In-Cylinder Airflow and Fuel Spray Characteristics for a Top-Entry, Direct Injection, Gasoline Engine

Steven Begg

Volume II of II

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In-Cylinder Airflow and Fuel Spray Characteristics for a Top-Entry, Direct Injection, Gasoline Engine

Volume II of II

Steven Begg

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Nomenclature

Roman Symbols

\( C_d \)  drag coefficient of droplet in air
\( d \)  liquid droplet diameter
\( D_a \)  Damköhler number
\( D_m \)  molecular diffusivity
\( f \)  frequency
\( F_c \)  In-cylinder Turbulence 'Cut-Off' frequency
\( i \)  individual engine cycle
\( k \)  turbulent kinetic energy
\( k \)  wave number vector
\( K \)  volumetric turbulent kinetic energy (per unit mass)
\( l_e \)  Eulerian integral length scale
\( l_k \)  Kolmogorov length scale
\( l_m \)  Taylor Micro length scale
\( n \)  total number of measurements or engine cycles
\( N, N_e \)  total number of engine measurement cycles
\( N_s \)  number of evenly spaced intervals across engine cycle
\( N_t \)  total number of measurements within a defined crank angle interval
\( P_f \)  firing in-cylinder gauge pressure
\( P_m \)  motoring in-cylinder gauge pressure
\( r \)  radius of swirl
\( R_i \)  spatial or Eulerian velocity autocorrelation function
\( R_{\tau} \)  temporal or Lagrangian velocity autocorrelation function (\( R_{\tau} \))
\( R_e \)  Reynolds number
\( R_c \)  cylinder radius
\( S_L \)  laminar flame burning velocity
\( S_t \)  mean turbulent burning velocity
\( S_RN \)  swirl Reynolds number
\( t \)  instantaneous time
\( u, U(t) \)  instantaneous normal velocity
\( \ddot{U} \)  mean normal velocity
\( u(t) \)  fluctuating normal velocity component
\( u', U'_{\tau} \)  turbulence intensity
\( u_{\infty}, u_R \)  transit or residence time-weighted mean velocity
\( v \)  instantaneous tangential or radial velocity
\( V'_{\tau} \)  increase in measured velocity variance due to a velocity gradient
Greek Symbols

\( \theta \)  
- crank angle 
- [°]

\( \delta_L \)  
- laminar flame thickness 
- [mm]

\( \epsilon \)  
- rate of turbulent energy dissipation 
- \([\text{m}^2\text{s}^{-3}]\)

\( \epsilon \)  
- standard error (in mean or RMS velocity estimates) 
- \([\text{ms}^{-1}]\)

\( \Delta t \)  
- particle transit time within LDA probe volume 
- [s]

\( \sigma \)  
- surface tension 
- \([\text{kg} \cdot \text{s}^{-2}]\)

\( \sigma_p \)  
- standard deviation in particle velocity distribution across probe volume 
- \([\text{ms}^{-1}]\)

\( \rho \)  
- density 
- \([\text{kg} \cdot \text{m}^{-3}]\)

\( \nu \)  
- kinematic viscosity 
- \([\text{m}^2\text{s}^{-1}]\)

\( \Phi \)  
- energy spectrum tensor 
- \([\text{m}^3\text{s}^{-2}]\)

\( \phi \)  
- phase angle, phase slot width angle with respect to crank angle 
- [°]

\( \Phi \)  
- vorticity 
- \([\text{s}^{-1}]\)

\( \tau \)  
- time increment or delay 
- [s]

\( \tau_i \)  
- Lagrangian integral time scale 
- [s]

\( \tau_L \)  
- chemical reaction time 
- [s]

\( T_m \)  
- Taylor micro time scale 
- [s]

\( \tau_T \)  
- turbulent eddy turnover time 
- [s]

\( \omega \)  
- swirl ratio based upon the ensemble-averaged swirl velocity 
- [-]

\( \omega' \)  
- swirl ratio defined as the ratio of \( \omega \) to the angular momentum of a solid body rotation 
- [\text{rads}^{-1}]

\( \omega_e \)  
- engine speed 
- [\text{rads}^{-1}]

Optical Reference Symbols

\( \theta \)  
- light beam intersection angle 
- [°]

\( \beta \)  
- LASER beam angle of divergence 
- [mrad]

\( \phi \)  
- light scattering angle between transmission and collection axes 
- in the horizontal plane 
- [°]

\( \psi \)  
- elevation angle of collection axis in vertical plane from the horizontal plane 
- [°]

\( \alpha \)  
- angle between surface tangent and incident light ray (in Figure 3.11.) 
- [°]

\( \alpha' \)  
- angle between surface tangent and refracted light ray (in Figure 3.11.) 
- [°]
α angle between \( \vec{V} \) and measured component of \( \vec{V} \), perpendicular to the fringe pattern in the plane of the incident LASER beams\[°\]

\( \nu \) angle between incident ray and \( p \)th order ray exiting a spherical droplet\[°\]

\( \vec{V} \) particle velocity vector\[ms^{-1}\]

\( \Phi \) phase shift measured at the PDA collection optic\[°\]

\( s \) fringe spacing\[\mu m\]

\( p \) light scattering designation order within a spherical droplet\[\mu m\]

\( q \) Mie size parameter

\( S \) Mie scattering function

\( m \) refractive index

\( n \) ratio of refractive indices between scatterer and surrounding medium

\( \vec{n}_{i,2} \) unit vector in direction of incident LASER beam

\( N \) number of diffraction grating line pairs

\( N_i \) number of fringes

\( \lambda \) wavelength\[m\]

\( d \) line pair spacing of diffraction grating\[mm\]

\( d_o \) Gaussian beam diameter\[mm\]

\( d_p \) normalised particle diameter

\( r_o \) Gaussian beam radius\[mm\]

\( r_o \) minimum Gaussian beam radius at beam waist\[mm\]

\( F \) transmission lens focal length\[mm\]

\( b \) beam separation at the transmission lens\[mm\]

\( I \) light intensity\[Wm^{-2}\]

\( E \) beam expansion factor

\( F_s \) optical frequency shift\[Hz\]

\( f_r \) rotating diffraction grating speed\[rpm\]

\( f_d \) Doppler frequency\[Hz\]

\( \delta_{x,y,z} \) LDA probe volume dimensions\[mm\]

\( R \) refraction Fresnel coefficient

\( r \) reflection Fresnel coefficient

\( U \) measured component of \( \vec{V} \), perpendicular to the fringe pattern, in the plane of the beams and perpendicular to the optical axis\[ms^{-1}\]

Subscripts

\( e \) LASER beam extremities

\( E \) LASER beam centre

\( EA \) ensemble-averaged

\( HF/LF \) high frequency / low frequency

\( l \) liquid phase

\( rel \) relative velocity between phases
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AFR</td>
<td>Air-to-Fuel Ratio</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom-dead Centre</td>
</tr>
<tr>
<td>CCVS</td>
<td>Combustion Control through Vortex Stratification</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>CNC</td>
<td>Computer Numerical Control</td>
</tr>
<tr>
<td>CVI</td>
<td>Closed Valve Fuel Injection</td>
</tr>
<tr>
<td>DFI</td>
<td>Direct Fuel Injection</td>
</tr>
<tr>
<td>DFVR</td>
<td>Dynamic Flow Visualisation Rig</td>
</tr>
<tr>
<td>DMI</td>
<td>Direct Mixture Injection</td>
</tr>
<tr>
<td>DOHC</td>
<td>Dual Overhead Camshafts</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust Gas Recirculation</td>
</tr>
<tr>
<td>EOI</td>
<td>End of Injection Timing</td>
</tr>
<tr>
<td>FIE</td>
<td>Fuel Injection Equipment</td>
</tr>
<tr>
<td>G-DI</td>
<td>Gasoline Direct Injection</td>
</tr>
<tr>
<td>HSDI</td>
<td>High-Speed, Direct Injection</td>
</tr>
<tr>
<td>IC</td>
<td>Internal Combustion (Engine)</td>
</tr>
<tr>
<td>IFT</td>
<td>Inverse Fourier Transform</td>
</tr>
<tr>
<td>IGN</td>
<td>Ignition Timing</td>
</tr>
<tr>
<td>IMAP</td>
<td>Inlet Manifold Absolute Pressure</td>
</tr>
<tr>
<td>IVC</td>
<td>Inlet Valve Closure</td>
</tr>
<tr>
<td>LALLS</td>
<td>Low Angle Light Scattering</td>
</tr>
<tr>
<td>LBT</td>
<td>Ignition Timing for Leanest Burn and Best Torque</td>
</tr>
<tr>
<td>LDA</td>
<td>LASER Doppler Anemometry</td>
</tr>
<tr>
<td>LIF</td>
<td>LASER Induced Fluorescence</td>
</tr>
<tr>
<td>LII</td>
<td>LASER Induced Incandescence</td>
</tr>
<tr>
<td>LLS</td>
<td>LASER Light Sheet Illumination</td>
</tr>
<tr>
<td>LTR</td>
<td>Linear Trend Removal</td>
</tr>
<tr>
<td>MBT</td>
<td>Minimum Ignition Timing Advance for Best Torque</td>
</tr>
<tr>
<td>MPI</td>
<td>Manifold Port Injection</td>
</tr>
<tr>
<td>MPS</td>
<td>Mean Piston Speed</td>
</tr>
<tr>
<td>NOP</td>
<td>Needle Opening Pressure</td>
</tr>
<tr>
<td>NOx</td>
<td>Oxides of Nitrogen</td>
</tr>
<tr>
<td>NTA</td>
<td>Non-Stationary Time-Averaging</td>
</tr>
<tr>
<td>OVI</td>
<td>Open Valve Fuel Injection</td>
</tr>
<tr>
<td>PDA</td>
<td>Phase Doppler Anemometry</td>
</tr>
</tbody>
</table>
PFI, Port Fuel Injection
PIV, Particle Image Velocimetry
PMT, Photomultiplier Tube
SC, Stratified Charge
SCV, Swirl Control Valve
SI, Spark Ignition
SMD, Sauter Mean Diameter
SNR, Signal to Noise Ratio
SOI, Start of Injection Timing
SONL, Start of Needle Lift
SRN, Swirl Reynolds Number
TAE, Trajectory Ambiguity Effect
TAF, Time-Average Filtering
TDC, Top-dead Centre
TTVR, 'Tipping' Tumble Vortex Ratio
UBHC, Unburnt Hydrocarbon
VFAM, Viscous Flow Air Meter
WOT, Wide Open Throttle
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5.0. Characteristics of In-Cylinder Airflows in a Pent-roof, Direct Injection, Gasoline Combustion Chamber: Experimental Measurements

5.1. Introduction to Chapter 5

The introduction of G-DI operating modes and the requirement for charge stratification through considered air control strategies has further emphasised the need for precise experimental data, especially through the intake and compression strokes. The role of the air motion is twofold with regards to fuel injection and mixture preparation. In the first instance, the air motion is required to produce a homogenous air and fuel mixture that can be completely burnt after spark ignition. In the second strategy, charge air motion is required to quickly stratify the fuel distribution in the cylinder in such a way that a reduced volume of fuel is successfully consumed by combustion within an overall lean environment. These two separate tasks are governed by the nature of the mean and turbulent characteristics of the flow. The characteristics of in-cylinder airflows are complex functions of geometry; piston, combustion chamber and intake tract shapes, piston velocity, valve timing and the gas exchange processes.

Mean or bulk flow is required for large scale mixing and transport of fuel. Turbulence acts to aid mixing on a much smaller scale. Its nature is of fundamental importance at the point of spark ignition and through the development of the early combusting flame. This requires knowledge of the characteristic length and time scales of the smaller scale structures. In combustion strategies that are reliant upon repeatable charge stratification, high levels of turbulence are not favoured for cyclic stability. A well-established bulk air motion is a preferred mean of stabilising such flows.
5.2. Air Flow Measurements in a DOHC, Gasoline, Pent-roof Direct Injection Cylinder Head with Optical Access

5.2.1. Experimental Test Procedure

The description of the test procedures that follow describe only those methods specific to these particular experiments that are not covered by the overview given in Chapter 3.

5.2.1.1. Engine Configuration

Two experimental airflow measurement studies were carried out using a motored single cylinder hydra engine fitted with a Ricardo RCE161 top-entry intake, four valve, pent-roof cylinder head. A schematic of the cylinder head, piston and intake geometries are shown in Figure 5.0. The mid-valve section shows the top-entry intake port and side exit exhaust port. The mid-cylinder section shows the location and orientation of the fuel injector and spark plug. The specification of the engine is given in Table 5.0. The valve-timing diagram for this configuration is given in Figure 5.1. The piston incorporates a spherical bowl in the crown and as such was unable to be modified to provide good optical access to the chamber with the use of a window and 'tipping' mirror arrangement. The piston crown geometry and valve cut-outs are shown in Figure 5.2. The piston topland to cylinder head clearance heights were measured at TDC for the high compression cylinder head and are shown in Figure 5.3.

The basic engine configuration was as discussed in previous sections. The piston surface and chamber walls were painted matt black to minimize internal glare. The intake and exhaust systems previously described were modified to fit the top-entry intake plenum and throttle assembly. A schematic of the engine, LDA and Seeding System installation is given in Figure 5.4.

5.2.1.2. Description of Optical Access Scenarios

Optical access was provided to the combustion chamber by two means; through the spark plug orifice and through a quartz annular slice of the cylinder liner. The two optical access methods can be summarised:

**Spark Plug Optical Access.** Optical access through the spark plug orifice enabled LDA measurements to be carried out at measurement points co-linear with the spark plug axis. A Ø14 mm Dantec FibreFlow Probe with 50 mm focal length was inserted into a specialised windowed probe adapter designed and manufactured at the University of Brighton. This is shown in Figure 5.5. and was designed to allow both rotational and axial movement of the LDA probe volume about the central spark plug axis. A small polished Perspex window (9.42 mm thick) situated in the tip of the holder allowed access to the combustion chamber as shown in Figures 5.6a, b and c. The refractive index of the Perspex was 1.495 at a wavelength of 589.3 nm. This was sufficiently close to the 514.5 nm green line of the Argon spectrum to ensure that effects due to dispersion were negligible. Movement in and out along this axis was achieved by an indexing collar on the probe holder that allowed 0.25 mm of axial displacement for each complete turn of the adapter body. Rotational position about the spark plug axis was achieved with the use of an etched disk and reference marker attached to the intake runners and aligned to the tumble plane.
The displacement depth of the LDA probe volume was calculated from the dimensions of the probe holder, internal shoulder height for the probe, thickness of the Perspex slug, the fixed focal length and beam separation at the transmitting lens and the refractive indices of the air and medium. Table 5.1. was then drawn up to relate the spark plug adapter position (number of rotations of the adapter body) to the true LDA probe volume intersection in the combustion chamber. The in-cylinder measurement locations were chosen at 5mm incremental depths from the spark plug gap (3mm from the spark plug body) to a maximum depth of 33 mm (approximately 20 mm below the gas face). These locations are shown in Figure 5.7, which also defines the positive tumble and cross-tumble velocity measurement directions. From the displacement table, the calculated error in final probe volume location due to the 0.25 mm pitch of the adapter was less than ±0.25 mm from the expected position. The final installation of the probe and adapter between the two intake runners was as shown in Figure 5.8, where a compressed air supply and thermocouple were used to ensure the assembly did not exceed 45 °C. The intake assembly with damping volume and plenum are also shown.

**Quartz Annulus Optical Access.** The second study was realised by the insertion of a fused silica annulus between the cylinder head and upper piston liner. The extended piston height was modified to compensate for the 20 mm thickness of the annulus and the retaining plate. The annulus internal diameter was the same as that of the engine bore. The assembly of the annulus was as shown in Figure 5.9a and b. The annulus sat upon a 0.25 mm gasket (material GS140) in a free-floating recessed disc within the retaining plate on top of the upper cylinder barrel. A second gasket was fitted between the top of the annulus and the gas face of the cylinder head. A 15 Nm torque was evenly applied to ensure that the gaskets were correctly crushed and provided an adequate pressure seal without inducing any torsional load upon the fragile annulus. Optical measurements were performed through the annulus wall in the vertical plane across the cylinder bore.

### 5.2.1.3. Definition of Measurement Locations

The in-cylinder measurement locations are defined separately for the two optical access scenarios. The first set of spark plug axis measurements are defined by the measurement plane and the distance from spark plug body as shown in Figure 5.7. These are listed as follows:

<table>
<thead>
<tr>
<th>Depth (mm)</th>
<th>Tumble</th>
<th>Cross-tumble</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>07TL</td>
<td>07CT</td>
</tr>
<tr>
<td>8</td>
<td>06TL</td>
<td>06CT</td>
</tr>
<tr>
<td>13</td>
<td>05TL</td>
<td>05CT</td>
</tr>
<tr>
<td>18</td>
<td>04TL</td>
<td>04CT</td>
</tr>
<tr>
<td>23</td>
<td>03TL</td>
<td>03CT</td>
</tr>
<tr>
<td>28</td>
<td>02TL</td>
<td>02CT</td>
</tr>
<tr>
<td>33</td>
<td>01TL</td>
<td>01CT</td>
</tr>
</tbody>
</table>

The engine speed and operating conditions are given in the relevant figures.

For the quartz annulus set of tests, the measurement locations are given in Figure 5.10, for three vertical planes, through the mid-cylinder tumble plane.
The points were selected to cover the intake and piston bowl regions. At each depth below the gas face, the origin \((ROOFOO)\) was selected as the mid-cylinder axis. The positive \(x\)-direction was defined as being towards the exhaust port. As in the previous study, the true positions of the measurement locations were verified by targeting the LASER. The file naming convention was as follows:

- \(R(xx)FOO\) for points in a positive \(x\)-direction from the origin
- \(N(xx)FOO\) for points in a negative \(x\)-direction from the origin

In both cases the value \((xx)\) denotes the distance in (mm) displaced from the mid-cylinder axis. The three \(z\) (depth planes) are defined in (mm) from the gas face \((z=0)\) and were selected as \(z=-9, -10,\) and \(-15\) mm. The choice of these planes was dictated by the LDA focal length and beam separation (distance across the chamber that could be attained) and the physical obstruction to the beams caused by parts of the engine assembly.

### 5.2.1.4. LDA System Set-up

A single component, backscatter LDA system was used for both investigations. A Spectra-Physics Stabilite 2017 Argon-ion LASER was used operating on the single green wavelength of 514.5 mm. This was operated with a beam splitter and Bragg cell with a 40 MHz negative frequency shift. For the spark plug study, the coherent beam pair was transmitted via fibre-optic to a \(\Phi 14\) mm Dantec FibreFlow Probe. The focal length of the transmitting lens was 50 mm and the separation was 8 mm. LDA measurements could be made in any direction perpendicular to the optical axis by rotation of the FibreFlow Probe.

For the annulus study, a 4-times beam expander was fitted to the FibreFlow Probe with a 160 mm lens. The probe volume details are given in Table 3.4. of Chapter 3 for both set-ups. A height-adjustable optical breadboard was manufactured at the University to ensure accurate mounting of the transmitting optics. The stainless steel breadboard was circular in shape and mounted around the engine as shown in Figure 5.11a, b. The centre of the table and the equi-spaced annular tapped threads were referenced to the engine cylinder axis. The table was vibration isolated from the test-cell floor by the use of silicone damping rings placed in each supporting leg.

The transmission optic was then mounted parallel to the optical bench surface and aligned to the breadboard reference holes. This allowed referenced axial translation, radial rotation and height adjustment of the transmission optics. The LDA transmitted beam pair were formed in the vertical plane through the annulus as shown in Figure 5.12. through the mid-cylinder tumble plane. This negated the lensing effect encountered with measurements performed through curved optical boundaries and enabled the true probe volume position to be calculated from simple refraction theory. More complex refraction corrections for the position of the LDA probe volume through curved optical sections are given by the formulae of Bicen (1982). Measurements in this LDA set-up were thus limited to the vertical (in the direction of the piston travel) direction.

The experimental determination (tuning) of the best LDA set-up at each measurement location was as described in Chapters 3 and 4.
For most measurement points, a 45 MHz bandwidth was selected which allowed measurements in the range of \(-32 \text{ ms}^{-1}\) to \(113 \text{ ms}^{-1}\) for the optical spark setup. For the longer focal length annulus studies, this was reduced to between \(-26 \text{ ms}^{-1}\) and \(90.5 \text{ ms}^{-1}\). In each case, the systems were first tested with the engine turned over by hand with seeding blown through the engine at a reduced flow rate as previously described in Chapter 4. The optimum 55X08 PMT voltage was achieved by careful attention to the form of the overloaded signal patterns particularly evident at the 01 (33 mm) and 02 (28 mm) locations that intersected with the piston crown towards TDC. The signal validation threshold on the velocity channel was maintained at \(-3 \text{ dB}\) and the encoder mode was set to count and mark on the once-per-revolution reset pulse. No external burst inhibit signal was supplied. In all sets of results, the number of validated samples (data validation rate) and the number of engine cycles is noted. Once again, the data was processed using the Dantec FLOware software, version 3.3.

5.2.1.5. Engine Preparation and Experimental Method

The general engine installation and instrumentation was as described in Chapter 3, Section 3.5.2. However, in this case, the cylinder head bolts were tightened to 15 Nm for the annulus build only. A modified extended barrel was used which did not incorporate an angled mirror. The crankshaft was replaced to ensure the correct stroke, whilst a new upper liner was procured for the correct bore dimensions. A 'bowl in piston' crown and upper piston were manufactured and the engine built to allow for a 0.3 mm piston stand proud. The engine was then turned over by hand for several cycles to ensure the valve timing was correct. The exhaust valve timing and low clearance heights meant that the piston-valve distance at TDC NF was very small and required precise valve timing to within 0.5 CA. The piston topland to cylinder head clearances (squish volumes) are given in Figure 5.3.

