}{\partial x}} + \sigma_{\frac{\partial \omega}{\partial y}}
$$

and from Equations A.33 and A.34, the mean vorticity has a relative error of 5.5% for the central difference scheme and 11% for the forward and backward difference schemes.

### A.7 Reynolds Stress Error

The variance in the quantity $\langle u'_i u'_j \rangle$ can be found, where $\langle u'_i u'_j \rangle$ is defined as,

$$
\langle u'_i u'_j \rangle = \frac{1}{N} \sum_{k=1}^{N} (u'_i u'_j)_k
$$

The variance in $u'_i u'_j$ is defined as,
The ensemble averaged Reynolds stress $\sigma_{u_i' u_j'}$ is determined by,

$$\sigma_{u_i' u_j'} = (u_i' u_j') \sigma_{u_i'} \sqrt{\left( \frac{1}{u_i'} \right)^2 + \left( \frac{1}{u_j'} \right)^2}$$ (A.37)

The ensemble averaged Reynolds stress $\sigma_{u_i' u_j'}^2$ is determined by,

$$\sigma_{u_i' u_j'}^2 = \sum_{k=1}^{N} \left( \sigma_{u_i' u_j'}^2 \left( \frac{\partial \langle u_i' u_j' \rangle}{\partial u_i' u_j'} \right)^2 \right)_k$$ (A.38)

$$\sigma_{u_i' u_j'}^2 = \frac{1}{N^2} \sum_{k=1}^{N} \left( \sigma_{u_i' u_j'}^2 \right)_k$$ (A.39)

where $\sigma_{u_i'}$ was found to be $3.6 \text{ mms}^{-1}$. Matlab was used to find the the value of Equation A.39, and a maximum relative error of 8% was found.
References


REFERENCES