Once the checks were complete, the engine was run up to 500, 1000 and then 1500 rpm ± 5 rpm and WOT for the LDA measurements. Motored WOT maximum pressures were recorded of 21.8 bar absolute at 1000 rpm and 22.8 bar absolute at 1500 rpm. This compared favourably with 25 bar absolute recorded in a normally firing engine under cranking conditions at 1500 rpm and with a standard piston ring set. This indicated that the upper piston ring configuration did not initially suffer from significant piston ring blow-by.

At each in-cylinder location, both along the spark axis and through the annulus, a minimum of 50,000 validated Doppler events were recorded where possible over consecutive engine cycles. The data acquisition and method of ensemble-averaged processing was as described previously

5.2.1.6. Specific Investigative Tests:

(a) Consecutive Test Repeatability. A first series of specific tests were performed to assess the repeatability of the measurement process from one data set to the next under identical engine operating conditions. The spark plug probe depth was fixed at 32 mm below the spark plug face and the engine motored at 1000 rpm and WOT.

The test point was then repeated after the engine had been brought to rest with identical engine, seeding and LDA instrument settings. The complete test procedure was then duplicated at 1500 rpm.
(b) Through Test Cycle-to-Cycle Variability. A second series of tests were performed to assess the cycle-to-cycle variability of the measurements at 1500 rpm and WOT for the 13 mm, tumble 5, spark location. The instantaneous measured velocity was logged against the LDA processor arrival time without an encoder reset pulse. Tumble position 5 was selected as this location consistently provided the highest mean data rate of validated Doppler bursts.

(c) Choice of Doppler Frequency Bandwidth. A third series of tests was used assess the magnitude of error in the ensemble-averaged mean and RMS velocity estimates associated with the incorrect choice of frequency bandwidth. Measurements were recorded in the cross-tumble plane at WOT, 500 rpm and 32 mm depth. The frequency bandwidth was varied between 4 and 12 MHz. This gave velocity ranges of between ± 6.45 ms^{-1} for the former and ± 19.3 ms^{-1} for the latter bandwidth selection.

(d) Effect of Engine Speed. In this series of tests measurements were performed at engine speeds of 500, 1000 and 1500 rpm with a 200 mbar intake depression. Each test was conducted at a depth of 32 mm in the tumble plane. The tests were then duplicated in a second independent series of engine runs.

(e) Effect of Intake Throttling. As in the previous tests, two independent data sets were measured at 1500 rpm for two different intake manifold pressure conditions at the 32 mm location and in the tumble plane. The engine was run at WOT and with a 200 mbar depression. The ensemble-averaged velocity data for the two conditions and two tests were logged against encoder crank angle position.

(f) Particle Residence Weighting. The effects of velocity bias in the ensemble-averaged estimate of the mean velocity was investigated for the measurement point 5 in the tumble direction at 1500 rpm and WOT. Two regions in the engine cycle were specifically chosen to assess the effects of velocity bias upon the ensemble average estimate. More precise details of the method are given in the results and discussion section below.

5.3. Experimental Results and Discussion

5.3.1. Temporal Characteristics of Instantaneous Velocity, Ensemble-Averaged Mean and RMS Velocity across the Engine Cycle.

5.3.1.1. Spark Plug Measurements — Radial Velocity Component

(e) Ensemble Averaged Data
The instantaneous and ensemble-averaged mean and RMS velocity results for the spark plug LDA measurements at the seven points co-linear with the spark plug axis are given in Appendix F for the tumble plane at 1500 rpm and Appendix G for the cross-tumble plane. In the first instance, it should be noted that both the tumble and cross-tumble locations 1 and 2 (33 and 28 mm depth respectively) exhibit two dead regions where the piston passes through the probe volume location at the deepest point in the piston bowl at TDC. The effect is most pronounced at position 1.

The tumble plane measurements are presented in Appendix F. The tumble plane was the more instructive of the two planes investigated.
This was to be expected in a high tumble combustion chamber. The recorded velocities at each of the measurement locations are best understood by observing their positions relative to the intake valve, during the intake stroke and piston bowl periphery, during the compression stroke. The early intake phase of the cycle is dominated at all locations by a rapid inflow of gas that reaches an instantaneous maximum velocity in excess of 60 m/s at 60-80 CA at tumble location 5, at 13 mm from the spark plug gap and approximately at the gas face level. The ensemble-averaged mean velocity peaks at 40 m/s. The highest velocity gradient is observed in the region of 20 to 30 CA at low valve lift. For locations below 13 mm, the magnitudes of the ensemble-averaged mean velocity increase from approximately 15 m/s at tumble point 1 to 30 m/s at tumble point 4. Above 13 mm, the peak decreases slightly to 30 m/s at point 6 and 32 m/s at point 7.

The instantaneous velocity distributions at these points also exhibit a more defined pattern that is reflected in the reduced magnitude of the ensemble-averaged RMS velocity estimate at these points. Both tumble 6 and 7 are located in the jet flow region that exits the inlet port over the top of the valve and is bounded by the pent-roof chamber. They also start the cycle with a negative velocity component up to the point of exhaust valve closure. This is due to gas inflow drawn from the exhaust port by the downward motion of the piston in the valve overlap period. Tumble 5 is on the edge of the main jet flow and closer to the intake valve. This point experiences both a positive velocity component and a component due to re-circulation and vortex shedding behind the valve lip. This is shown by the larger spread in instantaneous velocity measurements. The effect becomes more pronounced through points 4 to 1 at 33 mm depth (approximately 20 mm below the gas face and 5 mm from the mid-cylinder axis). At tumble point 1, the instantaneous velocity plot shows a degree of symmetry about the zero velocity axis similar to that observed in the cross-tumble measurements. As a result, this location recorded the highest levels of RMS velocity during the intake stroke. At these points, the measured velocity comprises of a component introduced by the largest jet flow that exits from below the valve, bounded by the cylinder wall. In this particular engine geometry, the inlet valve diameter is relatively small in comparison with that of the Volvo manifold injection engine presented in Chapter 4. The available flow area between the inlet valve and cylinder wall is greater and the jet flow is less restricted. The main jet flow follows the spherical piston bowl perimeter, filling all the available volume. Its motion generates the reverse tumble vortex.

From inlet valve peak lift to inlet valve closure, the ensemble-averaged mean and RMS velocities decreases almost linearly with crank angle at all the measurement points. At BDC, tumble points 1 and 2 reverse velocity direction. These points are the first to record the negative velocities directed towards the intake side of the cylinder. This is the top of the reverse tumbling vortex that has reached the dimensions of the cylinder stroke. Tumble 3 is the next point to record a negative radial velocity component at approximately 190 CA. Tumble 4 reverses at 200 CA, tumble 5 at 205 CA, tumble 6 at 205 CA and tumble 7 at 210 CA. As the compression stroke proceeds, the magnitude of the negative velocity increases and is then maintained for tumble 1, 2, and 3. At this point, the magnitudes of all the mean and RMS velocities are comparable.

At points 4, 5, 6 and 7, the magnitude increases with decreasing cylinder aspect ratio after mid-compression stroke. Between 270 and 360 CA a second measured velocity reversal is observed.
At tumble 1, the ensemble-averaged mean velocity radial velocity component changes direction at approximately 300 CA. The magnitude remains approximately similar. Tumble 2 is the next to change at 310 CA followed by tumble 3 at 325 CA and tumble 4 at 345 CA. Tumble 5 does not reverse direction until several crank angles after TDC firing. The reversal is momentary and of small magnitude, and the flow returns to the negative direction. Tumble 6 and 7 however do not change direction at all and remain negative throughout the power stroke. At tumble 4, the peak negative mean velocity magnitude is 5 ms\(^{-1}\) at 325 CA. This increased to 6 ms\(^{-1}\) (330 CA) at tumble 5 and almost 10 ms\(^{-1}\) (350 CA) at tumble 6. Tumble 7, located in the apex of the pent-roof recorded a peak mean velocity of 7.5 ms\(^{-1}\), 10 CA before TDC.

These changes in radial velocity direction during the latter stages of the compression stroke show that the tumbling vortex generated during the intake is conserved during compression. As the proximity of the piston to the cylinder head decreases, the tumbling vortex has less available volume within which to spin. Considering a frictionless system, a reduction in tumbling radius would be accompanied by an increase in angular velocity. This is often referred to as tumble 'spin-up'. In a conventional geometry, pent-roof gasoline engine, the tumble motion during late compression is eventually destroyed by distortion of the main eddy by the changing geometry and increase in friction. This can be identified by a crank angle at which the bulk flow breaks down to turbulence, recorded as an increase in measured turbulence intensity (e.g. Arcoumanis et al., (1990)). In the present study however, there would appear to be no discernible increase in turbulence intensity at the spark plug locations during the latter stages of the compression stroke, especially in the tumble plane. The turbulence intensities for tumble locations 3, 4, 5, 6 and 7 are presented in Figure 5.13. for the period of 40 CA BTDC. Locations 1 and 2 are not included as they intersect the piston bowl in this range. The plot would seem to indicate that there was little decay of the prominent reverse tumbling motion into small scale turbulent eddies. The turbulence intensity at TDC is within the range of 25 to 50%. Tumble 5 however, shows a sharp increase in turbulence intensity from 340 CA to TDC. Over this range, tumble 5 at 13 mm depth from the spark plug gap is approximately at the geometrical centre of the piston bowl. If the tumble vortex is assumed to completely fill the available area, then the recorded flow at tumble 5 will fluctuate intensely between a negative and positive measure. Indeed, from the instantaneous and ensemble-averaged results, it can be seen that tumble five showed an approximately zero mean radial velocity component at TDC whilst points 3, 4, 6 and 7 were always greater than \(\pm 2\) ms\(^{-1}\). It is also of interest to note that none of the measurement locations would appear to be effected by a squish flow from the exhaust to intake side of the chamber. All the tumble point locations, excepting tumble 5, contrarily indicate a reduction in RMS velocity prior to TDC. At locations 3 and 4, well within the bowl area, the RMS velocity decreases from between 320 CA to 340 CA to 2-3 ms\(^{-1}\) at TDC and a mean velocity of 4-5 ms\(^{-1}\). A decrease is also evident from 345 CA at locations 6 and 7. Both points show RMS velocities of 2-3 ms\(^{-1}\) at TDC and mean velocities as high as 7 ms\(^{-1}\). At tumble 5, an increase to 3 ms\(^{-1}\) in RMS velocity is observed over 350-360 CA.

In the cross-tumble plane, all the measurement points show a similar characteristic pattern. Relatively high velocities that fluctuate almost symmetrically about the zero velocity value dominate the intake stroke. A local velocity peak was observed in the region where the valve lift and piston speed begin to increase significantly.
The lowest peak value of 20 ms\(^{-1}\) was observed at approximately 70 CA and at position 1 at a depth of 33 mm. The peak value increased with decreasing distance from the spark plug until a maximum value of 50 ms\(^{-1}\) was recorded at location 5. At points 6 and 7, the peak values dropped to approximately 40 ms\(^{-1}\). The resultant ensemble-averaged mean and RMS velocity plots showed the magnitude and degree of symmetry in the fluctuating velocity component. The RMS velocity fluctuation rises to a sharp peak at all locations at approximately 70 CA. Again, the magnitude of the peak increases inversely with depth from the spark plug gap up to point 5. Points 5, 6 and 7 show maximum RMS velocities of 20 ms\(^{-1}\) in the same region. These locations are closest to the valve seat jet flows and regions of flow re-circulation behind the valve curtain. In addition, measurements in the cross-tumble plane are more effected by the four-valve geometry of the cylinder head. Jet flow interaction between both intake valves will produce a significant velocity component in the cross-tumble direction as each inflow competes to fill the cylinder volume. The flow instabilities induced in the intake system lead to intermittent jet mixing and flapping that is observed as an apparent increase in the measured RMS turbulence intensity (skewed velocity probability distributions) and has been reported in other steady flow studies, e.g. Nadarajah et al. (1999).

From approximately 70-80 CA, all the measurement points show a constant decay in instantaneous velocity magnitude, over 100 CA, up to the point of inlet valve closure. From this point, the RMS velocities show no discernible increase and are nearly constant in the range 2.5-4 ms\(^{-1}\) until TDC compression. Points 3 and 7 shows a small decrease in RMS velocity from 350 to 360 CA. This would suggest that the main tumbling vortex had remained coherent and had not undergone a breakdown into smaller eddies. The small cross-tumble RMS velocity recorded is that due to friction against the chamber walls and the shifting of the axis of rotation of the main tumble vortex across the cross-tumble plane. There is no appreciable mean velocity until just before TDC. A small positive velocity of between 0.5 and 1 ms\(^{-1}\) was recorded at points 3 and 4. Points 5, 6 and 7 indicate slightly larger negative velocities of between 2 and 3 ms\(^{-1}\).

Figures 5.14a. to d. show the normalised ensemble-averaged mean velocities in the tumble plane for all the measurement points against piston displacement for the four strokes. During the intake stroke, a typical fuel injection timing is displayed for a homogenous charge, full-load engine condition. With even a modest injector cone angle, fuel would be sprayed through the tumble points 1, 2, 3, 4 and 5. The magnitude of the mean and RMS velocities would ensure good mixing of the fuel with the intake charge.

The plot of the compression stroke better represents the features of the bulk flow motion described above. Included also are a series of injection timing for a late, stratified charge strategy. A well-defined rotational bulk motion would be required to effectively stratify the injected fuel load and transfer it to the spark plug region. At all measurement points up to approximately 25 mm piston displacement from TDC, the mean velocity is constant and directed towards the intake side of the chamber. Tumble position 1 is the first location to exhibit a change in velocity direction when the clearance height is approximately 23 mm.

This is followed by tumble 2 at 17 mm, tumble 3 at 8 mm and finally tumble 4, at 3 mm from TDC piston position. Tumble 5, 6 and 7 do not change direction but increase significantly in magnitude towards the intake side at TDC.
The form of the plot can be explained by the translation of a coherent, reverse tumbling body, which is driven up the cylinder by the piston motion, conserving angular momentum and which interacts initially with the measurement volume at the lowest point in the bowl (tumble 1). This point records the radial velocity component of the top of the vortex. The continuing upward piston motion pushes the bulk tumbling vortex through the remaining co-linear (spatial fixed) measurement points. As the decelerating piston moves upwards, the lowest points in the bowl encounter the bottom of the tumbling vortex. The radial velocity component measured at this point is opposite in sign to that at the top of the vortex. As the piston approaches TDC, the four deepest measurement points consecutively measure a radial component of the angular velocity that is positive towards the exhaust side of the chamber. Points 5, 6 and 7 confined within the pent-roof continue to measure the top of the vortex. The results confirm that a tumbling body is present in the compression stroke up to and beyond TDC. Figure 5.16c. shows that the sign and magnitude of the velocities in the early phases of the power stroke indicate a reverse tumble motion that slowly decays until exhaust valve opening. These phenomena are not generally observed in 4 valve pent-roof combustion geometries with 'regular' intake geometry, where the bulk tumbling motion is seen to dissipate into smaller scale eddies after the break-up of the predominant vortex.

5.3.1.1.1. Specific Investigative Test Results – Radial Velocity

(a) Consecutive Test Repeatability. The repeatability in the ensemble-averaged mean and RMS velocities from one independent test to the next are shown in Figures 5.15a., b at 1000 rpm and Figures 5.16a., b at 1500 rpm. In both sets and at both engine speeds, approximately 10,000 validated Doppler events were captured. The results are used not only to assess the measurement technique but also the repeatability of the engine test conditions. For both engine speeds, the ensemble-averaged mean and RMS estimates follow the same characteristic pattern throughout the four engine strokes. However, the oscillations in both curves are not as well reproduced from one data set to the next. This is because the ensemble-averaging procedure requires a large amount of data per crank angle window with which to make a statistically accurate mean estimate over that interval. The seeding rate throughout the cycle and with location, from cycle to cycle and from test to test is not constant and mean estimates are made from varying sample sizes. It can be seen from the figures that the magnitude of the difference in the ensemble-averaged mean velocity estimate is greater at the higher engine speed. Figures 5.15b. and 5.16b. show the difference in estimates between the sets over the intake and compression and then through the power and exhaust strokes. The difference bar chart highlights the need for large data sets for mean estimates. The largest differences are observed in the turbulent phase of the intake stroke at 1500 rpm that shows particularly poor repeatability. The difference in this region exceeds 50 % of the measured value at some crank angles. The difference at 1000 rpm is much lower in magnitude through early intake and the compression stroke, being of the order of ± 1 ms\(^{-1}\) with some exceptions.

It should be noted also, that in addition to insufficient data points per averaging window, a very slight phase shift in crank angle encoder signal would result in large difference fluctuations.
This would have more of a potential overlapping effect at 1500 rpm, where one averaging window is equivalent to only 80 \( \mu \text{s} \).

(b) **Through Test Cycle-to-Cycle Variability.** A qualitative comparison of 2 and 6 individual engine cycles is presented in Figure 5.17, for the tumble point 5 at 1500 rpm. The validated Doppler burst arrival time from the start of the experimental run was used to extract the instantaneous velocities from each consecutive engine cycle. At 1500 rpm, one complete engine cycle lasts for 40 ms. The measured engine cycle time on the plot is however 80 ms due to the two revolutions of the four stroke cycle registered by the Doppler processor prior to post-processing. The results show the irregular arrival times of the validated velocity measures throughout a single cycle and from cycle to cycle. At only a few points in the cycle, where seeding concentration is high, can the sampling rate be considered as nearly continuous. The total number of collected validated signals for this engine run was in excess of 50,000. However, these results suggest that there are too few measurements for an in-cycle (or cycle-to-cycle) resolved analysis.

(c) **Choice of Doppler Frequency Bandwidth.** Two separate test cases are presented for a spark plug axis measurement at a depth of 32mm. Velocities were recorded in the cross-tumble plane at 500 rpm. The first set of tests was conducted with a 4MHz bandwidth, which in this optical configuration, gave a velocity range of \( \pm 6.45 \text{ ms}^{-1} \). The second set were performed with a 12 MHz bandwidth corresponding to a velocity range of \( \pm 19.3 \text{ ms}^{-1} \). The results of instantaneous, mean and RMS velocities are presented in Figure 5.18. In the first plot, the choice of a smaller bandwidth restricts the measurements to a narrower velocity window and 'clipping' of the total velocity distribution is evident. This results in an apparent reduction in RMS velocity at those parts of the cycle most affected; i.e. the intake and early power stroke phases of this plot. The mean velocity plot shows that the smaller bandwidth both over and under predicts the ensemble-average mean velocity of the total measures even in flow regions within the velocity bandwidth bounds. For these points, a reduction in velocity bandwidth reduces the burst detector bandwidth and improves the validated signal data rate (or the system ‘sensitivity’). Increasing the measured velocity range, reduces the system ‘sensitivity’ to small velocity fluctuations and is more likely to introduce errors due to velocity bias effects.

(d) **Effect of Engine Speed.** Two sets of independent tests were performed to study the effect of engine speed upon the mean and RMS turbulence velocities. The engine was motored at 500, 1000 and 1500 rpm and data recorded at a depth of 32mm from the spark plug body. Figures 5.19a and b show the effect of engine speed over the entire engine cycle with the intake throttled to 800 mbar absolute pressure. Data is presented for ensemble-averaged mean and RMS velocities recorded in each set. An initial inspection of the results for this single in-cylinder location, suggests that the ensemble-averaged estimates scale linearly with engine speed. The same characteristic patterns and even some of smaller scale fluctuations are present at all speeds through the intake stroke. However, during the compression stroke, the mean velocity in both data sets begins to behave differently at the 1500 rpm condition. To highlight the differences, the plots were normalised by mean piston speed and are shown in Figures 5.20a. and b. for the mean and RMS values respectively.

To assess the linearity across the engine cycle, the mean and RMS velocities are also plotted against engine speed for fixed crank angles of 90, 180, 270, 450, 540, 630.
A linear curve fit is used as an approximation to the data points. During the turbulent intake (90 to 180 CA) and power stroke (450 CA) phases of the cycle, the normalised mean velocity does not scale with engine speed. Instead, its value decreases with increasing engine speed at a rate of between 1.7 to 1.8 ms\(^{-1}\) per 1000 rpm. However, at 270, 540 and 630 CA, the normalised mean velocity is constant over the three engine speeds. In contrast, the best linearity of normalised RMS velocity with engine speed is observed 90 and 540 CA. The poorest linear approximation to the measured points occurs at 180 and 450 CA. At 270 CA in the compression stroke and 630 CA in the exhaust stroke, the normalised RMS velocity increase by 0.5 and 1 ms\(^{-1}\) over the speed range respectively.

(e) Effect of Intake Throttling. Figures 5.21a and b present the ensemble averaged mean velocity and RMS turbulence velocity for two intake manifold pressure conditions under repeat test conditions. The first set of measurement were performed at WOT and 1 bar atmospheric pressure conditions, whereas the second set were recorded with a 200 mbar manifold depression. In each case the engine was motored at 1500 rpm. In each case the mean and RMS velocities are both altered in magnitude. In addition, the mean velocity estimate shows a change in direction at two points in the cycle. Throttling the intake reduces the mass flow rate of the airflow into the engine and as such, the magnitude of the mean velocity is reduced during the intake phase. At BDC, the characteristic velocity evolution initiated by the tumbling vortex is observed at this measurement location for WOT conditions. In contrast, throttling of the intake flow has the result of changing the mean flow radial component direction but not its magnitude, in both data sets. This sharp change is associated with a peak in RMS velocity at approximately 180 CA. From TDC firing to the end of the cycle, the velocity remains almost continuously opposed in direction to that measured at the WOT conditions. A second peak is observed in the RMS velocity distribution shortly after TDC firing. The results suggest that throttling of the intake has significantly altered the magnitude and direction of the measured radial component of velocity at this in-cylinder location. In doing so, it has also induced high-levels of velocity fluctuation. The position of the maximum intake peak mean velocity between the two conditions is of particular interest. Under WOT conditions the intake peak spreads across approximately 100 CA. Under throttled operation, a sharp peak is observed initially at 60 CA followed by significant drop in mean velocity (in excess of 50%) and then a secondary peak at around 160 CA. For a measurement point in this location (cf. tumble 01), the main jet flow from beneath the valve is predominant in this phase. With a reduction in mass flow rate, the velocity magnitude of both intake jets is reduced and at peak valve lift, the flow from above the valve attains a comparable magnitude and the averaging reduces the mean value. As the piston moves downward, the lower jet once again predominates until intake valve closure. The mean flow motion, observed during the compression stroke, is in the forward tumble direction at this location.

(f) Particle Residence Weighting. The effects of velocity bias in the ensemble-averaged estimate of the mean radial velocity are shown in Figure 5.22 for the measurement point 5 in the tumble direction. In this test, the mean data rate was 0.8 kHz and 52,000 validated data points were recorded. The results are presented for two regions in the engine cycle that were specifically chosen to assess the effects of velocity bias upon the ensemble-average estimate.

The regions were chosen in the intake stroke where particle seeding of the engine cycle takes place following the exhaust process. The total particle seeding volume is not complete until
the intake valve closes. The first region is from TDC 0CA to 20CA during the intake process during the valve overlap period and when the intake valve lift is low and the RMS velocity fluctuation is high. The scatter plot indicates that the number of relatively low velocity measurements in this region is less than that for the rest of the intake stroke. That is, the 'local' data rate in this region was likely to be less than the recorded mean data rate taken over all engine strokes and cycles.

The second region is further into the intake stroke, from 40CA to 50CA. In this region, the intake valve has a greater lift and a high-speed jet flow enters the cylinder. The RMS velocity fluctuation is at a minimum and the velocity scatter plot indicates large amounts of validated data. Again, the 'local' data rate is likely to have exceeded the mean data rate of all cycles.

The seeding particle statistics are used to weight each velocity measure with the inverse of the instantaneous volume flow rate in the probe volume. The distribution of the particles is assumed homogeneous. The instantaneous volume flow rate through the probe volume is proportional to the particle transit (residence) time, \( \Delta t \). The transit time weighted mean velocity, \( u_N \), is then calculated from the following expression:

\[
\frac{\sum u(t_i) \Delta t_i}{\sum \Delta t_i}
\]

The lower plots in Figure 5.22 show the ensemble-averaged and particle residence time weighted average velocities for both regions of interest. The vertical velocity scales in each plot are of the same 12 ms\(^{-1}\) dimensions. In the 0-20CA range, the ensemble-averaged estimation under predicts and over predicts the weighted curve. The region is highly turbulent and particle seeding is sparse with approximately 20-30 samples recorded for each crank angle window. In the 40-50CA region, the ensemble-average prediction is always greater than that calculated by the weighted average. The samples are much greater in number; increasing from 200 to 500 samples per 1.44CA window over the 10 CA range. The ensemble-averaged value will therefore tend to a closer mean value estimate. The difference in total absolute velocity magnitude between the two sets of mean data is at most approximately \( \pm 1 \) ms\(^{-1}\). The graphs clearly show that for mean plots of early turbulent intake processes with sparse seeding and relatively low velocity, the ensemble average is a poorer indication of the true flow velocity than in other parts of the cycle. However, even in such cases, the ensemble-average estimate is within the uncertainty of the experiment and provides an adequate estimation for standard analyses.

5.3.1.2. Quartz Annulus Measurements – Axial Velocity Component

The general bulk flow patterns have already been described in the previous section for the radial velocity component measured along the spark plug axis.
Therefore, this section only describes those features pertinent to this data set during the intake and compression strokes. Large velocities and steep velocity gradients once again dominate the early intake phase. The largest velocities are found at 20 and 25 mm from the cylinder axis towards the intake side. The magnitude of the mean velocity peak diminishes to half the value at mid-cylinder and 5 mm from mid-cylinder towards the exhaust side. These positions are removed from the main jet flows. Also, moving towards the intake side cylinder wall, the ensemble-averaged mean velocity is reduced by 10 to 15 ms\(^{-1}\) in this early phase where the piston topland is in close proximity to the cylinder head gas face. This highlights the influence of the curved intake port geometry and the sharp angled valve seat that deflects the main flow away from the wall and into the piston bowl. The RMS velocity distribution at all points during the intake stroke shows the same characteristic pattern and peak with similar magnitude.

During the compression stroke, measurement locations from mid-cylinder to 20 mm from the cylinder axis show negative mean velocity values up to the point of piston interaction. At mid-cylinder a mean velocity of \(-5\) ms\(^{-1}\) (i.e. in the upwards direction) is maintained until three-quarter stroke. Moving across the data points towards the intake side shows a decrease in the magnitude of the negative component until the mean velocity component tends to zero at N25F00, 25 mm from the cylinder axis. At N30F00 and N35F00, the mean velocity is zero during the entire stroke. The measured vertical component is that due to the reverse tumble motion moving up the stroke. The points at the extremities of the cylinder are outside of the tumble flow and may be influenced by the roll-up vortex created in the step between the piston edge and the cylinder wall. The mean and RMS velocities recorded at N25F00 and N30F00 indicate high levels of turbulence intensity in this region. The RMS turbulence velocities at all the measurement points during the compression stroke are relatively small in magnitude suggesting a well-defined bulk fluid motion.

5.3.2. Spatial Distribution of Air Flow Ensemble-Averaged Mean Velocity.

5.3.2.1. Radial and Axial Mean Velocity Components

The ensemble-averaged mean velocity for the measured radial component normal to the spark plug axis in the tumble plane is plotted for each measurement location at a constant crank angle. Figures 5.23a. to d. show the velocity vector component in each of the four-strokes. The scale vector is set at 10 ms\(^{-1}\) in all the plots. During the early intake, a strong jet flow is observed through the 3, 4, 5 and 6 locations and the highest radial velocities of the whole cycle are recorded. A strong mean flow at these locations is still observed at 172 CA in the forward tumble direction. During the compression stroke, the magnitude of the mean velocity radial component is seen to increase and the reverse tumble flow pattern becomes established. Early in the power stroke, a reverse tumbling motion persists, albeit small in magnitude, until approximately 460 CA. A sink flow is then established from the point at which the exhaust valve starts to open and the magnitude of the mean velocity component increases to a maximum at peak valve lift.

In Figures 5.24a to c, the axial velocity component is presented in the same vector format for the quartz annulus measurement locations. The engine and annulus geometries are superimposed on the plot. In addition, the piston bowl location and piston topland height is marked at each crank angle.
These are illustrated in the first plot at 20 CA along with the spark plug axis that defined the measurement locations of the radial velocity component. The plot for 60 CA shows the magnitude of the mean radial component. In all the plots, large-scale arrows are used to describe the mean flow pattern through the intake and compression strokes.

At the beginning of the intake stroke, the predominant jet flow exits from beneath the valve and is measured at approximately 40 CA. Between 60 and 80 CA angles, both jet flows in the tumble plane are captured as the piston bowl region passes through the horizontal measurement line. The mean flow axial component then continues to follow the downward motion of the piston. The magnitude is greatest towards the intake side cylinder wall. At BDC, the magnitude reduces to a minimum and the velocity component fluctuates about zero until 220CA. At this point, the mean velocity near the cylinder axis begins to increase to approximately 5 ms⁻¹ at 280 CA. As the piston continues to approach TDC, the mean velocity is seen to diminish but the tumble motion is still present at 300 CA. However, the location 25, 30 and 35 mm from the cylinder axis towards the intake side recorded negative velocities at 300 CA due to the piston motion and wall interaction.

The vertical plane of the LASER beams and the height of the piston topland limited the last possible measures to 320 CA. Even so, the large negative velocity recorded at 25 mm from the cylinder axis is thought that due to PMT overload, also present in the first plot at 20 CA. This was a particular feature of this measurement location in the proximity of the piston with a high reflection signal from within the bowl. At 340 CA and 360 CA, the sporadic mean velocity measures are also due to PMT effects and can be easily eliminated.

Experimental measurements using the annulus method were limited to approximately 40 CA BTDC in the compression stroke. The spark plug measurement however permitted measurements up to TDC. To study the characteristics of the tumbling flow within the bowl, the radial velocity components were selected at 1.44 CA increments from 320 to 360 CA. These are presented in Figures 5.25a. to d. and clearly show a reverse tumble motion, whose centre translates up the cylinder stroke. As the piston approaches TDC the magnitude of the radial component increases.

5.3.2.2. Evaluation of a Tipping Tumble Vortex Ratio

A tumble ratio can be defined as the ratio of some characteristic parameter of a rotating body to that of a solid body rotation. Arcoumanis et al., (1990), defined the swirl ratio of the in-cylinder flow as

\[ \frac{1}{\omega_c R_c} \int_0^R \frac{\bar{W}}{r} dr \]

where \( \omega_c \) is the engine speed in rads⁻¹, \( R_c \) is the cylinder radius and \( \bar{W} \) is the ensemble-averaged swirl velocity. They then defined the non-dimensional angular momentum per unit mass around the center of rotation (zero velocity location). The ratio of swirl ratio to angular momentum of a solid body was then used to define a new swirl ratio

\[ \omega' = \frac{2}{n R_c} \frac{1}{R^3} \int_0^R r^2 \bar{W} dr \]

where \( n \) is the engine speed in rev/s.
The final swirl ratio was summed over the number of axial velocity profiles measured.

An equivalent ratio can be used to describe the tumble motion in the late stages of compression based on several assumptions. These are referred to in Figure 5.26. At 320 CA, the piston topland is approximately 10 mm from the cylinder head gas face and all the measurement points are located within a circle drawn about the geometric centre of the piston bowl and of equal radius. The tumbling flow is then assumed to fill the whole available area as indicated by the dashed line ellipse. As the measurement locations are spatially fixed, it must be assumed that the tumble centre, or point of zero velocity lies along the axis and at the point that the velocity profile changes direction. This method was also preferred in the studies of Arcoumanis et al., (1990) and Baby and Floch (1997). In this study, the choice of spark plug axis for tumble estimation late in the compression was made based upon the tumble centre LDA investigation undertaken by Jaffri et al., (1997) using an AVL single cylinder research engine fitted with a single cylinder of a V-6, 24-valve gasoline engine, with both tumble and swirl flows. They compared tumble ratios in the vertical plane for a fixed tumble centre, located at TDC and a moving tumble origin, located at the instantaneous centre of the cylinder volume. In general, the moving origin approach showed far less fluctuation in tumble ratio compared to the fixed origin. In this study a moving origin approach was adopted with its motion along the spark axis. This was felt to be a better choice than that of the more traditional mid-cylinder axis, due to the offset nature of the tumble vortex induced by the chamber geometry. At each crank angle, the ensemble-averaged mean velocity is plotted against the distance from the spark plug gap as shown in Figure 5.27. The point at which the velocity sign changes is the distance along the axis to the approximate point of zero velocity or centre of tumble rotation. Interpolation between the points to smooth the curves did very little to alter the point of cross-over and the original data points were considered adequate.

The swirl ratio referred to by Arcoumanis et al., (1990) was then applied in the tumble plane along the spark plug axis using the horizontal component of the inclined (8 degrees to the vertical) radial measured velocity component. The integral over the total measured depth was divided into two regions about the tumble centre point; a region of negative velocity magnitude and a region of positive magnitude. This is shown in Figure 5.26., where R1 and R2 denote the distances over which the velocities moments are negative or positive respectively. The revised inclined or 'tipping' tumble vortex ratio (using the notation of Arcoumanis et al., (1990) is given by:

$$w = \frac{2}{mR} \left[ \frac{1}{R1} \int_0^{R1} r^2 V_{z,0} dr + \frac{1}{R2} \int_0^{R2} r^2 V_{z,0} dr + \right]$$

where $V_{z,0}$ is the ensemble-averaged mean radial velocity component at a distance, $z$ along the spark plug axis. Positive tumble rotation is taken towards the intake side.

The 'Tipping' Tumble Vortex Ratio (TTVR) is plotted in Figure 5.28a. against both crank angle and piston displacement. Note the x-axis directions are opposed to aid clarity in the plot. A solid body rotation would give a TTVR of ±1. From 320 to 330 CA (approximately 6 mm from TDC position), TTVR decreases in positive magnitude. This indicates that the centre of the tumble vortex is lower down the spark plug axis and the moments are weighted in favour of the values above the zero velocity crossing.
In this case TTVR is not a useful definition of tumbling rotation with such a limited number of measurement points. From 330 to 345 CA, the tumble ratio increases again in magnitude in the negative direction until a maximum condition is reached at 350 CA (less than 1 mm from TDC). At this crank angle, the tumble rotation approaches that of a solid body. Five measurements were taken from 350 to 360 CA and show that the tumble ratio decreases minimally in magnitude over this range. Between 350 and 360 CA the change in TTVR is less than 0.16 ratios. The final tumble ratio is 0.83. This shows an absence of tumble breakdown into turbulent eddies commonly observed in pent-roof, tumble flow, gasoline engines from 330 CA where the tumble vortex ratio tends sharply to zero.

The plot also includes the cylinder height to bore aspect ratio over the same range of crank angles. Arcoumanis et al., (1990) identified that tumble vortex destruction started at an aspect ratio of 2.5 in a motored gasoline engine using a directed, tumble generating, inclined intake port. This is equivalent to approximately 335 CA. At an aspect ratio of 6, they report that the large structures had broken down into a quiescent flow that exhibited high turbulence levels. In Figure 5.13, the turbulence intensities at the spark plug locations were plotted for the late compression stroke over the same crank angle interval as the TTVR. Tumble points 3 and 4 showed turbulence intensities that were twice that of the other spark locations at 330 CA. From 350 to 360 CA only tumble 5 shows a marked increase in turbulence. Its location between these crank angles (aspect ratio of 5.5-6) is very close to the estimated tumble centre and as such a highly fluctuating velocity component is to be expected. It is not an indication of bulk flow disintegration. In addition, it is likely that the tumble centre moves radially as well as axially and that its precession will induce an apparent low frequency turbulent component to the results as observed by Baby and Floch (1997). In Figure 5.28b., the distance from the tumble vortex centre to the instantaneous geometrical bowl centre is plotted from 320 to 360 CA. This shows the precession of the vortex centre as the piston moves up to TDC, relative to the set of spark axis points. A linear fit is a good approximation to the data points. The rate of change in displacement is of the order of 1.43 times axial to radial. The lowest points occur at 320 CA and then 325 CA. The rate of vortex centre travel relative to the piston position can be estimated between these two points. At 1500 rpm, 5 CA is equivalent to 0.55 ms. Therefore the speed of the centre of the vortex relative to the centre of the piston bowl is approximately 0.64 ms$^{-1}$ between these two angles.

5.3.2.3. Comparison of Late Compression, Tumble Plane, RMS Velocity Profiles between a Manifold Infection and Direct Injection Engine.

An estimation of the crank angle of vortex breakdown can be derived by comparison of the tumble-plane RMS velocity fluctuation observed at the spark plug locations in the Volvo manifold engine with those measured in the direct injection build. The Volvo B230 cylinder has two side-entry, forward tumble, intake ports. The chamber has a pent-roof geometry to aid tumble formation and the piston has a flat top. The RCE direct injection cylinder head utilises a top-entry, reverse tumble air motion principal. The chamber geometry is also pent-roof and the piston incorporates a bowl to conserve charge angular momentum and direct injected fuel.
Towards the end of the compression stroke, conventional combustion systems rely on the destruction of the bulk flows to generate turbulence for the early combustion phase. Figure 5.29. shows the measured RMS radial velocity components over the final 40 CA of the compression stroke. Both sets of measurements have been carried in close proximity to the spark plug location and in the region of early flame development. Trend lines have been added to the plots to approximately indicate the gradient of the curves over the last 15 CA. A typical spark ignition timing might be in the range of 325 to 335 CA. In the Volvo plot, each measurement location shows an increase in RMS velocity from 330 CA excepting SP4 which remains constant and then rapidly increases at 350 CA. SP2, SP4 are located closest to the spark plug and SW3 is located in the corner of the pent-roof. These three points show the greatest increase in fluctuation velocity from 320 CA. SW11, located in the apex of the pent-roof shows the smallest increase. In contrast, the RMS fluctuation observed in the direct injection engine along the spark plug axis is constant up to 350 CA. From this point to TDC, only location 5 shows an increase in RMS velocity as discussed above. The remaining points decrease in magnitude. The type of breakdown mechanism observed in the Volvo engine from 330 CA (and in many other studies in the literature) is not present and the tumble motion is seen to be conserved well into the early phases of combustion.

5.3.2.4. Indirect Integral Length Scale Estimation:

Single Position LDA

In Chapter 2, a distinction is made between the two methods most often utilised to estimate the scale for LDA data: the direct and indirect methods. The direct method requires measurements to be collected at two separate points in the flow field simultaneously and was therefore beyond the scope of the current study. The indirect estimation of the Integral Length Scale can be derived from the Integral Time Scale using a single point method. However, several assumptions must be made about the flowfield. The first is that a strong mean flow must exist that can be considered as quasi-stationary over some time interval. Secondly, the RMS fluctuations in the mean flow must be small compared to the magnitude of the mean flow (typically less than 30%). Thirdly, the turbulent flowfield is considered isotropic and homogeneous. For the G-DI reverse tumbling motion, the TTVR indicates that the tumble motion from 350 to 360 CA shows little decay and may be considered as quasi-steady. In addition, the radial component of the ensemble-averaged mean velocity at spark plug locations is between 5 and 10 times greater than the RMS fluctuation between 350 and 360 CA. The RMS fluctuations in the cross-tumble plane are of the same magnitude.

Two methods for the indirect length scale estimation were presented in Chapter 2. One method involved an individual-cycle and ensemble autocorrelation approach. This was not considered suitable following the examination of the data arrival times outlined in the specific investigations section previously discussed. The other method involved an ensemble-averaged approach considered by Corcione and Valentino (1990, 1991), Hilton (1991) and Hong and Chen (1997) where a phase angle (lag range) was added to the crank angle to account for the non-stationary nature of the flow. The autocorrelation function is then integrated over a small phase range until velocity fluctuations are no longer correlated. They state that the maximum value of the phase angle range is most difficult to predict when the piston speed is at a maximum.
This was observed also by Hilton, (1991), who used a maximum phase angle range of 90 CA. In doing so, however, the lag range slot widths increased from 0.18°, for a lag range of 1.44 CA to 5.76° at 90.76 CA. In the range of 320 to 360 CA the instantaneous piston velocity tends rapidly to zero and the quasi-stationary assumption of the tumble flow indicates that velocity samples should correlate over only a small phase angle range. For a small lag range region, the autocorrelation function would be expected to change more significantly. Therefore, for high temporal resolution, or where the autocorrelation changes rapidly, the requirement is for the smallest possible phase angle slot width with the largest possible sample size to reduce the variance of the estimate.

Hong and Chen (1997) defined a Temporal Autocorrelation Coefficient, $R_t$

$$\begin{align*}
R_t(\overline{\theta}, \phi) &= \frac{1}{n} \sum_{i=1}^{n} \left[ \frac{u(\overline{\theta}, i) u(\overline{\theta} + \phi, i)}{u'_{EA}(\overline{\theta}) u'_{EA}(\overline{\theta} + \phi)} \right] 
\end{align*}$$

where $\phi$, the phase slot width angle with respect to crank angle, $\overline{\theta}$ is the chosen reference crank angle, $i$ is cycle number and $n$ is the total number of cycles. The ensemble-averaged turbulence intensity, $u'_{EA}$ is defined as

$$
\begin{align*}
u'_{EA}(\overline{\theta}) &= \left\{ \frac{1}{Nt(\overline{\theta})} \sum_{i=1}^{NC} [U(\overline{\theta} \pm \Delta \theta / 2, i) - \overline{U}(\overline{\theta}, i)] \right\}^{1/2}
\end{align*}$$

where $Nt$ is the total number of measurement in the crank angle ensemble-window and $NC$ is the total number of cycles. The integral time scale is defined as the area under $R_t$

$$\begin{align*}
\tau_I &= \int_0^{\phi_{max}} R_t(\overline{\theta}, \phi) d\phi
\end{align*}$$

and the integral length scale using Taylor’s hypothesis is obtained from

$$\begin{align*}
L_I &= \left[ u'_{EA}(\overline{\theta}) \right] \tau_I(\overline{\theta})
\end{align*}$$

The temporal autocorrelation coefficients were calculated at two in-cylinder, spark axis locations, in the longitudinal direction of the tumble plane. Tumble position 7 (3mm from the spark plug body) and Tumble position 6 (8 mm from the spark plug body) were selected. The ensemble-averaging time bin of 1.44 CA was utilised in this case as the phase angle range; that is 0.0 to 1.44 CA. Within that range the individual slot width was 0.36°. This choice gave a high crank angle resolution. At 1500 rpm, the temporal resolution of each slot is 0.04 ms. A minimum of 100 validated Doppler samples were available in each 0.36 CA slot. For Tumble 7, the mean cycle data rate was 0.417 kHz, the validation rate was 99.6% and data was collected over 2899 cycles. At Tumble location 6, the mean cycle data rate was 1.384 kHz, the validation rate was 99.5% and data was collected over 935 cycles. It should be noted however, that the average data rate in the crank angle interval chosen for the autocorrelation estimation is contained within the greatest peak of the validated count measurements presented in the relevant tumble plane plots. Within these regions, the instantaneous data rates are of an order of between 10 and 20 times that of the mean cycle data rate at crank angles close to TDC. The accuracy of the determination of the integral time scale is dependent upon the resolution of the autocorrelation function.
This is in turn dependent upon the validated data rate. Under ideal conditions, the validated data rate must be sampled at a rate that is much higher than the correlated time scale to ensure adequate resolution of the autocorrelation function. In this study, the mean validated sampling rate within the considered time intervals was between five and thirty times that of the integral time scale calculated over the same range.

At tumble location 7, three crank angle reference ranges were considered. The first was selected from 328.68 to 330.12 CA around typical spark ignition timing. The second was selected form 357.12 to 358.56 CA and the third, from 358.6 to 360 CA. At tumble location 6, the 330 and 360 CA ranges were selected. The autocorrelation coefficients were then estimated for each 0.36° slot in the phase lag range from the reference crank angle of interest using Equations 5.4 and 5.5. No averaging of the coefficients was performed between reference crank angles. The estimates were normalised (by dividing by the zero lag values), as is often presented in the literature. This ensures that the estimate normalises to a zero-lag value, $R_t(0)$, equal to unity. The non-normalised and normalised autocorrelation estimates are shown in Figures 5.30 and 5.31, respectively for the phase lag range of 1.44 CA. A common empirical form of the autocorrelation function that is a good representation of the temporal correlation of turbulent in-cylinder flow fields is an approximately exponential decay (e.g. Fansler and French (1988) and Hadded and Denbratt (1991)). These are plotted in the relevant figures and take the general form

$$R_t(\phi) = Ae^{-B\phi}$$  \hspace{1cm} (5.8)

where $A$ and $B$ are empirical constants. In Figure 5.30. The $R$-squared values (coefficient of determination) are included as an indication of how closely the values of an exponential curve fit correspond to the actual data over the correlated range. In Figure 5.31. the normalised autocorrelation estimates are plotted with an exponential curve fit. The tumble 7 location for a reference crank angle of 360 is the least correlated of all the measurement points and lag ranges. The point at 330 CA and tumble location 6 shows the best fit to an exponential decay.

To derive the integral time scale, the approximated exponential function of the non-normalised autocorrelation function was then integrated in the limit of $\phi,0 \rightarrow \infty$. This was performed for spark plug locations tumble 6 and 7 at 30 CA BTDC to aid comparison with the limited results previously published in the literature and tabulated in Chapter 2. Measured integral length scales in conventional gasoline geometries have been shown to scale with clearance height (independently of engine speed) and decrease to approximately one fifth of the clearance height at TDC. In the G-DI geometry at 330 CA, the clearance height can be considered as the sum of the piston displacement and bowl depth to spark plug gap. This is approximately 30 mm. For tumble location 6, the longitudinal integral time scale is estimated at 1.9 milliseconds and the length scale, using Taylors hypothesis, at 9.5 mm. At tumble location 7, at the spark plug gap, the integral time scale is of the order of 1.1 milliseconds and the integral length scale is estimated at 6.2 mm. These values are similar to the 6.9 mm estimated by Glover et al., (1988) at the spark plug gap in a pancake combustion chamber at 1200 rpm using scanning LDA and a spatial autocorrelation analysis. Hong and Chen (1997) estimated 6.0 mm in a pancake chamber at 500 rpm using a direct scale estimation from a two point LDA system.
In all cases, the autocorrelation coefficient peak was less than unity; that is validated measurements that occur with short temporal separations are not perfectly correlated. This has been reported in the literature by many research groups including Glover et al., (1988a, b), Hilton et al., (1991), Hong and Chen (1997). This form of the autocorrelation coefficient compared favourably with that obtained from simulated data used to construct an expected autocorrelation estimate. Glover et al., (1988a, b) and Hilton (1991) used simulated data that included a cycle-to-cycle variation in the mean flow. They concluded that these cyclic fluctuations were responsible for a slowly varying component in the autocorrelation estimates as they were not included in the ensemble-averaged estimate of the mean velocity. In Figure 5.30, the tumble 6 and 7 locations at 360 CA are most affected by cycle-to-cycle variations in the bulk flow, exhibiting a wavy type curve. Indeed, all the series of points show a degree of unsteadiness but this is less pronounced in the 330 CA case.

It is thought that this is in part due to system noise but also the low frequency tumble vortex centre jitter about the cross-tumble plane axis, reported by other research groups and manifested as an apparent turbulent contribution. In this analysis, no filtering method (based upon an arbitrary choice of turbulent cut-off frequency) has been applied to the raw data.

5.4. Assessment of Errors and Statistical Accuracy of the Mean and RMS Velocity Estimates

5.4.1. Experimental Uncertainties

The assessment of the contribution of experimental errors in the LDA measurements performed has been extensively covered in Chapter 3 and Chapter 4. In almost all cases of engine LDA measurements, it is the uncertainty due to the limited number of velocity samples in a crank angle window that is the significant factor in assessing experimental error.

The following appraisal relates to those errors specific to this series of tests, as the general experimental systematic errors were detailed in Chapter 4.

(a) Beam Intersection Position. For the spark plug axis measurements, the pitch of the spark plug probe adapter and the refraction calculations for the beam pair through the Perspex window combined to give an experimental error in the positioning of the probe volume within the combustion chamber. This was estimated to be within one turn of the probe holder. This equates to ±0.25 mm along the spark axis. The error in selection of the required spark plug depth based upon matching the number of screw pitch turns with the refraction through the 9.42 mm thick Perspex slug was of the order of ± 0.02 mm.

In the quartz annulus study, the position of the transmission optic on the optical bench and its reference to the mid-cylinder location, with refraction correction through the annulus, determined the uncertainty in the measurement location. This was estimated at ± 0.5 mm, the approximate visible thickness of the targeted LASER beam. The same order of error was observed in setting the height of the transmitting probe. The largest potential error was introduced due to the precision of the vertical orientation of the beam pair. Any out of plane rotation would lead to refraction effects through curved surfaces and the beam pair intersection would be shifted from the mid-cylinder tumble plane. The vertical alignment was performed on the engine with the cylinder head removed.
The surface of the engine upper barrel block was checked with a spirit level. A plumb line, suspended from a magnetic milling dial clamp at the centre of the cylinder, was used to target the beam pair. A final check was performed with the annulus placed on top of the barrel. After performing this procedure, the error in radial translation of the refracted beam pair through the quartz annulus was considered to be negligible.

(b) Optical Set-up. The estimated error in the optical set-up was discussed in Chapter 4. The largest estimated error was that due to the fringe spacing. This gave estimated velocity errors of less than 5%.

(c) Engine Specific Tests. The series of specific tests described in the results and analysis sections showed evidence of potential errors in the ensemble-averaged mean and RMS velocity estimates.

In summary, the repeatability tests revealed that potential errors of between ±1 ms⁻¹ could be expected through the compression stroke where insufficient data were collected over too few cycles. Simple sensitivity tests at tumble location 7 revealed that small displacements of up to ±1 mm of the measurement volume resulted in significant differences in the mean velocity and turbulence magnitudes during intake, but relatively small differences during the rest of the cycle. This was due to the position of the tumble 7 measurement point with respect to the intake valve. The particle residence weighting of individual ensemble averages also indicated that during particularly turbulent phases of the intake and compression strokes, a difference of ±1 ms⁻¹ could be expected between those averages weighted by the particle transit time and those simply phase averaged. This can introduce a large uncertainty where mean velocity values are low.

5.4.2. Statistical Accuracy of the Mean and RMS Velocity Estimates

The equations that describe the standard error in the mean and RMS estimates have been given previously and their application to LDA engine measurements is given in great detail in Chapter 4. Here, the result of applying these techniques to the G-DI data sets is discussed. Generally each data set comprises at least 50,000 instantaneous velocity measurements. The data arrival rate is not constant and the distribution of data samples within each crank angle interval varies with location and position in the cycle. This analysis is therefore limited to the intake and compression strokes for the motored operation of the engine.

Figures 5.32a, b, c and d. show the standard error (standard deviation of the sample) in the ensemble-averaged mean and RMS velocity estimates for a typical data set (tumble 03) over the four strokes. The mean validated data rate over all engine cycles was approximately 0.4 kHz. The total data collected was approximately 52,160 of which 52,150 were validated data points over approximately 3284 cycles at 1500 rpm. The standard error, ε in each crank angle window is plotted along with the ensemble-averaged mean data. The error bars on the mean data are the ±3ε (three times standard error) indicators. The probability of the true value being within these error bands is approximately 99.7 %. This applies to all but 6 of the 500 ensemble time bins. For clarity, only every second error bar is shown on the ensemble-averaged data. The results show that in the region of compression TDC, where the data rate is high and turbulence intensity relatively low, there is very good statistical accuracy.
The 3E value at TDC is 0.3 ms\(^{-1}\) in the mean estimate and 0.05 ms\(^{-1}\) in the RMS estimate. This compares favourably with Hilton, (1991), who recorded 0.2 ms\(^{-1}\) for a mean estimate in a gasoline, pent-roof combustion system. Elsewhere in less seeded and more turbulent parts of the cycle, Hilton, (1991) reported 3E values in the range of 1 ms\(^{-1}\) for a 450,000 data set point. In the present study, the highest 3E values were recorded close to maximum inlet valve lift and were of the order of 3 ms\(^{-1}\) for the mean and RMS estimate.

5.4.3. Velocity Histograms
The velocity probability distribution is often calculated over selected crank angle intervals to identify any distributions that may deviate significantly from a Gaussian form. A comparison is made between the distributions of the velocity samples at two locations in the engine cycle for tumble location 5.

These are shown in Figure 5.33., for a 10 CA interval centred around 330 and 360 CA. A velocity slot width of 1 ms\(^{-1}\) was chosen. The two curves show a degree of Gaussianity in the distribution of velocity samples. In the 330 CA plot, the distribution is approximately symmetrical about the mean. At 360 CA, the curve is asymmetrical and shifted towards the lower velocity samples. This is unusual in that velocity bias effects in single position LDA usually skew the distribution towards the higher velocities; that is the probability of obtaining a sample is related to its magnitude. That said, Hilton (1991) reported non-Gaussian distributions for single position data in some regions of low mean velocity. It is therefore likely that this is a particular feature of this measurement location. It can however be concluded that from results of single position LDA, sampling velocity bias is responsible for the departure from a Gaussian distribution and that the effect varies across the engine cycle. A quantitative measure of the degree of Gaussianity (e.g. skewness and kurtosis) was not covered in this analysis.

5.4.4. Crank Angle Broadening
The effect of crank angle broadening is illustrated in Figure 5.34a and b. for tumble location 3. Velocity gradient effects are most pronounced when ensemble averages are computed over large crank angle intervals and where the flow velocity is undergoing rapid acceleration. The ensemble-averaged mean velocity is plotted for the complete engine cycle. In each case the four stroke engine cycle is divided into 25, 50, 100, 250, 500 and 1000 time bins. This corresponds to crank angle intervals of 28.8, 14.4, 7.2, 2.88, 1.44, and 0.72 CA. The ensemble-averaging procedure is as presented previously. It is clear from both sets of plots that it is not until at least 250 time bins are used, that the mean curve starts to represent a good average and is able to follow some of the smaller perturbations. In the highly fluctuating parts of the cycle, the spread between 25 and 1000 time bins is at a maximum and diminishes towards TDC. Crank angle broadening can be approximately quantified by the ratio of the variation in the ensemble-averaged mean velocity to the RMS velocity across the averaging window. For an averaging window centred about 100 CA, the crank angle broadening for the above crank angle intervals is shown in Table 5.2. The crank angle ratio centred at this crank angle generally decreases with decreasing ensemble-averaging crank angle interval.
The difference in mean velocity difference decreases with a reduction in interval width. The smaller intervals follow the RMS fluctuations more readily and are less likely to smooth out peaks and troughs as observed in the larger intervals. The ratio is a compromise and it is therefore important that these results should be utilised in conjunction with the standard error estimation described above.

5.5. Conclusions of Chapter 5

The characteristics of the bulk and small-scale air motions in a top-entry, intake geometry, gasoline, direct injection engine were studied using LDA. Two methods of optical access to the combustion chamber permitted the measurement of the instantaneous gas velocity in the radial and axial directions in the mid-cylinder tumble and cross-tumble planes. Radial gas velocities were measured along an axis, co-linear with the spark plug axis and inclined at 8° to the vertical cylinder axis. The measurements were made in the pent-roof of the chamber and to a depth of 33 mm from the apex of the chamber, which extended into the bowl of the piston crown. Axial velocity measurements were made across the cylinder bore at a depth of 10 mm from the cylinder head gas face. In each case, instantaneous velocity measurements were collected over consecutive engine cycles until a statistically significant number of results were achieved within each discrete crank angle interval.

The results show that a large-scale reverse tumble motion is set-up very early in the intake period. This was contrary to the Volvo findings, where the forward tumble motion was not established until IVC. In the G-DI, the mean velocity and RMS turbulence intensity measurements in the tumble plane were of much greater magnitude than those measured in the cross-tumble plane. This indicated that the principal bulk flow motion was in the tumble plane and that there existed only a relatively small amount of ‘out-of plane’ motion. The intake phase was dominated by a rapid inflow of gas, with mean radial velocities in the tumble plane, in excess of 60 ms\(^{-1}\). These occurred in the jet flow regions, above and below the intake valve curtains. The intake velocities were greater in magnitude than those measured at the same locations in the Volvo study, when normalised by mean piston speed. This would suggest that the jet flows were less restricted in the upright intake and bowled chamber configuration, despite the fact that for comparable valve lifts, the ratio of the intake valve cross-sectional area to the bore cross-sectional area is greater in the G-DI case. In the MPI engine, the bore area is 4.5 times greater than the combined valve areas, whereas the G-DI bore area is only 3.84 times greater than the two valve areas. This result is also observed in the measurements of the axial velocity component, where the curved intake port and sharply angled intake valve seat, deflect the flow away from the cylinder wall and into the centre of the piston bowl. In both studies, the RMS velocities at these locations were reduced in the strong jet flow region.

The persistence of a coherent, reverse tumbling body in the compression stroke was recorded by the points distributed along the spark plug axis. The sequential velocity reversal at these spatially fixed measurement locations indicated that the tumble was driven up the cylinder by the piston motion and experienced a ‘spin-up’ as the piston approached TDC. The structure was first recorded at the lowest measurement point in the piston bowl and was measured up to and beyond TDC firing. The motion was then observed to slowly decay until EVO. These measurements (excepting one) did not show an increase in turbulence intensity, often used to quantify tumble breakdown in MPI chambers or to identify squish flows in Diesel chambers.
The turbulence intensity at TDC was estimated at between 20% and 50% depending upon location. Tumble location 5, however, showed an increase in turbulence intensity from 340 CA. At this angle, tumble 5 was at the geometrical centre of the piston bowl. If it is assumed that the vortex fills all the available area, then the mean and RMS velocities recorded at tumble 5 will have instantaneous velocity contributions from both the positive and negative measurement directions. In the cross-tumble plane, the instantaneous velocities fluctuated symmetrically about zero velocity; with the greatest RMS velocity fluctuations recorded during the jet flow interactions at peak valve lift. There was no discernible increase in the measured RMS velocity magnitude during the compression stroke.

A series of logical steps were carried out to assess whether the in-cylinder air motion during the late compression strokes could be treated using Taylor's hypothesis for stationary, turbulent flow. In the first instance, a tumble vortex ratio (TTVR) was defined in order to establish whether a coherent, quasi-steady structure existed at a typical spark ignition timing and at TDC. From 340 to 350 CA, the tumble vortex angular momentum approached that of a solid body. From 350 CA to TDC, the tumble ratio remained constant with a final value of 0.83. Secondly, the rate of tumble vortex centre precession along the spark axis relative to the geometrical piston bowl centre was calculated using a moving origin approach for a series of crank angles over the same period. The results showed that the relationship was linear and that the tumble centre velocity relative to the piston position was 0.64 ms\(^{-1}\) between 320 and 325 CA. In the third analysis, the ensemble-averaged RMS velocity fluctuation close to TDC was compared with those measured in the Volvo combustion chamber at the same location. In the Volvo engine, there was a marked increase in RMS values from 330 CA. In the G-DI, the RMS velocity was approximately constant up to 350 CA.

The mean velocity over these ranges was between 5 and 10 times greater than the RMS contribution close to the spark plug gap. Based upon these results, Taylor's hypothesis was used to calculate the integral length scale of turbulence from the integral time scale derived by integration of the temporal autocorrelation function. The temporal autocorrelation coefficients were calculated at two locations and for 3, 1.44° CA intervals at ignition and TDC. The best correlations for an exponential curve fit of the autocorrelation coefficient estimates were obtained at tumble 6, for 330 CA and tumble 7, for 359 CA. The integral length scales were calculated at a spark ignition angle of 330 CA for tumble locations 6 and 7. At tumble location 6, 8 mm from the spark plug body, the longitudinal integral length scale was estimated at 9.5 mm. At tumble location 7, at the spark plug gap, 3 mm from the spark body, the integral scale was estimated at 6.2 mm. This compared favourably with other studies that suggested that the integral scale decreased to approximately one fifth of the clearance height at TDC. In the G-DI combustion chamber, the clearance height based upon the depth of the piston bowl is approximately 30 mm.

A series of specific tests showed that the ensemble-averaged estimates were repeatable; that the sampling rate could only be considered continuous at a few select points in the cycle; that the RMS but not the mean velocities scaled with engine speed during the intake stroke and that intake throttling introduced two velocity peaks in the intake, altered the magnitude of the radial mean velocity at BDC and the direction of the radial component during the compression stroke.

A quantitative appraisal of velocity bias, crank angle broadening and the standard error in the ensemble-average estimates showed very good statistical accuracy in the presented results and highlighted the requirement for adequate particle seeding of the airflow in highly turbulent regions of the engine cycle. In well-seeded parts of the cycle, the ensemble-average velocity estimate always over predicted the weighted velocity estimate. The maximum difference was 1 ms\(^{-1}\) in the range 25-30 ms\(^{-1}\). For tumble point 3, all but 6 of the 500 ensemble intervals showed data that was outside of the ±3 standard error deviations. These values were 0.3 and 0.05 ms\(^{-1}\) for the mean and RMS velocity estimates respectively.
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![Diagram](image)

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Repeatability of Results

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Repeatability of Results

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800 mbar intake / 32 mm depth / 500 bins

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![Graph showing the comparison of the Temporal Autocorrelation Coefficient for Indirect Integral Length Scale Determination at Tumble Locations 6 and 7, about a Typical Ignition Angle and at TDC.](image)

Figure 5.30. Comparison of the Temporal Autocorrelation Coefficient for Indirect Integral Length Scale Determination at Tumble Locations 6 and 7, about a Typical Ignition Angle and at TDC
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<table>
<thead>
<tr>
<th><strong>Base Engine</strong></th>
<th><strong>Ricardo Mk1 Optical Hydra Engine</strong></th>
</tr>
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<tbody>
<tr>
<td>Cylinder Head</td>
<td>Ricardo RCE161, 0.325 litre, DOHC, Four-stroke, Single, Direct Fuel Injection</td>
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<td>No of Cylinders</td>
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<td>No of Valves</td>
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<td>Compression Ratio</td>
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<td>Combustion Chamber Geometry</td>
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<td>Piston geometry</td>
<td>Hemispherical, offset bowl in piston with valve cut-outs</td>
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<td>Spark Plug Location</td>
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<td>Injector Location</td>
<td>Intake side, between valves inclined 36° to horizontal</td>
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<td>Intake System</td>
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<td>Exhaust System</td>
<td>Straight, side exit, siamesed port</td>
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<td>Direction of Air Motion</td>
<td>Reverse tumble</td>
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<td>Valve Timing</td>
<td>IVO 16° BTDC</td>
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<td></td>
<td>IVC 48° ABDC</td>
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<td>EVO 46° BBDC</td>
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<td></td>
<td>EVC 18° ATDC</td>
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<td>Maximum Valve Lift</td>
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<td>Intake Valve Diameter</td>
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<td>Fuel Injector</td>
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<td>Operating Pressure</td>
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<tr>
<td>Optical access</td>
<td>‘Through-spark plug’, quartz annular slice and ‘Three-windowed’ cylinder head</td>
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**Table 5.0. Direct Injection Engine Specifications**
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<table>
<thead>
<tr>
<th>Bin Number</th>
<th>Centre Angle</th>
<th>Interval</th>
<th>$\Delta U$</th>
<th>$\Delta u'$</th>
<th>Ratio</th>
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<td>0.7</td>
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<td>100</td>
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<td>1.1</td>
<td>0.25</td>
<td>4.4</td>
</tr>
<tr>
<td>100</td>
<td>100</td>
<td>96.4-103.6</td>
<td>0.9</td>
<td>0.5</td>
<td>1.8</td>
</tr>
<tr>
<td>250</td>
<td>100</td>
<td>98.56-101.44</td>
<td>0.5</td>
<td>0.55</td>
<td>0.91</td>
</tr>
<tr>
<td>500</td>
<td>100</td>
<td>99.28-100.72</td>
<td>0.2</td>
<td>0.8</td>
<td>0.25</td>
</tr>
<tr>
<td>1000</td>
<td>100</td>
<td>99.64-100.36</td>
<td>0.35</td>
<td>0.5</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Table 5.2. Effect of Crank Angle Broadening about 100 CA at Tumble Location 3

In Chapter 5, the airflow characteristics in a direct injection, pulse-mesh injection engine were analyzed in detail. The combustion strategies utilized in such an engine require careful consideration of the interaction between the in-cylinder flow fields and the injector fuel spray properties. Typically, airflow motion and fuel spray characteristics are investigated separately due to the difficulty in achieving such measurements. The fuel spray is often visualized in an atmospheric or low-pressure chamber with good optical access but in the absence of any air motion. In-cylinder fuel spray measurements, when compared with diagnostic chamber results, provide insight into the influence of the complex interactions that take place throughout the intake and compression strokes. The control of the production of the air and fuel mixture is essential; earlier processes are critical to obtaining combustion stability in GDI concepts operating in the high load. In addition, it should be possible to relate the instability in the combustion process to properties of the air-fuel spray system. In practical terms, the interactions are far more complex, involving many more variables, both instantaneous, in-situ measures of air and fuel spray become difficult to simulate the critical events for subsequent analyses.

Chapter 1 reviewed the literature pertinent to the nanoliter portion of the liquid droplet and properties of fuel injectors and fuel spray for numerous direct injection gasoline engine concepts. Much of the empirical and theoretical data utilized in these approaches was gathered from extensive studies involving high-pressure Diesel engines applied to direct-injection Diesel combustion systems. In these studies, the central spray structures are observed to form over a short distance termed the "atomization break-up regime." The spray regime is not formed instantaneously from the injection liquid at injection but requires a finite time to establish a different, "steady-state" state regime. Any instabilities in this break-up process that manifested as short-to-short variations in the spray shape and penetration length.
6.0. The Characteristics of a High-Pressure Fuel Spray for a Gasoline Direct Injection Engine: A Quasi-Steady State and Dynamic Fuel Spray Study

6.1. Introduction to Chapter 6

Chapter 6 investigates the influence of the air motion on the turbulent fuel spray jet break-up and trajectory, geometrical form, penetration, air entrainment and evaporation and the spatial and temporal distribution of droplet diameters and velocities and their role in fuel-air mixing. More efficient air-fuel mixing can be achieved through increased injection pressures (better atomisation); through the control of injection scheduling and duration and through the introduction of controlled and structured air motions within the combustion chamber. The short time period available for injection, mixing and evaporation in a direct injection strategy, highlights the additional role of such injectors over more standard MPI types. The aforementioned relationships may be determined through a comparative study of the fundamental spray properties under ambient, quiescent conditions with those observed during motored, dynamic engine conditions. In a direct fuel injection strategy, the need for such control is twofold; a fuel spray strategy is required that can accommodate both homogeneous and stratified charge engine operation and the inherently different elements that comprise the air flows; swirl, tumble and turbulence intensity, encountered within each case.

In Chapter 5, the airflow characteristics in a direct injection, pent-roof, gasoline engine were analysed in detail. The combustion strategies utilised in such an engine require careful consideration of the interaction between the in-cylinder flowfield and the injected fuel spray properties. Typically, air motion and fuel spray characteristics are investigated separately due to the difficulty in achieving such measures. The fuel spray is often characterised in an atmospheric or low-pressure chamber with good optical access but in the absence of any air motion. In-cylinder fuel spray measurements, when compared with quiescent chamber results, provide insight into the influence of the complex interactions that take place throughout the intake and compression strokes. The correct preparation of the air and fuel mixture in a repeatable, stable process is central to obtaining combustion stability in G-DI concepts operating at the lean limits. In addition, it should be possible to relate the instabilities in the combustion processes to properties of the air-fuel spray mixture. In practical terms, the interactions are far more complex, containing many more variables, but nonetheless, in-situ measures of air and fuel sprays serve to establish the criteria for subsequent analyses.

Chapter 1 reviewed the literature pertinent to the research concerned with the selection criteria and properties of fuel injectors and fuel sprays for numerous direct injection gasoline engine concepts. Much of the empirical and theoretical data utilised in these approaches was gathered from extensive studies involving high-pressure sprays applied to direct injection Diesel combustion systems. In these studies, the conical spray structure is observed to form over a short distance termed the 'atomisation break-up regime'. The spray structure is not formed instantaneously from the issuing liquid at injection but requires a finite time to establish a coherent, ‘quasi-steady state’ regime. Any variations in this break-up process are manifested as shot-to-shot variations in the spray shape and penetration length.
In a PDA study of fuel spray characteristics, the penetration length of a spray can only be considered in terms of the distance travelled by atomised fuel droplets. Photographic shadowgraphs or illuminated high-speed film can reveal penetration of contiguous ligaments or cords of fuel that would otherwise pass unrecorded by the PDA validation criteria. However, these methods depend largely upon an ambiguous intensity threshold limit that can distinguish between the liquid spray and the background light level and as such, can only be utilised subjectively.

As a means to achieving these aims, the physical properties of a high-pressure, direct injection, single fluid, pressure-swirl atomiser were studied under varying conditions. An initial series of tests were performed in a low-pressure, quiescent air chamber to establish the range and characteristics of the measurable parameters required. In addition, this ensured that the PDA system could be set-up, assessed for sensitivity to operating parameters and ‘tuned’ in a simple and familiar environment. The injector was then placed into the engine configuration fitted with the quartz annulus as described in Chapter 5. PDA measurements of the fuel spray droplet diameters and velocities were made along the same plane and at the same locations as those relating to air motion described in Chapter 5. The tests were repeated with the engine at rest and then under motored operation. The large-scale structure of the spray under static and motored engine conditions was photographed using LASER light sheet illumination and shadowgraphy, under the same test conditions. In all cases, injection characteristics were matched to real engine conditions at low speed, full and part load operating points.

6.2. Fuel Spray Measurements in an Ambient Pressure, Quiescent Fuel Spray Chamber

6.2.1. Fuel Injector Properties

The fuel injector used in these studies was manufactured by Siemens Automotive. It is a high-pressure, swirl injector, type T5-853. The nominal operating injection pressure is 100 bar. Unleaded pump grade 95 RON gasoline fuel was utilised in all tests. The injector was triggered by a TTL rising edge input to a Siemens Automotive ECU/Driver Unit. The ECU supplies a conditioned signal to the injector solenoid at 12V and up to a maximum of 10A. The width of the TTL pulse determines the injection duration. In these studies, the injection timings (SOI) and durations (PW) were chosen to match representative fired engine injection timings for WOT, homogeneous and stratified charge operation at a range of engine speeds applicable to the optical engine. A full description of the injector required measurement of the orifice size and fuel mass delivery rate.

6.2.1.1. Determination of Injector Orifice Diameter

The injector orifice diameter was determined by examination of the injector tip using a compound microscope instrumented with a television camera. This was performed at the University of Brighton, Electron Microscopy Unit (EMU). The camera was linked to measurement software that allowed the superimposition of simple geometrical shapes onto the magnified image. Once calibrated, the dimensions of the shapes were output as an ASCII file and a TV camera captured colour stills of viewable images.
Three overlay regions were selected from the video images of the injector orifice hole. Outer, middle and inner rings were superimposed such that:

(a). The outer ring circle interpreted by eye to ‘best fit’ all features on outer most rim of nozzle orifice as perceived on nozzle surface.
(b). The middle circle circle interpreted by eye to ‘best fit’ observed narrowing of orifice from outer to inner at ‘some’ depth of focus within the orifice.
(c). The inner circle circle that ‘just touched’ an apparent deformation or fouling deposit evident deep within the orifice.

The results of geometry overlaying are presented in Table 6.0. and Figure 6.0. along with the captured images that clearly show an inner surface deformation or fouling deposit. The middle ring was considered as the best approximation to the orifice diameter at 270 \( \mu \text{m} \).

### 6.2.1.2. Injector Fuel Mass Flow Rate

Measurement of the fuel injection mass flow rate was required to ensure that fired and motored engine air-to-fuel ratios were correctly replicated. The dynamic flow performance of a direct injection fuel injector plays a more critical role than that of an MPI injector due to the significantly shorter injection periods and higher fuel mass flow rates. The injector manufacturers’ mass flow data was available for a fuel pressure of 68.97 bar and therefore tests were carried out as close as possible to this pressure to ensure validity in the method. Further tests were then carried out at the nominal injection pressure of 100 bar to determine the mass of fuel per injection. Additionally, the high-pressure fuel spray rig required calibration. Therefore, both the fuel rig and injector were tested together and the results compared to several other sources of injector data. The method involved the use of the Engine Simulator and Cycle Resolver and Timing Unit. This enabled the simulation of all the signals necessary to trigger the injector driver unit and user-defined injection duty cycles. The same timing unit was also used during engine operation and input with an encoder pulse train and crank and cam reference markers. A simulated engine speed of 500 rpm was selected with an arbitrary SOI crank angle and injection pulse width range as summarised in Table 6.1. Injection pressures were chosen at 70 bar, 85 bar and 100 bar for best comparison with available data. For all tests the injector was clamped into a machined plate and injected fuel was collected in a lightweight, sealed metal container. This minimised errors due to splashing and escaping vapour that led to uncertainties in injected masses.

The first set of tests was conducted at 100 bar fuel pressure utilising the measuring burettes provided on the side of the Fuel Spray Rig. The time taken for the injection of a known volume of fuel was recorded once the pumps and low/high pressure circuits were primed. The second set of tests were based on fuel injection recorded over a known time interval recorded with a stop-watch. The resultant fuel mass was recorded to a precision of \( \pm 0.1 \text{g} \) using calibrated 0-200g optical scales. The scales were re-calibrated before each set of measurements. The measurement container was weighed before and after the tests to eliminate errors due to fuel resting in any small crevices. After each series of injections the container was left to settle for 1 minute to ensure the vapour had condensed before the plate was removed. A preliminary test was used to ascertain the significance of the timing interval duration on mass per injection. For injection pressures of 70 bar, mass flow rates for fixed time intervals of 150 s, 300 s, and 600 s were recorded.
These are plotted with the other results for comparison. It was considered that 300 s (1250 injections at 500 rpm) was sufficient within the accuracy of the experiment to provide good data for the comparison. The available comparison data is summarised in Table 6.2. The available data sources for comparison were as follows:

(a). Siemens Automotive Manufacturers Dynamic Flow Test Data. 11/15/94. Stoddard Test Fluid injected at 88.97 bar and pulse width range of 0.20 - 4.5 ms. Siemens identified a linearity region from between 0.5 ms to 3 ms (within ± 5%).

(b). Ricardo T5-853 G-DI Load Range Curves at 1500 rpm. 12/09/96. 95 RON SURGE injected at 100 bar. MBT with varying BMEP range from 1.5 – 8.32 bar.

Table 6.3. summarises the series of timed injection tests performed. Figure 6.1. shows the plots for all measured dynamic flow rates with those used for comparison. The timed mass experiments at 70 bar showed good agreement with the manufacturers data for short pulse widths. At 85 bar, a similar mass flow rate is observed over the full range of pulse widths investigated. At 100 bar, the data recorded at Brighton matches that provided by Ricardo. With an increase in pressure, the results show that for a typical injection pulse of 1.9 ms, there is a difference of approximately 0.8 mg of fuel injected (10% of the total duty) between that recorded by the Ricardo and University of Brighton at 100 bar, and the data supplied by Siemens at 88.97 bar. Therefore, for the purposes of the work carried out in this study, the 100 bar timed mass flow rates were utilised for injection timing purposes.

6.2.2. Experimental Fuel Spray Test Chamber

An atmospheric pressure and temperature chamber was constructed for the investigation of the fuel spray characteristics of the pressure-swirl injector and to enable off-engine set-up of the PDA system. The test rig is shown schematically in Figure 6.2. and photographically in Figures 6.3a, and b. The chamber is comprised of a rigid steel frame that holds four Perspex sheets that make up the walls of the chamber. A 50 mm thick steel plate on top of the chamber was machined to allow the injector to be clamped into a vertical position. The injector tip protruded 2 mm into the chamber. A second steel plate was used as the base of the chamber and was machined to allow liquid fuel to drain to the centre. A small tap underneath the base plate allowed fuel to be drained off to a beaker. The frame and metal plates were held together by the use of four threaded bars that ran the height of the rig and allowed the assembly to be clamped in position. A cork gasket was utilised between the frame and plates to ensure that fuel vapour did not leak from the chamber.

Preliminary injection tests revealed that the chamber filled rapidly with fuel vapour for even relatively low injection rates. A low-speed purge system was then utilised to remove spent fuel vapour from the chamber without altering the injected fuel spray formation. Fine wire gauze was inserted into the bottom of the chamber in several layers. The top layer of gauze was the coarsest, with subsequent layers becoming much finer. When the fuel was injected, the plume passed through the layers of gauze breaking up the droplets but without splashing liquid throughout the chamber. The lower part of the chamber then became filled with fuel vapour.
Part of the injected liquid stayed within the gauzes and dripped to the bottom of the chamber to be collected. A small air jet was positioned between the gauzes and an extraction pipe fitted to the other side of the chamber. The extraction pipe was attached to the cyclone filtration system. A throttle valve regulated the airflow rate through the cyclone. Air from the small jet was then utilised to move the spray vapour below the gauze towards the extraction side of the chamber. The modification to the chamber allowed PDA measurements to be made at injection rates of up to 1 Hz without significant signal deterioration.

The complete chamber was mounted upon a rectilinear, automated traverse fixed to an optical, air-cushioned, breadboard. This was used to position the chamber relative to the PDA transmission and collection optics. A CNC software routine allowed probe volume referencing in the x-y plane of the surface of the optical breadboard, relative to the injector orifice. The z-height adjustment of the transmission and collection optics was manual. At a given height, the traverse was utilised to move the spray through the probe volume intersection in fixed increments.

6.2.3. Assessment of Spatial Measurement Location in Fuel Spray Chamber

An important criterion in the optical configuration for measuring sprays is the avoidance of multiple droplet occupancy in the probe volume. The effect is most pronounced in dense regions of the spray where the incident beams are obscured by droplets. Ruff and Faeth (1995) report that current signal processors are capable of rejecting these signals based upon inferior signal quality. They suggest that in such cases, a high data rate must be maintained to ensure valid signals. In addition, they noted that attempts to eliminate spurious signals through reducing the PMT gain would only serve to bias the measurements towards the larger droplets. Ren et al., (1998) performed ensemble-averaged PDA measurements on a pressure swirl gasoline injector at 70 bar into atmospheric conditions. The closest distance to the nozzle that they considered was 15 mm. With their PDA set-up, working distances nearer to the nozzle resulted in validation rates of less than 50%.

Three axial measurement distances from the injector tip were chosen in a single vertical plane that bisected the plane of the injector nozzle orifice. A preliminary study was conducted to determine the minimum centre-line distance from the nozzle at which valid PDA measurements of the droplet sizes could be recorded. At distances of less than 8 mm from the injector tip, the PDA validated data rate fell to below 30% and the signal processor consistency error check exceeded 25%. Generally good working practice should ensure that this value is always less than 10%. This suggested that for this particular PDA configuration, the confidence in measurements performed for distances of less than 8 mm from the nozzle would be low. Measurements were thus performed at three axial distances from the nozzle exit: 8, 12 and 20 mm. These are approximately equivalent to 30, 45 and 75 nozzle diameters downstream of the orifice respectively. The maximum distance presented within these results was 20 mm. This was comparable to the limiting thickness of the optical annulus of the engine.
6.2.4. PDA System Set-up

The PDA system set-up was initially carried out in the manner extensively covered in Chapter 3. In addition, several other tests were carried out to further optimise the rate of acquisition of valid data. These tests served to provide a check of the system set-up when poor data rates were recorded in the dense, near nozzle regions of the fuel spray. These were performed on a monodisperse water spray (refractive index of 1.334) injected into free air. The spray was continuously generated by a pharmaceutical nebuliser. The characteristics of the water spray are well known and as such, the nebuliser is often reported in the literature as a reliable means by which a PDA system can be 'tuned' to minimise measurement errors and return signal validation rates that tend towards 100 % (e.g. Koo and Martin, (1991)). The experimental set-up for the PDA sensitivity tests is shown in Figure 6.4. In each case, the beam polarities and intensities were verified as well as the Bragg cell drive signal and PMT high-voltage balance levels. The PMT signal quality and balance were checked on a four-channel digital storage oscilloscope. The data acceptance rate and error checks were verified by using the internal trigger burst detector mode at 0.8 or 26 kHz. The method of PMT 'high-voltage plateau', as described in Chapter 3, was used to determine the droplet diameter measuring sensitivity. A study of the effect of spherical and SNR validation was also conducted. In all cases the PDA is operated initially as a 1-D, forward scatter LDA system. The validation rate is maximised in this mode, before the droplet sizing is turned on.

6.2.4.1. Determination of Optimum System Configuration

(a) LASER and Phase Doppler Validity Checks

The water spray generated by the pharmaceutical nebuliser was utilised to compare the mean droplet velocities obtained using forward scatter LDA and PDA. A transmission lens with a focal length of 400 mm and a beam separation of 32 mm was selected. The probe volume was placed close to the nebuliser orifice and the PDA collection optic aligned to and focused upon the beam intersection. The diameter diagnostic dialogue in the PC software was then utilised to fine tune the focus, measurement angle and height alignment of the instrument. The first test was performed using 1-D LDA, optimised to give the highest possible data validation rate with a low level of PMT voltage.

The PDA mode was then selected and the PMT balance set using a four-channel digital storage oscilloscope. A comparison of the two data sets is presented in Figure 6.5. for 15000 validated measurements. The data was sorted into 200 equi-spaced velocity bins over the total measured velocity range of ± 3.862 ms⁻¹. The results were then plotted over a smaller velocity range of -0.3 to 1.0 ms⁻¹. The mean value and ± 3 standard deviations from the mean are denoted by the dotted lines. Generally, all but a few of the measured and validated droplets fell outside the significance levels. For the LDA measurements, the data validation was 100 % and the data rate was 0.712 kHz. The total time for acquisition was 21 seconds. With the PDA turned on, the attempted recorded sample size was 24557, of which 91% were validated. However, only 15000 size measurements were validated within the relatively high 15% sphericity band. Hence, 7440 validated velocity measurements were not recorded as 'valid' diameters. The data rate in the PDA experiment was 0.523 kHz and the elapsed time, 47 seconds. The mean velocity recorded by the LDA was 0.3691 ms⁻¹ compared to 0.3416 ms⁻¹ measured with the droplet sizing included.
This equates to a 7.45% reduction in mean velocity. All further measurements reported were taken by the LDA and PDA separately, except for the size-velocity correlation plots.

(b) Effect of Probe Volume Dimensions
A series of tests were performed on the water spray using three different transmission lenses with focal lengths of 50, 160 and 400 mm and beam separations of 8, 32 and 32 mm respectively. The 160 and 400 mm lens had a 4 times beam expansion. The aim of the study was to assess the effect of the probe volume size and fringe spacing upon the measured droplet diameters and velocity profiles. Preliminary tests confirmed that the Nebuliser spray velocity and droplet diameters could be measured repeatably with a high degree of confidence with a single lens set. Velocity and diameter measurements were carried out independently using the LDA and then the PDA until 50,000 measurements had been collected. For each measurement set, the probe volume was focused at the same location by the method of anode current peaking using a small pin probe.

All PDA parameters were kept identical excepting the spherical validation that was varied from between 10 and 15% to observe the effects. The results are plotted in Figure 6.6, for measured mean values against increasing probe volume fringe spacing. In the first instance, the mean droplet velocities as measured by the LDA and PDA show good agreement and only a small variation between the probe volumes. In contrast, the PDA mean droplet diameter is observed to fall markedly for the 32/400 setting. Additionally, there is a fall in the percentage of validated droplets to the total attempted. The validated data rate however increases with fringe spacing. The data rate for the 32/400 probe volume is approximately twice that of the 8/50 and three times that of the 32/160. The probe volume length of the 32/400 is approximately four times that of the 8/50 and six times that of the 32/160.

The significant drop in mean diameter observed with the 32/400 lens was investigated by studying the distribution of the droplets for all sets, sorted into equi-spaced diameter bins. The 32/400 lens data set showed a significant number of counts below 2 μm that had skewed the distribution towards a lower mean value. These amounted to approximately 5000 droplets or 10% of the total validated population. In the 8/50 and the 32/160 data sets, there was a distinct absence of droplets below this level. The droplet diameter distribution statistics were then recalculated with all droplets below 2 μm removed. The resulting resorted mean droplet diameter is plotted for reference and falls within 4% of the 32/160 measurement and 12% of the 8/50 measurement. All measurement sets appear insensitive to the change in spherical validation from 10 to 15% for this spray.

The results suggest that the complex interaction between the mean droplet diameter distribution, fringe pattern construction, probe volume length and validated data rate and percentage is as much a function of the characteristics of the fuel spray than the system settings. The 8/50 and 32/160 lens showed the best agreement between the droplet size and velocity measurements. The 8/50 lens was physically constrained in its attainable measurement sweep across the cylinder bore. The 32/400 lens has a long probe volume length and is most likely to generate spurious signals in dense regions of a spray. It also produced a larger maximum diameter range (at least twice that of the other two sets) and therefore was liable to increased uncertainty in the phase angle determination.
The 32/160 lens was thus selected for the fuel spray work based upon the above study and the expected droplet diameter distributions in the fuel spray estimated from the literature.

6.2.5. Experimental Method for Fuel Spray Chamber

The characteristics of the fuel spray were recorded at each of the three axial distances from the injector orifice. The traverse was used to move the spray relative to the fixed PDA measurement location. The spray was not assumed symmetrical and an initial starting reference point was chosen close to the wall of the chamber, far from the spray plume and input to the computer software. The traverse was then jogged towards the spray in 1 and 2 mm increments until PMT activity was detected on the Doppler processor monitor channel. The increments were varied depending upon the axial distance and the width of the spray and were chosen as follows: at \( z=-8 \), 18 steps of 1 mm; at \( z=-12 \), 14 steps of 2 mm and at \( z=-20 \) mm, 16 steps of 2 mm. This gave an approximate spray coverage of 17 mm at \( z=-8 \) mm, 26 mm at \( z=-12 \) mm and 30 mm at \( z=-20 \) mm. At each measurement location, the PMT high-voltage levels, high-voltage balance, system gain and bandwidth were chosen to ensure that the complete droplet velocity range was included; the PMT voltage was high enough to detect the smallest droplets; the PMT voltage was as low as possible for this condition so as not to accentuate signal noise and to ensure that the mean droplet diameter was insensitive to small changes in PMT voltage at this level. The optimum settings for signal validation were as follows: the SNR was set in the range of \(-3\)dB to \(-6\)dB and the system gain to 'high'. The maximum closed-loop phase error was set to 15 degrees and the maximum spherical deviation was set at 5% in this initial study. The PDA settings used are given in Table 6.4.

A signal generator was used to supply the fuel injection driver circuit with a simulated injection pulse. The rate of pulses and the rising edge pulse duration could be varied. In this way, simulated skip injection was utilised to eliminate vapour build-up within the chamber. A simultaneous TTL signal was supplied to the Dantec PDA processor to indicate the start of the internal arrival time clock. The mode of triggering was selected so that the arrival clock did not reset at each trigger interval allowing absolute (total time-averaged) data to be collected. No burst inhibit gating of the burst detection mode was used.

The fuel pressure was regulated at 100 bar and in the temperature range of \( 23 \pm 1 \) °C. The ambient pressure and temperature within the spray chamber were approximately 762 mmHg and \( 22 \pm 1 \) °C respectively. At each measurement location 15,000 validated Doppler data events were collected. This ensured that all measurements were within \( \pm 3 \) standard deviations of the mean value. The number of injections required to meet these criteria varied with measurement location within the spray and the validated data rates achieved. In addition, the velocity measurements were repeated using LDA to investigate the effects of spherical validation upon the temporally-averaged results. Finally, the repeatability of the traverse positioning and data acquisition was verified at two adjacent radial locations in the \( z=-8 \) axial plane.
6.3. Fuel Spray Measurements in a Stationary and Motored Direct Injection Engine.

6.3.1. Experimental Test Procedure
High-pressure gasoline fuel spray droplet diameters and velocities were measured in a top-entry, wall-guided, direct injection gasoline engine. For briefness, the principal differences in the experimental approach with that utilised in the ambient pressure chamber are summarised in Table 6.5. The first set of measurements was performed with the engine at rest and the piston positioned late in the exhaust stroke. The cyclone filtration system was used to extract spent vapour from the chamber through the exhaust ports. The second set of measurements were carried out with the engine motored by the dynamometer. The SOI and pulse duration of the fuel injection event were varied using the Engine Timing Resolver Unit.

6.3.2. Optical Engine Configuration
Two different optical engine configurations were utilised for the PDA and photographic studies. The geometry of the cylinder head and combustion chamber in each case remained unchanged. The injector was angled at 36° to the horizontal (cylinder head gas face) and the spark axis, at 8° to the vertical cylinder axis. Both of the cylinder heads were instrumented with a Kistler type 6121 piezoelectric pressure transducer and type 5001 charge amplifier. These instruments were calibrated every 100 engine hours to give a linear 10 mechanical units (bar) per volt output. The transducer sensitivity was 16 pC per mechanical unit. The Low pass filter was set at 30 kHz and the time constant (high-pass filter) was set at 100 seconds (medium setting). The transducer and charge amplifier were powered up for a minimum of two hours prior to each test and reset between each set to eliminate instrument drift. Both optical engine configurations were instrumented with a Kistler piezoresistive absolute pressure transducer and a thermocouple within the intake plenum. The transducer was calibrated to give a linear response of 0.5 bar per volt over a 0 to 5 bar absolute range. The intake thermocouple recorded a hot, motored intake temperature of 30 °C ± 1°C. The ambient laboratory temperature increased from 22 °C to 25 °C over the duration of the testing.

The PDA transmission and collection optics were positioned around the engine cylinder head observing the forward scattering angle of 70°. The system installation is shown in Figures 6.7a. and b. A preliminary digital image of the spray plume within the chamber relative to the vertical beam pair is shown in Figure 6.8. The Sony Digital Video camera was utilised to capture the injection event with the aim to providing a 'sense' for global spray structure and plume position with respect to the beam measurement locations. The engine was stationary and the exhaust valves opened to provide extraction of the vapour by the cyclone filtration system. In Figure 6.9, the digital camera was utilised to provide a series of images over consecutive cycles. In this case, the spray was injected at 290 CA and clearly shows impingement upon the piston crown and reverse flow from the edge of the bowl. At this beam location, the PDA would record post-impingement droplets deflected from the piston crown that collide with the injected flow. The velocity direction would be opposed to that of the main spray.
A second elongated optical cylinder head was available for the photographic studies. These were performed in the ‘Three-windowed’ cylinder head as shown in Figure 6.10. The design of the cylinder head allowed optical access to the pent-roof combustion chamber through two opposed windows. The two parallel windows were mounted in the front and rear of the cylinder head. A third window was positioned in place of one of the exhaust ports.

The inside surface of the window formed part of the pent-roof chamber. The location of this window uniquely permitted a view of the piston crown, intake valves and the injector hole in the cylinder head.

6.3.3. Definition of Measurement Locations.
The PDA measurement locations were replicated from the ‘Annulus’ LDA measurements presented in Chapter 5. These are shown in Figure 6.11. superimposed upon a photograph of the engine cylinder head, annulus and injected spray. Under static engine conditions and at 10 mm below the gas face, the approximate spray width across the bore is from mid-cylinder (0 mm displacement) to -22 mm from the axis towards the intake side of the chamber. The results recorded at z=-10 mm are presented over the radial range of +5 mm to -35 mm about the vertical cylinder axis. This represented the maximum attainable range due to clipping of the beam pair by the cylinder head castings. At z=-9 mm and z=-14 mm, the obstruction of the beam pair with fixed separation did not permit measurements to be made across the same range.

6.3.4. Experimental Method: PDA Measurements
The experimental determination of the best operating settings for the PDA was carried out as described previously for the nebuliser and spray chamber measurements. In addition, an air-cooled mechanical shutter and phase-locked shutter driver/timing module (UniBlitz, model SD-1000 Shutter Drive/Timer) were utilised to attenuate the LASER beams when the piston intersected with the probe volume. Preliminary tests revealed that the PMT overload protection resulted in large sets of spurious data generated by optical flare and background reflections. These were manifested as zero droplet diameters with spurious velocities spread across the entire measurement range. The PDA PMT voltage settings and balance were then determined at each measurement location in-situ. This produced a PMT working voltage range that was chosen such that measured droplet diameter distribution did not alter across the range. Generally, the PMT operating voltage for the motored engine studies was of the order of 100 to 200 volts less than that utilised in the spray chamber to avoid optical flare effects and signal saturation. However, this resulted in reduced data rates and longer test runs. In part, the reduced data rates were also due to the optical vignetting of the PDA collection aperture due to the 20 mm height of the annulus.

Injection conditions were chosen to match fired engine studies undertaken by Ricardo Consulting Engineers using a similar non-optical engine. The homogeneous charge, fired engine conditions were for WOT and MBT timing at 1500 rpm and 2.5 bar IMEP. The injection duration was 61 CA with a SOI angle varied from 0 to 180 CA ATDC non-firing. For stratified operation, injection was much later in the cycle. At 1500 rpm, the SOI was in the range of 280 to 301 CA for a duration of 17 CA. For stratified operation, this was equivalent to an injection duration of 1.889 ms at 1500 rpm, which corresponds to a measured mass flow rate of 8.18 mg/inj from Figure 6.1.
For the static engine tests, the TTL injector trigger was used to mark and reset the arrival time clock in the Doppler Processor. For the motored tests, the engine timing and cycle resolver was used to send a marker trigger at TDC NF (0 CA) in the four-stroke cycle. The encoder input on the Doppler processor was set to count and reset for each marker signal. In this way, time from SOI to first Doppler measurement across all the measurement locations gave an indication of the spray tip velocity in the radial direction.

6.3.5. Experimental Method: High-Speed Photography

The two EPSRC loan pool, high-speed photographic camera systems were described in detail in Chapter 3. The individual merits of each camera system were utilised to study particular characteristics of the spray structure and airflow interaction. The Kodak system has a slower frame rate and reduced resolution but is capable of transferring many consecutive images to the digital storage buffer. In contrast, the Imacon 468 camera can capture up to only eight images at a time, but the image resolution is much higher and the inter-frame time can be as short as 10 ns. It was therefore necessary to take consecutive photo series with the Imacon camera to capture the entire injection event at the higher frame rates. Both cameras captured images of the fuel injection events using the 'Three-windowed' cylinder head and a combination of LASER light sheet illumination and halogen flash and halogen continuous, diffuse backlighting.

The Kodak and Imacon cameras were initially set-up to view the injection event through the plane window mounted in the front of the cylinder head. The Imacon camera was also positioned above the exhaust port window. This is shown in Figure 6.12, along with the Spectra-Physics Argon Ion LASER used as a light source for the light sheet experiments. A set of cylindrical lenses were utilised to generate a thin light sheet of approximately 0.5 mm in thickness. The alignment of the light sheet, focus of the cameras and flash trigger and camera/injection trigger tests were achieved by the use of a spark plug mounted target plate. The small target comprised of a copy of a spark plug body with a thin, narrow plate attached in the position of the electrode. The plate was inserted through the spark plug hole and rotated until it was parallel to the tumble plane of the engine. The plate was machined with a square slot of precise dimensions that allowed calibration of the final image dimensions. In addition, the plate was angled at 8° to the vertical to ensure that the final image of the target edges through the camera were parallel to the bore of the cylinder liner. The camera systems were then focused onto the slot in the target plate as shown in Figure 6.13. The light sheet was aligned at low LASER energy by observing the reflection of the beam against the thin edge of the target. The camera trigger, injection timing, spray location and instrument settings were verified by capturing a series of test shots with the target in place. A single sequence of shots is shown in Figure 6.14. In the exhaust port study, the LASER light sheet was rotated into plane of the injector axis at 36° to the horizontal plane of the gas face. This was achieved by rotating the cylindrical lens through 54°. The subsequent illumination allowed the camera to capture images through the exhaust port window and in the direction of spray propagation.

The backlit photography was performed by using halogen lamps positioned behind the rear cylinder head window. For the Kodak study, two relatively low power spot lamps were focused onto the window that was masked with 1 mm graph paper.
The lamps were positioned in such a way as to project a homogeneous distribution of light across the paper as shown in Figure 6.15. The Imacon study utilised the two, high power halogen flash lamps integrated with the Imacon triggering system. The lamps were positioned in the same manner behind the rear window. The timing of the flash lamps was input by the user in the software and was temporally positioned to ensure that each of the captured images had equal brightness and contrast. This required the independent control (timing, exposure, gain etc.) of each of the eight CCD units within Imacon unit. The set-up tests were performed in the static engine. For both camera systems, it was found beneficial to use some additional front lighting. Both camera systems were triggered to record image capture by the injector TTL rising edge pulse. The engine speeds and injection timings were duplicated from the PDA droplet and LDA airflow measurements for 500, 1000 and 1500 rpm. Both early and late injection events were captured. The full set of engine test conditions and camera settings are presented in Appendices I and J. for the Kodak and Imacon cameras respectively. A subset of the total result set is presented in the following sections.
6.4. Experimental Results and Discussion

6.4.1. Spatial Distribution of Temporally-Averaged Characteristics of Instantaneous Velocity and Droplet Diameter.

The spatial distribution of the temporally-averaged (absolute) gasoline fuel spray measurements of velocity and droplet arithmetic mean diameter (AMD or D_{10}) are presented in Figure 6.16. for the three vertical, axial, displacements from the nozzle exit. The velocity measurements were taken using the LDA and the diameter distributions collected using the PDA. At 8 mm (30 orifice diameters downstream) from the nozzle, the velocity profile across the spray rises to a sharp peak either side of the central region of the jet and mean velocities of 50 to 60 ms^{-1} are observed. The droplet diameters are relatively small, with the largest diameters located either side of the nozzle axis. The same pattern is exhibited at the 12 and 20 mm planes, approximately 45 and 75 diameters downstream of the nozzle respectively. The results indicate that the greatest liquid mass flux is distributed in an annular ring that propagates along the axis of the spray plume. With increasing distance from the nozzle, the mean velocity distribution becomes more uniform and the mean velocities are reduced to 10 to 20 ms^{-1} at 20 mm from the nozzle. Nevertheless, the characteristic form of the mean velocity profile across the spray is comparable, with the low momentum, smaller diameter droplets concentrated about the injection nozzle axis.

An important result is presented graphically in Figures 6.17a and b. for the measured droplet arithmetic mean diameters. The range of the mean droplet diameters is between 6 and 20 μm. Both plots illustrate the increase in measured mean droplet diameter between the 8 mm and 12 mm planes and a reduction at 20 mm. The magnitudes of the mean droplet diameters are approximately comparable in the cone wall regions between the 8 mm and 20 mm planes. In the same region of the spray cone, the diameters are seen to increase by in excess of 50% between the 8 and 12 mm cases. This is contrary to the temporally-averaged absolute mean diameter results published by some other research groups, at similar locations in pressure-swirl atomiser sprays, such as Park et al., (1998).

These groups have reported that rapid break-up and atomisation of the liquid occurs after a short distance from the nozzle and that the mean droplet diameters then decrease with increasing displacement from the orifice. The low mean droplet diameters observed at 8 mm from the nozzle were investigated by examining the spatial distribution of the mean data rates and signal validation percentages obtained for the PDA and LDA measurements as shown in Figures 6.18a and b. and Figures 6.19a, b, c, d and e. respectively. The mean validated data rates in both the LDA and PDA measurements show the greatest magnitude at 20 mm from the nozzle. In the PDA results, the data rate is at a maximum in all planes at a point approximately along the nozzle axis. The mean data rate drops away towards the periphery of the spray. The data rates for the 8 and 20 mm planes show a comparable peak magnitude whereas the peak value along the nozzle axis for the 12 mm plane is only approximately 25% of these values.

The width of the spray at the three measurement locations can be defined at the radial position where the PDA mean data rate tends to a minimum. Although the choice of an acceptable mean data rate is considered ambiguous, the validation of measured droplets can be used to give an approximation to the steady state spray periphery width over the total injection event.
These were approximately 12 mm at z=-8 mm, 16 mm at z=-12 mm and 30mm at z=-20 mm. The mean data rates obtained by the LDA measurements of droplet velocity show again that the 20 mm plane yielded the highest rates and greatest degree of fluctuation. The periphery of the spray yielded higher rates than the core. Outside of the periphery, the data rates were seen to sharply decrease. The mean validated data rates at 8 and 12 mm from the nozzle were very low in comparison, with the 8mm plane marginally higher in the core of the spray. The results would suggest that the droplet concentration within these planes was high enough to produce a poor signal to noise ratio and high measurement rejection rates. It should be noted that SNR validation is required for velocity validation and subsequent diameter validation.

The presentation of validation data to describe the differences between measurement planes close to the nozzle in automotive fuel sprays is rarely described in the literature or is generally reserved for high-pressure, Diesel sprays. In port-injection gasoline studies, the size and velocity validation criteria have been used to compare the proportion of spherical to non-spherical drops in the total drop population (e.g. Vannobel, (1996)). Figures 6.19a, b, and c. show the comparison between the droplet mean velocities measured with the LDA and PDA and the velocity and size percentage validation achieved using the PDA system at 8, 12 and 20 mm from the nozzle. In this way, the observed effects of all validation criteria are presented. The velocity percentage validation rate using LDA is shown in Figure 6.19d. for reference purposes. At 8 mm from the nozzle, the LDA mean velocity profiles across the spray plume are of the order of four times greater in magnitude than those measured using PDA. The difference in magnitudes measured reduces with increasing distance from the nozzle. At all three planes, the forms of the profiles are similar between LDA and PDA and characteristic twin peaks are observed. At 8 mm the first LDA velocity peak is measured at approximately 1 mm from the nozzle axis and the second opposing, smaller, peak, at approximately 3 mm from the nozzle axis. This gives an approximation to the steady state, hollow cone ring diameter, measured from the injection axis of 4 mm. Following the same order, at 12 and 20 mm, the second peak is larger than the first. The change in distribution of the peaks is caused by the asymmetry of the spray induced by the rotational motion of the liquid fuel as it exits the swirl injector and passes through the measurement plane. At 12 mm from the nozzle, the first peak is observed at approximately 3 mm from the axis and the second, at 4 mm. The hollow cone ring diameter is then approximately 7 mm. At 20 mm, the peaks are approximately symmetrically placed at 4 mm on either side of the axis. This would suggest a ring diameter of approximately 8 mm. These results can be used to determine a quasi-steady state inner cone angle for the fuel spray as illustrated in Figure 6.19e. The outer bounds of the spray were determined using the PDA validation rates as described in the previous section. The inner cone angle is estimated at 28° when determined at the 8 mm depth. The central cone value is however of the order of 50°. The cone angle evolves with injection and is dependent upon the temporal and spatial reference. It is interesting to note, that at 20 mm edge locations, the PDA measured negative velocities at the periphery of the spray of up to 0.25 ms⁻¹. This reveals the generation of a vortical structure along the spray periphery at this depth. These structures are not however evident in the mean LDA data due to the method of temporal averaging and the high peak velocities recorded.
In contrast to the velocity profiles, the PDA spherical and velocity percentage validation rates, defined as the ratio of ‘attempted’ to validated signals measured, are highest at the periphery of the spray and sharply decrease towards the axis in the 8 mm plane. However at 12 and 20 mm from the nozzle exit, the lowest spherical data validation rates are observed at the same locations as the peak velocities and largest droplet diameters. The central core of the spray shows an increase in percentage spherical validation but a marked decrease in velocity validation using the PDA.

The results suggest that the densest regions of the spray are close to the nozzle. In addition, they confirm the presence of the dense region in the spray, named above as the hollow cone ring. In these regions, the concentration of droplets must not exceed the maximum particle concentration for the instrument. For the Dantec Classic PDA 57x10 collection optic, the maximum particle concentration was calculated to be $1\times10^9$ particles per cm$^3$ using the formula provided in the Sizeware software based upon the calculated mean droplet diameter at that point.

The temporally-averaged droplet mean velocities and turbulence intensity distributions across the spray radius for each of the displacement locations as measured with the LDA are shown in Figures 6.20a, b and c. The mean droplet velocity profile is as described previously. In Figure 6.20b, the decrease in mean droplet velocity is plotted in the spray direction for the different radial locations. The central regions about the spray core exhibit the highest levels of droplet deceleration observed over the total injection event. The gradient of the curves are the steepest at the axis and ± 2 mm either side Those points located towards the periphery of the spray exhibit lower levels of velocity retardation and the relationship between these three planes becomes closer to linear than those at the core. These results also highlight the asymmetry of the spray properties. At each location, two velocity peaks are observed on either side of the injection axis in the jet regions of the cone structure. The peaks correspond to two local minima in the turbulence intensities at 8 and 12 mm (and a slight indication at 20 mm) from the nozzle as shown Figure 6.20c. Either side of the troughs, the turbulence intensity increases sharply towards the turbulent edges of the spray periphery between the liquid/air interface.

However, there is no increase at 20 mm from the nozzle in these regions. The central region of the cone is marked by an important increase in turbulence intensity of approximately 50% of the minimum value in each case at 8, 12 and 20 mm locations. Excepting, the 20 mm plane, these central peak values generally do not attain the same magnitude as those observed towards edges. These results suggest that a more ordered jet flow is found in the cone wall regions of the spray and that central core and edges are subjected to an increased level of velocity fluctuation.

The profile of the turbulent intensity distribution across the spray cannot be explained by the validation data rates presented previously in Figure 6.19d. In LDA and PDA systems, dense regions of sprays yield low data rates but high PMT activity. This is often manifested as PMT overloading, high range and consistency errors, spurious velocity values and the reported turbulence intensity increases dramatically in error.

The three-dimensional plot shown in Figure 6.21. shows the radial and axial distribution of the droplet arithmetic mean diameter across the injection plane investigated under quasi-steady state conditions. The interpolated surface plot was generated using a fine mesh superimposed upon the measurement locations.
The method of interpolation between the measured adjacent mean droplet diameters projected onto the mesh was as described in Chapter 4. The contour plot clearly illustrates the migration of the larger diameter droplets to form a cone region, asymmetrically placed about the injector nozzle axis as measured in this particular plane. The arrows indicate an approximate, hollow cone, steady-state, spray angle for this injector fuel spray.

6.4.2. Mass Flux Estimates

An investigative study was performed to compare the liquid mass flux estimated by the PDA in each plane with that measured by the method described in Section 6.2.1. Two approaches were considered with the available PDA measurements. The first involved using the volume flux estimated by the PDA software based upon the cross-sectional area of the probe volume (normal to the spray axis) projected onto the collection optic. The selection of a correct, cross-sectional area is a topic of much current research and discussion.

In this case, however, the effective cross-sectional area used in the Dantec calculations is based upon a particle size and trajectory dependant detection area and based upon signals of a certain minimum burst length. The second approach used the measured mean droplet diameters from the PDA system to provide a size-dependent cross-sectional area. The droplet velocity chosen was that due to the 1-D, LDA measurements. The mass flux was then obtained in each plane by the summation of the flux at each measurement point.

In the size-dependent mass flux estimation, an assumption is made that all droplets have passed through at least one of the measurement points and that the main droplet velocity component is in the direction of the spray axis and that the rotational, tangential velocity component is comparatively small. This is a poor assumption in a swirling spray of this nature. In both approaches, the mass flux calculations are based upon the total validated data set. The results therefore do not include droplets rejected on non-spherical criteria that were most apparent in the denser regions of the spray. These oblate or prolate spheroids are most likely to be large in diameter and travelling at high velocities. These contain a significant proportion of the liquid mass and momentum. In addition, smaller droplets may often be rejected for scattering insufficient light intensity to validate the SNR criteria.

The earlier studies of research groups such as Hoffman et al., (1997) reported that mass flux estimates for a pressure-swirl atomiser using PDA measurements did not compare favourably with those taken using a spray patternator. Chen et al., (1996) using the same commercial PDA system estimated that the combined error in droplet diameter and the size-dependent cross-sectional area used to calculate the mass flux was of the order of ± 35%. In the two PDA approaches investigated in this study, the order of the differences in the mass flux estimations with that measured by the weighing method (and corroborated by several sources) suggested that this PDA system could not be used in anyway to provide an accurate mass flux estimate within this spray. In contrast to the above remarks, the measured mass fluxes from both approaches were at least an order of magnitude smaller than that obtained by the timed mass experiments, suggesting that the error was considerably greater than 35% for this PDA configuration and spray.
In a comprehensive review of spray diagnostics, Bachalo, (2000) highlights the error associated with the incorrect choice of projected sampling volume area. The paper states that for research applications, PDA flux data must always include biasing corrections to minimise this experimental uncertainty.

6.4.3. Comparison of Time-Resolved Droplet Characteristics in a Static Engine.

In the static engine measurements, the injected fuel spray transfers momentum to the surrounding air at rest within the combustion chamber. The acceleration of the surrounding air results in a local pressure drop around the plume and air flows towards the spray. For a pressure-swirl fuel spray, this effect is required to entrain air into centre of the spray, thus forming the hollow cone shape. In motored studies, early injection occurs under atmospheric pressure conditions. As the in-cylinder pressure is increased, as in the case of late injection where the motored pressure was measured to be in excess of 7 barg, the gas density and droplet drag force are increased. The injected droplets are decelerated at a greater rate and momentum transfer between the phases is reduced. The resulting spray structure shows less of the hollow cone characteristics of the early injection spray plume.

Under motored conditions, the airflow jet velocity below the valve seats into the chamber can attain peak values that are comparable and generally in the same direction as the fuel droplet velocities. In Chapter 5, the ensemble-averaged mean airflow velocity at 500 rpm, in the direction of the cylinder axis, at a typical early SOI timing of 60 CA were measured at 10 mm below the gas face and 20, 25 and 30 mm from the cylinder axis to be approximately 20 ms⁻¹. A local mean velocity maxima was observed at approximately 90 CA and a turbulence intensity of 200%. The highly, turbulent and re-circulating flows below the valve curtains act directly upon the uppermost surfaces of the plume. The location of the injector between and below the two intake valves means that for early injection, the fuel spray is subjected to two competing jet flows. The complex processes of momentum transfer during this period alters the shape of spray plume and the axis of injection from cycle-to-cycle. The effects become more pronounced with increasing engine speeds.

The interpretation of time-resolved PDA droplet diameter and velocity measurements of fuel sprays of this nature is complex and in most reported cases, requires the temporal decomposition of the data into specific phases of the injection event. The initial phase of fuel injection that originates from the early needle lift period imparts momentum to the gas phase resulting in a local acceleration surrounding the spray. Droplets originating later in the injection event experience less drag force and arrive behind those injected previously. These droplets will either coalesce with those travelling at a slower rate in front of them, collide and break-up or pass straight through. Droplet collision and coalescence results in a droplet being recorded by the PDA measurement whose temporal origin cannot be resolved.

The instantaneous droplet velocities and diameters measured by the PDA system through the optical annulus under static engine conditions are presented in Figures 6.22a to i for the in-cylinder locations from mid-cylinder (filename: D00s001) to the edge of the bore (filename: D35s001), described in Figure 6.11. at 10 mm below the gas face, in increments of 5 mm.
The positive velocity vector is parallel to the cylinder axis and in the direction away from the spark plug. The in-cylinder air pressure was approximately atmospheric. A low-pressure extraction was applied to the exhaust port with the valves opened by 2-3 mm to aid evacuation of the chamber. The results are presented over the time interval from SOI to 0.05 s after the rising edge of the injection pulse. Fuel injection was simulated for 10 skip injections at 1500 rpm and stratified charge conditions, with an injection duration of 17CA which equates to 1.89 ms fuel injection duration. The end of the injection pulse (EOI) is indicated on the plots. The piston position was close to BDC. At all measurement locations, the greatest density of measurements were recorded in the range of 0 ms (SOI trigger less the inherent electronic and electrical delay) to approximately 10 ms. Droplet velocities after this period tended towards zero and were considered as inter-injection droplet measurements; i.e. droplets from the tail of a previous injection suspended in the chamber prior to evacuation. The instantaneous droplet velocity profiles change considerably across the chamber from mid-cylinder to the cylinder bore. The photographic studies suggested that the mid-cylinder location, (D00S001) and those at the bore edge, (D35S001), plotted in Figures 6.22a, g and h, respectively were situated on the periphery of the spray and that the location at 20 mm from mid-cylinder, (D20S00), (Figure 6.22e), lies approximately along the spray injection axis, under quiescent air conditions. Simple geometric calculations based upon an injector inclination of 36° to the horizontal places the injector axis at 20.5 mm from the cylinder axis at 10 mm below the gas face. This assumes that the nozzle exit is recessed by 2 mm into the cylinder head.

The PDA results show that the data density at the outer locations, shortly after the end of injection, is less than that for all other locations along the measurement line. The maximum instantaneous droplet velocities were also amongst the lowest recorded. At the mid-cylinder location (furthest from the nozzle), droplets are recorded at approximately 3ms after SOI. At this point, a broad spread in droplet velocities and diameters is observed. The velocity-diameter, cross-correlation plots are presented in a following section. The same characteristics are observed at 5, 10 and 15 mm from the mid-cylinder axis. However, the magnitude of the early droplet velocities increases greatly as the measurement locations move towards the apparent spray axis. Little or no data is recorded in the period of 0 to 3 ms at these locations. At 20 mm from the mid-cylinder axis, however, data is recorded at approximately 0.5 ms after SOI. Instantaneous droplet velocities were measured in the range of 5 to 95 ms\(^{-1}\) with diameters between 2 to 70 \(\mu m\). The duration of this short ‘pulse’ of data was of the order of three tenths of a millisecond. This ‘pulse’ was then followed by a period void of data (estimated at 2 ms) until approximately 3 ms from SOI, when the main injection event occurred. The measurements at this point were repeated at least ten times and in each case the observed data void remained. One of the repeat tests is shown in Figure 6.22i for the same location. The scale is adjusted to show the ‘early pulse, data void and main injection event. In this plot, the scatter symbol size represents the individual droplet diameter. The plot reveals that the ‘early pulse’ or ‘initial slug’ of fuel delivered contains relatively few droplets in comparison with the main injection event.

The first phase starts with the injection of several droplets of approximately 30 \(\mu m\) in diameter that exit the nozzle with velocities in excess of 90 ms\(^{-1}\). These are followed by a series of much larger droplets with significant momentum.
In addition, there is a scatter of smaller droplets with velocities in the range of between 20 and 80 ms\(^{-1}\). From approximately 8 ms to 2.8 ms after the SOI trigger there is an absence of droplet measurements. From this point onwards, a large spread in droplet diameters and velocities is observed, with the larger droplets having the highest velocity.

There are other instances of these phenomena reported in the literature (e.g. Zhao et al., 1997). For example, Comer et al., (1998) also show similar ensemble-averaged velocity and diameter characteristics along the injection axis at 20 mm from the nozzle of a Simplex swirl injector with fuel pressure of 100 bar and an injection pulse of 1.5 ms. The experiments were performed under atmospheric pressure and temperature conditions. An initial 'slug' of mean droplet diameter of 10 \(\mu\)m and velocity of 50 to 60 ms\(^{-1}\) is observed at 0.75 ms after SOI. There then follows a data void until approximately 1.75 ms after SOI. In this study, the void observed in the transient data from 2 to 25 mm from the nozzle is interpreted as the formation and subsequent collapse of the hollow cone structure. These findings are corroborated by measurements performed at 50 bar, which do not exhibit a complete data void behind the leading edge pulse. At the reduced pressure, it is less likely that a fully developed hollow cone had been produced.

Gasoline fuel spray PDA experiments performed by Preussner et al., (1998) showed the same characteristic pulse in fuel delivery for a pressure-swirl injector. They observed an early injection phase, comprising large droplet diameters with relatively high velocities. It was concluded that these droplets originated in the primary phase of injection where a small amount of 'swirlless' fluid was injected. This was described as an initial period after SOI where the internal flow upstream of the orifice is unsteady and had not attained a swirling annular flow. The main velocity component is therefore axial and the injector is assumed to behave as a simple orifice.

At 25 mm from the mid-cylinder axis the measurement location has moved closer towards the nozzle. A similar 'early pulse' was measured with a similarly broad range of droplet diameters and velocities. However, at this location, the 'pulse' of data does not precede a void of data and droplets are present until the main injection event arrives at approximately 3 ms from SOI. At 30 mm from the mid-cylinder axis, a few individual droplets follow the same behaviour as that observed at 25 mm. The majority of droplets, however, arrive shortly after injection but with a velocity that is recorded in the negative direction. A maximum negative velocity component of approximately 8 ms\(^{-1}\) was recorded at EOI timing in the direction parallel to the axis of the cylinder, i.e. towards the cylinder head. At 35 mm from the axis, the instantaneous velocity characteristics after SOI are similar again to those observed at 30 mm with all instantaneous droplet velocities accelerated to a peak, negative velocity of 6 to 7 ms\(^{-1}\) at approximately EOI timing. At both the 30 and 35 mm locations, the mean droplet diameters and spread about that mean value are significantly lower than at all other regions in the measurement plane.

The mean droplet velocity characteristics at all the in-cylinder locations along one horizontal line are summarised in the Figure 6.23. This shows the ensemble-averaged mean droplet velocities at 10 mm below the gas face. The ensemble averaging was performed by dividing the period from 0.5 ms (post 'early pulse') to 50 ms into 50 equal time bins, each of 1 ms. The results are plotted for the initial period of between 0.5 and 15 ms from SOI. For clarity, every second symbol is plotted on the graph.
The 20 mm location exhibits the highest peak mean velocity component parallel to the cylinder axis. Both the 20 and 25 mm locations show rapid deceleration of the droplet velocities up to approximately 5 ms from SOI. Conversely, the 0, 5, 10 and 15 mm locations experience an increase in ensemble-averaged mean droplet velocities over the same period. The mean velocities at the 30 and 35 mm extremities of the spray exhibit an increase in negative velocity suggesting the presence of a vortex structure within the spray, close to the cylinder wall. In all cases, the mean droplet velocities tend towards zero (excepting the 10 mm location).

Pitcher et al., (1990b) and Pitcher and Wigley (1991a) studied the size and velocity distributions in a Diesel spray under ambient and high-pressure conditions using a Dantec LDA/PDA system. The injection pressure was up to 400 bar and the injection duration was 1.67 ms. Two conclusions were drawn from this study. The velocity profiles exhibited a wavy structure superimposed upon the main profile with four main peaks. Within these fluctuations, the droplet size distributions showed a broad range of droplet concentrations (arrival times) centred about these wavy peaks. Their study also highlighted a region behind the leading edge of the spray of approximately 2 CA in duration (for a simulated 500 rpm, ambient pressure test) that was void of droplet measurements. The data void became less pronounced with increasing distance from the nozzle and from the injection centreline. Under motored, pressurised, engine conditions, the spray leading edge was dramatically reduced and the data void lengthened close to the nozzle. The amplitude of the longitudinal wavy structures was suppressed by the elevated in-cylinder air density and turbulence. However, during this void, the instrument PMT activity was described as intense. Pitcher and Wigley (1991a) concluded that the period void of data could be attributed to the lack of discrete droplet parcels and that is was more consistent with the presence of a liquid jet. They stated that the dense spray region extended approximately 75 to 100 nozzle diameters downstream of the orifice of diameter, 270 μm. They do not however include an analysis of the data validation rates at these locations.

The characteristics measured in the series of static annulus tests carried out in this work can be attributed to a combination of effects; that is, physical phenomena related to the injection process and the spray formation and a consequence of the measurement limitations of the PDA system. These observations of the spray development are summarised in Figure 6.24, with respect to four radial PDA measurement locations, A, B, C and D. Position A is located on the injection axis and will therefore record droplet data that will be different to those locations further from the axis. Initially, the swirless fuel ‘slug’ from the injector sac volume is interpreted as the spray leading edge. The spray at this stage has a narrow cone and little or no data is recorded at locations B, C and D. In the second sketch, a liquid sheet is formed as the fuel gathers rotational momentum. The PDA detection criteria will reject signals at position A. Position B however may record relatively small amounts of discrete droplets along the asymmetric periphery of the sheet.

Primary break-up of the sheet occurs in the third stage followed by the transition to a hollow cone structure through interaction with the gas phase. At this point, all droplet velocities are recorded as positive in the direction of injection. By the end of this secondary break-up stage, PDA measurements are recorded at A, B, C and D. The mean velocity and droplet statistics will however be strongly correlated with their physical location due to collisions, coalescence and flow reversal.
The greatest data rate is likely to be recorded at location C, situated near the cone wall. The transition from position C to A occurs at location B, situated along the inner wall of the cone.

Positions A and B show individual droplets with relatively small diameters of opposing velocity direction due to air entrainment into the hollow cone. The hollow cone structure is achieved at approximately EOI timing. Secondary airflow into the cone entrains smaller drops adding thickness to the walls and begins to fill in the hollow cone. This process was termed 'spray contraction' by Preussner et al., (1998). At some point later, the top part of the cone, concentrated about the centre-line, reaches those droplets previously injected. The lower droplets have imparted momentum to the surrounding gas and thus those droplets at the top of the cone experience less drag force and are less readily decelerated. As both groups reach the same distance from the nozzle, a toroidal ring is formed around the lower part of the cone as shown in the final sketch. The ring size increases as the spray moves further from the orifice until finally, the droplets are dispersed. At this point, the mean droplet velocities at A, B, C and D tend to zero in the absence of gas motion and the tail of the injection comprises predominantly small droplets.

6.5. Fuel Spray Characteristics in a Motored Engine

6.5.1. Comparison of the Spatial Distribution of Mean Droplet Diameters during Static and Motored Engine Operation: Early and Late Fuel Injection Strategies

The distribution of droplet diameters for five in-cylinder measurement locations are presented in Figures 6.25a, b, c, d and e. for the mid-cylinder axis and for distances of 5, 10, 15 and 20mm from the axis. These positions were chosen to highlight the differences in droplet populations for locations considered to be within the hollow cone; those along the injection axis and those on the spray periphery nearest to the point of piston crown impingement. The position of each measure was chosen from photographic evidence and from the static analysis performed in the preceding section.

Probability density functions of the droplet populations under static and motored conditions are presented for 1μm diameter bins over the total measured range. For motored conditions, both early and late injection cases are included. At certain measurement locations the piston intersected the probe volume and measurements were not possible. In these cases, the start of injection timing was altered to allow measurements. The mean diameter is shown on the plots along with a table of diameter statistics.

At each of the measurement points, the droplet diameter distribution for the static tests is much broader than for those performed under motored conditions. The distributions are positively skewed towards the larger droplet diameters. However, in all but one instance, the mean droplet diameters were less than those observed under motored conditions. In contrast, the motored droplet distributions, excepting the early injections in Figures 6.25d and e., show less flatness and more symmetry about the mean values. The late injection cases show a trend for a small percentage of over-sized droplets that are in excess of 70 μm. These droplets do not constitute a significant proportion of the droplet population during early injection.
Under late injection, the mean droplet diameter is always greater than for the static cases and generally greater than for injection close to piston BDC. The two exceptions for early injection are illustrated in the Figure 6.25d and e. where insufficient data has been collected to generate statistically significant values.

These findings are similar to those reported by Yamauchi et al., (1996) who made PDA measurements in a gasoline, high-pressure, swirl injector at 70 bar injection pressure into ambient gas pressure and temperature conditions. They reported a 'coarse droplet phenomenon' within their results. Large droplets were observed along the injection axis 0.4 ms after SOI and then distributed at two radial points either side of the axis from 0.6 ms after SOI. These two separated regions were reported as being located in the walls of the hollow cone. In such studies, many research groups have utilised the Dv0.9 diameter value in preference to the AMD or SMD to highlight the significance of these 'coarser' droplets in the total liquid mass distribution. Glaspie et al., (1999), reported that the Dv0.9 diameter might illustrate those droplets that were of origin in the opening and closing phases of the injection and that the parameter was a measure of the 'degree of preparedness' of the injected fuel. In an extensive G-DI injector study, Glaspie et al., (1999) has, importantly, shown that for a narrow cone injector of approximately 25-30°, the Dv0.9 diameter was relatively less reduced with increasing injection pressure when compared with medium and wide cone types. They conclude that for narrow cone injectors of this type, the initial pulse of fuel made up of the larger diameter droplets has a greater relative effect upon the total spray diameter distribution. The review of Zhao et al., (1997) suggests that this parameter may provide a better correlation for uBHC emissions in direct injection engines.

The previous LDA air motion studies presented in Chapter 5 showed that the bulk in-cylinder air motion at piston intake BDC was relatively low compared to that during the latter stages of the compression stroke. It can therefore be surmised that the difference in mean droplet diameters observed between static, early and late injection scenarios is due to the dispersion of the smaller droplets by the gas phase. For late injection, only the larger droplets will possess sufficient momentum to penetrate the bulk air motion, at elevated pressures, to reach the probe volume. Indeed, the mean droplet diameter at 5 mm from the mid-cylinder axis (furthest from the injector) yielded the greatest mean droplet diameter. Those smaller diameter droplets are retarded, collide and coalesce and are transported away by the air motion. The early injection cases show a mean droplet diameter that is also generally greater than the static tests. This may in part be due to the dispersion of small droplets away from the probe volume by the turbulent gas flow exiting from behind the valve curtains and persisting until the final stages of the stroke. These droplet distributions will also then be biased towards the larger droplet diameters.

Figure 6.26a. and b., show a summary of the droplet diameter statistics for all locations and for each injection scenario. The two early injection cases described above have not been included in the linear fit and therefore this curve is a best fit of a limited set of points and must be treated as inconclusive. Both of the static and late injection cases show the same trend in mean diameter with increasing distance from the injector nozzle. The mean droplet diameters are seen to increase in an approximately linear manner to a local maximum at 5 mm from the cylinder axis.
It is likely that the reduction in droplet diameter beyond this point; i.e. from the mid-cylinder axis, is due to the interception at the probe volume of post-impinged droplets rebounding through splashing from the piston surface.

*Figure 6.26b.* shows the coefficient of variation in the diameter statistics across the chamber. This is equivalent to the ratio of the standard deviation to the mean droplet diameter and gives an indication of the ‘stability’ of the droplet distributions. For each of the cases, the coefficient of variation appears to be unrelated to in-cylinder position. The greatest value of 50% is observed for the static engine tests, whilst those performed at early and late, motored conditions are approximately equal at 40%. The results show that the ‘stability’ of the droplet distributions, whether along the injection axis or in the hollow parts of the cone, is approximately constant. The high percentage values however suggest that the repeatability from injection cycle-to-cycle is poor. This is in part due to the physical location of the fuel spray structures (core and cone) relative to the measurement locations. This relationship changes as the air motion and density changes act to retard or deflect the spray depending upon injection scenario. This is illustrated in *Figure 6.27.* for a hollow cone spray. Under static conditions, the injection axis is uninterrupted and the PDA measurements are a complex function of measurement position. The points encounter a wide range of spray conditions that vary with distance from the nozzle exit. These include droplet and velocity distributions along the injection axis, within the cone walls and along the spray periphery. The nearside of the fuel spray is bounded by the cylinder wall and a vortex is formed where smaller droplets are entrained within the local gas flow recirculation zones. As the engine is motored, the incoming gas flow deflects the spray downwards and strips away the smaller drops. The measurement points will therefore observe different features of the spray structure. In particular, the position of the jet flows in the hollow cone will be directed downwards. The variation in the stability of the distributions measured at the same locations for static and motored operation cannot be readily compared. They will differ as a function of position in the spray in addition to cycle-to-cycle fluctuations. However, it would be expected that the coefficient of variation between the early and late injection events differ greatly. The intake air charge is responsible for perturbing the upper edge of the fuel spray and inducing flapping instabilities in the cone structure. Under late injection conditions, the spray cone angle is much narrower and points upon the periphery of the spray are less likely to return valid data. These features are reported in the following section where high-speed photographic techniques have been utilised.

6.5.2. Comparison of the Spatial Distribution of the Droplet Velocity-Size Cross-Correlation during Static and Motored Engine Operation: Early and Late Fuel Injection Strategies

The effect of the fuel injection timing upon the droplet velocity-size cross-correlations for static and motored engine operation are shown in *Figures 6.28a, b, c, d, and e.* for the in-cylinder measurement locations described in the previous section. The static engine operation plots show for each of the in-cylinder locations, excepting 20 mm from the mid-cylinder axis, little or no significant correlation between the droplet sizes and droplet velocity. The scatter of instantaneous measures is distributed in a narrow, horizontal band positioned above the zero velocity level and extending across the droplet size range.
The shape of the distribution broadens in the velocity direction as the location is moved towards the injector axis. For the early injection cases, there is a significant difference in the correlation. Each of the plots clearly show a positive velocity-size correlation as the in-cylinder, intake generated gas flow accelerates the liquid phase.

Stanglmaier et al., (1998) reported the presence of the pre-slug from a hollow cone swirl injector using high-speed LLS photographic studies performed in an optical engine with a transparent cylinder liner. The slug was present in a static test and at 750 rpm. However, at 1500 rpm, evidence of the pre-slug was less pronounced as the intake airflow became more dominant. At the mid-cylinder location, a small proportion of the droplets show a negative velocity trend. The spread in droplet diameters is greatly reduced in comparison with the static test at the same position and with all the other motored tests. There are far fewer negative velocity droplets of size greater than 20 μm than shown within any of the other plots. These droplets are smaller in diameter and have a velocity component in the upward direction. The change in distribution is due to the presence of the strong tumble air vortex that has been established within the chamber. The air motion has the greatest effect upon the smaller droplets and disperses them in airflow where they are carried towards the cylinder head. It is also clear at this location, that the turbulent air motion has assisted the spray break-up process as the injected droplet diameter range is greatly reduced.

The late injection plots clearly show a retardation in velocity magnitudes and less of a spread in droplet diameter, although some of the larger droplets have still managed to ‘break-through’. At 5 and 10 mm from the cylinder axis, there does not appear to be a strong correlation between diameter and velocity. However, closer to the spray axis, at 15 and 20 mm, the correlation is positive with a steep gradient. The gradient is greatest at 20 mm and twice that of the early injection cases. Figures 6.28d and e. however show the incomplete data collected for the early injections at these points, which precludes them from any statistical analysis.

6.5.3. Temporal Evolution of Ensemble-Averaged Droplet Velocities and Diameters.

Vannobel (1996) summarises the contribution of the gas flowfield upon the fuel spray dynamics. Two terms are described; turbulence generation and droplet dispersion. The first relates to the generation of turbulence by the distortion of the gas flowfield due to the wakes of the particles passing through the flow. Droplets travelling within wakes may overtake leading drops, collide and coalesce. Droplet surface tension and viscous forces are overcome and droplet break-up is increased. The second mechanism relates to the dispersion of the droplets throughout the local flowfield according to their density. The fuel spray droplets are re-distributed due to differences in their momentum and drag forces and through interaction with the flowfield structures. The droplet-eddy interaction is complex and is related to the eddy size, motion and lifetime. The rate of droplet dispersion is then estimated from the droplet relaxation time or ‘eddy residence’ time. The dispersion rates are usually not constant due to the range of droplet and eddy size distributions. A more complete description of three currently proposed stochastic droplet dispersion models can be found in the experimental and numerical study of Chen et al., (1996).

In a PDA study of port injection, gasoline fuel sprays, Vannobel, (1996) showed how the relative liquid/air velocity influenced the droplet size distribution.
For relatively low, steady state, air volume flow rates, the AMD was seen to increase when compared to static air measurements. This was also observed in this work as reported in the previous section. Drop sizes in this regime were governed by the injection pressure and orifice size. With increasing flow rates, the AMD decreased and secondary atomisation through interaction with the flowfield was reported as becoming a more dominant mechanism in droplet sizing.

The analysis of the results in the previous sections showed a subset of the total droplet population that exhibited relatively high velocities and large diameters. These droplets play a significant role in the momentum exchange between the liquid and gas phases. A measure of the individual contribution of a droplet to the total liquid spray momentum is a product of the droplet velocity and volume where all droplet densities are equal and constant. An ensemble-averaged, volume-weighted, mean velocity was then calculated for the total number of droplet measures within one crank angle interval over many consecutive injections. These are plotted at three in-cylinder locations in Figures 6.29a, b, and c and for early and late injection conditions. These positions are chosen to illustrate the differences in the temporal spray development for injection at two early injections of 120 and 180 CA and two late injections of 280 and 290 CA. The number of droplet counts per crank angle interval is also plotted. The droplet data was not considered for counts of less than 50 per CA averaging window. A volume mean diameter ($D_{30}$) was then calculated from the ensemble-averaged, volume-weighted, mean velocity within the interval. These are plotted against crank angle. A linear curve fit was used as an approximation to the droplet scatter plots for only those points above the 50 counts confidence limit. In Figure 6.29b, the acceptable counts band was reduced to 30 counts per 1 CA interval due to the relatively small size of the total data set. This was in part due to the physical location of the measurement point during late fuel injection at 290 CA. At this instant, the piston comes close to the probe volume and the reflected light intensity is high. In addition, the spray structure (confirmed by the high-speed photographic studies) is very narrow and hence the probability of recording a discrete droplet at a more distant point is greatly reduced.

Figures 6.29a and b. show the instantaneous and volume-weighted, mean velocities and volume mean diameters for the in-cylinder radial location at 5 mm from the mid-cylinder axis towards the injector. The results show the comparison between SOI at 120 CA and 290 CA. At a SOI of 120 CA, both the volume-weighted, mean velocity and volume mean diameters show a positive linear increase over the confidence crank angle range. The greatest liquid momentum is observed at the tail of the injection. In this interval, the volume mean diameter increases by approximately 0.3 $\mu$m per CA. The ensemble-averaged, volume-weighted, mean droplet velocity increases at a rate of approximately 0.5 ms$^{-1}$ per CA. At a SOI of 290 CA, the in-cylinder gas pressure was approximately 8 bar absolute and the spread in individual droplet velocities is much reduced. There is no discernible increase in the volume-weighted characteristics. The trailing droplet volume mean diameters are however approximately 10 $\mu$m smaller in size.

In all plots, the ensemble-averaged, mean gas velocity obtained from the LDA study in Chapter 5 is included for comparison. It should be noted that these measurements were performed separately to the PDA measures and in the absence of injected fuel. They therefore serve to illustrate the characteristics of the gas phase prior to the injection event.
In Figure 6.29a., the ensemble-averaged mean airflow velocity component at SOI is approximately 18 ms\(^{-1}\) in the direction of the measured injected fuel velocity component. The injected droplets are retarded less by the gas phase due to their relative velocity component when compared to the late injection event. In Figure 6.29b., the smaller droplets are dispersed by the gas motion but the larger droplets are unperturbed and pass with little deviation to the probe volume. These measurements must therefore be considered as biased towards those droplets that are capable of attaining the probe volume location. Under the late injection conditions presented in Figure 6.29b., the gas velocity component opposes the measured fuel velocity component. At the SOI, the ensemble-averaged, mean air velocity is approximately \(-7\) ms\(^{-1}\) at this location and the ensemble-averaged, volume-weighted, mean droplet velocity is retarded by approximately 1 ms\(^{-1}\) in the crank angle interval, 306 to 312.2 CA. The absence of any change in the volume mean diameter within the interval highlights the difference in droplet break-up compared to the early injection case. It should be noted that at 318 CA, the probe volume was obscured by the piston topland.

In these cases, it is the relative air/droplet velocity that determines the droplet diameter distribution. For relatively large positive velocities, the droplet distribution is seen to increase in size. For large negative velocities, the droplet distributions are little changed. If the magnitude of the air velocity is large, the smaller droplets are more readily accelerated towards the probe volume, leaving the larger volume droplets in the tail of the injection. In Figure 6.29c., the SOI was advanced to 180 CA and measurements were recorded at 10 mm from the mid-cylinder axis and 10 mm below the cylinder gas face. At this point in the cycle, the in-cylinder air motion results from the previous chapter confirmed the presence of an established bulk, reverse tumble air motion. The measured velocity component at SOI was approximately 3.5 ms\(^{-1}\) in the downward direction. During the early compression stroke, the mean velocity direction is reversed with an ensemble-averaged mean value of approximately \(-5\) ms\(^{-1}\). The ensemble-averaged, volume-weighted, mean droplet velocity is reduced over the confidence range of crank angles at a rate of approximately 0.2 ms\(^{-1}\) per CA. The volume mean diameter remained approximately constant at 24 \(\mu\)m.

Figure 6.29d. shows measurements recorded for late injection (SOI 280 CA) for the in-cylinder location at 15 mm from the mid-cylinder axis. The momentum exchange between the phases is greatest at this location and under these conditions. The droplet ensemble-averaged, volume-weighted, velocity decreases at approximately 0.5 ms\(^{-1}\) per CA. The volume mean droplet diameters are approximately 3-4 \(\mu\)m smaller over the crank angle range than those recorded in the early injection cases. In addition, the variation in these mean diameters is small across the range; of the order of 0.1 \(\mu\)m per CA. At this location, a significant proportion of the smaller diameter droplets are recorded with a negative value velocity component. In contrast to the static PDA tests, the piston position is close to the cylinder head under motored injection timings. Smaller post-impinged droplets are recorded with a negative velocity of up to \(-15\) ms\(^{-1}\) as they splash from the piston surface. The droplet impingement process is governed by the droplet Weber number, the angle of incidence, the piston temperature and the air velocity and turbulence characteristics. In simpler terms, droplets impinging upon the piston surface will either ‘bounce’, ‘stick’ or ‘splash’.
Post-impinged droplets represent a subset of the total droplet population that can separated using the droplet arrival times or by some comparison with a static test measured at the same location in the spray. In the motored tests, the smaller droplets are deflected after impingement with the piston crown towards the spark plug region. They are assisted by a mean airflow velocity component of approximately -5ms⁻¹. The greatest droplet momentum transfer occurs shortly after SOI. This is in contrast to the local minima observed at a SOI of 120 CA in Figure 6.29a.

6.5.4. High-Speed Photography in a Motored Engine.

The high-speed photographic studies were used to provide insight into the spray geometry and location within the chamber and to observe the influence of the piston and air interaction under motoring engine operation. The LASER light-sheet (LLS) illumination technique produced a global view of the large-scale structures at the spray tip and along the spray periphery during the injection event. The backlit shadowgraphs revealed the internal structures of the plume and showed the high levels of cyclic variation due to the airflow interaction beneath the intake valves.

A selection of the results are presented that are specific to enhancing the description of the fuel spray characteristics observed by the PDA studies. In each case, still images were captured from high-speed sequences. Interpretation of the finer structures observed in the spray movies are therefore much more difficult to identify from a series of still images.


Figures 6.30. and 6.31. show a typical sequence of images recorded by the Imacon 468 camera. The images are read from top to bottom and left to right across the page. The individual image delays from the SOI trigger are recorded with each image. The timings were chosen to ensure that the piston crown impingement was captured. In both the early and late injection cases, the fuel delivery pulse width was kept constant at 20 CA. This is equivalent to a fuel injection duration of 6.7 ms at 500 rpm. In Figure 6.30. early injection, occurs with the intake valves open and the piston crown in view for SOI's of 40 and 50 CA. The fuel spray rapidly forms a broad cone structure with a hollow cone and a sinuous, flapping periphery. The upper edge of the plume is more distorted due to the intake air interaction. At SOI's of 40 and 50 CA, the spray impinges upon piston crown shortly after the onset of fuel injection. In the full-sized images, some fuel is observed exiting the piston bowl on the exhaust side of the chamber. For later injection timings, the fuel spray ‘chases’ the piston downwards. The results suggest that an optimum SOI timing for this configuration chosen to best avoid piston wetting and significant levels of uBHC in the exhaust, yet provide adequate time for evaporation and homogeneous mixing, would be from 60 CA. Under cold, motored conditions, these results are best representative of cold, start transient engine operation.

In Figure 6.31., fuel is injected late into the compression stroke. The start of injection timing (piston position) and the time taken to impingement will govern the mixture stratification (preparation time) and the transport of the fuel to the spark plug. The images show a much narrower injection plume and evidence of a small spray tip vortex. As the SOI is retarded, the fuel impingement upon the piston crown occurs earlier.
At SOI 290 CA, impingement occurs approximately between 0.65 and 0.70 ms following SOI. At 300 and 310 CA, this occurs between 0.55 and 0.60 ms. At 320 CA the spray impinges upon the piston at approximately 0.50 ms after SOI.

The contrast in spray structure between early and late injection scenarios can also be observed through the exhaust port window. Figure 6.32a and b. show the differences in fuel spray behaviour at 500 and 1000 rpm respectively for the initial injection phase from 0.4 ms after SOI to 0.75 ms in equal time steps of 0.05 ms. In this plane, the early injected fuel spray attains a broad cone shape that extends outwards close to the raised intake valves. During late injection, the liquid jet is narrow and sinuous in appearance with clear distortions along the periphery. The difference in spray geometry extends up to the EOI phase as illustrated in Figures 6.32c. and d. at 1000 and 1500 rpm respectively for time steps of 0.05 ms from 2.35 ms to 2.70 ms after SOI. In the late injection cases, the piston crown with bowl geometry moves into the image.

The magnified photographic images recorded through the exhaust window showed the asymmetry in the spray structure during the early, transient phase of fuel injection. The developing spray asymmetry in the exhaust window plane is shown in Figure 6.33. for a late fuel injection strategy at 1000 rpm. The images were recorded at 0.5 ms after SOI for two identical test runs. A developing conical sheet that penetrates in a spiralling manner follows the initial slug of swirlless fuel. The maximum spray penetration at this point is approximately along the fuel injector axis, coincident with the mid-cylinder plane and the PDA measurement grid. The photographic evidence of asymmetry under such conditions corroborates the conclusions presented in the ambient chamber and motored PDA spray studies.

The fuel spray impingement upon the cylinder wall is shown in Figures 6.34a, b and c. for a static engine test simulating early injection at 9000 fps using LLS illumination and the optical annulus engine build with the Kodak 4540 camera. The camera was triggered using the SOI signal. The piston was positioned near to BDC to allow observation of the spray across the complete chamber for an injector angle of 36°. The engine cylinder liner was heated to 80 °C but the quartz annulus had no external heating. The image sequences are read from left to right and top to bottom. In Figure 6.34a. a pointed jet of fuel is seen to exit from the injector in the third image. By the end of the first sequence, a spray cone is developed. In Figure 6.34b. the injected fuel is seen to impinge upon the cylinder wall and a wall-bounded vortex is formed in the lower right-hand corner of the later images. In the final series of images presented in Figure 6.34c., the injection period finishes and the spray collapses. However, in the absence of any bulk air motion or high cylinder wall temperatures, there exists a significant amount of liquid fuel deposited upon the exhaust side of the chamber well after EOI. The same test was performed under motored operation at 500 rpm for a SOI at 60CA for 20 CA. Figure 6.35. shows that at low engine speeds, during the mid-injection period, the spray is deflected across the surfaces of the piston. A significant proportion of the fuel is carried over towards the exhaust side of the chamber as observed in the static tests. At 1500 rpm however, the spray impinges within the piston bowl but does not appear to be carried over to the exhaust side as shown in Figure 6.36.
6.5.4.2. Early and Late Fuel Injection: Effect of Engine Speed upon Spray Structure

The spray structure and location within the cylinder chamber is shown in Figures 6.37a, b, c and d. for early injection conditions at 500, 1000 and 1500 rpm. The Imacon image series are read in the same manner as described previously. The image series was chosen such that the total fuel injection event is separated into four periods. The first period is from 0.4 ms from SOI to 0.7 ms in equal time steps of 0.05 ms. Period 2 ranges from 0.75 ms to 2.25 ms after SOI in steps of 0.25 ms. The third period is from 2.75 ms to 5.75 ms in steps of 0.5 ms. The final period was captured over the interval of 6.5 ms to 11 ms after SOI in time steps of 0.75 ms.

In each case, the exposures were matched to the temporal delays an maintained constant at each engine speed. The fuel injection scheduling was scaled with engine speed based upon a stoichometric fuelling of 61 CA pulse duration at 100 bar and 1500 rpm with WOT. The SOI trigger was at 60 CA in each case. In the first period of fuel injection, the liquid is observed to leave the injector as a narrow jet. The effect is most pronounced at 1500 rpm, where the intake jet flows around the intake valves strip away the smaller droplets that are present at the lower engine speeds. At 500 rpm, the cone shape starts to form earliest and impingement upon the piston surface occurs. By period 2, a cone had developed at all engine speeds. The hollow cone form was most prevalent at 1500 rpm. This is clearly observed as two liquid cords issuing from the injector. This is shown in Figure 6.38. for a sequence captured at 1500 rpm using the Kodak camera at 18000 fps during the initial phase of early injection. An approximate inner cone angle under these conditions was estimated at 10°.

At 1500 rpm, the upper portion of the fuel spray appears distressed in Figure 6.37d. and the spray injection angle is steeper in comparison with the lower engine speeds. By period 3, a hollow-like cone is observed at all engine speeds. Again, the 1500 rpm image series is more influenced by the incoming air charge as peak valve lift is approached. In certain images, the cone structure is momentarily detached and flapping of the plume is observed as the spray is deflected downwards. The fourth period covers the end of injection and highlights the difference in spray angles due to increasing engine speed. The 500 rpm series shows the spray to be much denser than those recorded at the higher speeds.

The change in spray angle associated with increasing volume flow rate through the intake valves was simplistically analysed using a greyscale threshold technique. Individual images captured at the same instance in time at each engine speed were extracted from the image series during mid-injection. A bisecting line was placed upon the image such that the spray plume either side of the line was approximately symmetrical. This was verified by plotting the intensity of the greyscale image along a line through the spray that was perpendicular to the bisector. The angle between the bisector and horizontal window reference was taken as the spray injection angle. A static engine injection test was conducted to ensure that the method yielded the geometric injection angle of 36°. The results are plotted for static, 500, 1000 and 1500 rpm engine tests in Figure 6.39. The images show that as the engine speed is increased, the spray injection angle is increased. The angles can only be approximated by this method, but they serve to highlight the significant increase in spray angle between 500 and 1500 rpm of between 38° and 45° at these relatively low engine speeds.
These results show one of the effects of the intake airflow upon a side-entry injection system. At higher engine speeds, the fuel spray is directed down the chamber after the piston and increased homogeneity of the charge is achieved with less wall-wetting. In Figures 6.40a, b, and c, the effect of engine speed upon late injection is presented in the same manner. The shorter injection event is divided into three periods. Period 1 ranges from 0.4 ms after SOI to 0.75 ms in steps of 0.05 ms. Period 2 occurs during mid-injection from 1.55 ms to 1.9 ms in steps of 0.05 ms. The final period is from 2.75 ms after SOI to 3.1 ms in equal steps of 0.05 ms. Once again, the fuel injection scheduling was matched to the engine speed. In each case the SOI was at 301 CA. In contrast to the early injection series, all the images show that the injected fuel spray takes the form of a narrow, solid jet. At all of the engine speeds, the form of the spray remains similar in the first period. Throughout the second period, the 500 rpm spray has a larger cone angle than those at the higher engine speeds and in-cylinder pressures. In all cases, the spray injection angle does not appear to be altered.

6.6. Assessment of Errors

6.6.1. Experimental Uncertainty

The sources of error and experimental uncertainties associated within LDA and PDA measurements in engines were covered extensively in Chapter 3. The statistical accuracy of the mean and RMS velocities were estimated in Chapters 4 and 5 for the in-cylinder air flow measurements using the standard error for a sample of a population. In PDA measurements, an additional error is attributed to non-linearity's in the droplet size and phase relationship. These can be minimised by careful alignment of the transmission and receiving optics and measurement of the optical parameters. Pitcher et al., (1990b) state that errors of less than 1% in the scattering geometry were achievable by following good experimental practice. They suggested that signal processing errors were more significant at low SNR's measured over the full electronic bandwidth. Nevertheless, they estimated errors in individual velocity and phase measurements of less than 3% and 5% respectively.

The greatest source of error is therefore due to the correct interpretation of the mean values from a limited data set. For the droplet diameter distributions, data was collected where possible until all droplets were within ± 3 standard deviations of the mean value. These plots have been shown in the previous section. In the cases where this was not possible, due to nozzle or piston proximity or within regions of high optical glare, a reference has been made in the text. Also, some of the diameter distributions show large droplets generated in the initial injection phase that are not within these limits. These droplets diameters are however included due to their importance in gaining knowledge of the spray process.

The validity of the PDA measurement procedure was assessed by various methods, which have been discussed in the previous sections. In addition, the repeatability of the process and the standard error in the motored, mean gasoline droplet velocities were analysed.
6.6.1.1. Repeatability of Measurements in the Spray Chamber

The repeatability of the droplet size measurements and traverse positioning performed in the ambient spray chamber were assessed at several locations within the spray plume. Figures 6.41a and b. show the frequency plots for two measurement locations in the near nozzle region at an axial distance of 8 mm from the nozzle orifice. These locations were chosen due to the tendency for PMT overload and low data validation rates in this plane about the injector axis. The two points were located at 3 and 4 mm from the injector orifice axis and exhibited approximately the same data validation rate. In each case, 15,000 validated data points were collected. The tendency for the PMT to overload in this region is highlighted by the asymmetry in droplet frequency distribution. Both of the populations are skewed towards the smaller droplet sizes and there are a considerably large number of 'zero' diameter readings which suggests spurious signals were gathered from PMT overload. However, both locations show good correlation between the independent measurements despite the observed difficulties at these locations and the biasing of the droplet sizes.

6.6.1.2. Standard Error In the Mean Droplet Velocity: Motored Engine Conditions.

The standard error, $\epsilon$, in the fuel spray mean droplet velocity per crank angle interval under motored conditions was plotted for three in-cylinder locations and for both early and late injection conditions. These are presented in Figures 6.42a, b, c and d. as error bars superimposed upon the ensemble-averaged mean values against engine crank angle. The standard error was calculated from the ensemble-averaged mean velocity and not the volume-weighted, ensemble-averaged mean velocity. The error bars chosen here represent $\pm 1$ standard error about the mean value. These error margins are significantly narrower than the $\pm 3\epsilon$ bars presented in Chapter 4 for the LDA airflow measurements. As such, the probability of the true mean droplet velocity value being within these error bands is approximately 67%. The high statistical uncertainty is not wholly due to a low data density within each crank angle window as commented previously. It is instead a product of the fuel spray/air and piston interactions and the method of temporal averaging of the data. During a fixed-point measurement technique employed in this manner, the output data relates to a proportion of fuel droplets of indeterminable temporal (and spatial) origin. If the proportion of these droplets becomes significant, then large fluctuations within the mean velocity profiles are observed despite the relatively large data sets collected. The proportion of these droplets will vary from location to location and with the type of injection scenario imposed. In the literature reviewed in Chapter 4, experimental values for $\pm 3\epsilon$ standard errors are quoted for seeded LDA airflow studies without fuel injection. However, none of the studies reviewed have approached the analysis of standard errors within crank angle intervals for fuel spray droplets directly or indirectly injected into a motored engine.

The measurement position at 5 mm from the mid-cylinder axis during early injection (Figure 6.42a.) shows a lower standard error than that of the late injection case at 290 CA in Figure 6.42b. In the early injection case, the standard error across the complete recording window is approximately within the range of $\pm 1.0$-$1.5$ ms$^{-1}$. Under late injection conditions, the data density is reduced, piston crown impingement occurs and the standard error is increased to $\pm 2$ms$^{-1}$ following the end of injection.
In the advanced injection case presented in Figure 6.42c., with SOI at 180 CA, the standard error is also increased when compared to the 120 CA case. The maximum standard error is approximately $\pm 4$ ms$^{-1}$. At 15 mm from the mid-cylinder axis, the standard error presented in Figure 6.42d. is also of the order of $\pm 3.0 - 4.0$ ms$^{-1}$. for a late SOI of 280 CA. The increase within these cases is most likely due to two particular effects. The first relates to the nature of the in-cylinder airflow and the second; to the proximity of the piston crown surface to the measurement location during late injection. For a SOI of 180, 280 and 290 CA, the measured in-cylinder airflow component prior to injection opposes that of the fuel spray injection. This is greatest at a SOI of 180 CA.

The fuel spray and air interaction is therefore most turbulent and the measured RMS fluctuations in the fuel droplet ensemble-averaged, mean velocity reach a maximum. In the late injection cases, the location of the measurement probe relative to the instantaneous piston position (injector-piston-stroke geometry) relates to the amount of injected fuel that is 'splashed' backwards. This is observed to a greater extent for fuel injection at 280 CA and 15 mm from the mid-cylinder axis towards the injector side. It is therefore expected that measurements within these locations would result in a greater degree of statistical uncertainty.


A detailed experimental study of a high-pressure, gasoline injector for a direct injection engine was undertaken. PDA and High-speed photographic techniques were utilised to describe the fuel spray under quasi-steady and dynamic flow conditions, within a special fuel chamber and within the stationary and motoring engine. The injector nozzle orifice and fuel mass flow rates were measured.

A series of different types of measurements were performed using water and fuel sprays, to establish the best operating parameters for a PDA system and to explore the inherent link between the technique and the interpretation and analysis of the measured data. It was concluded that the velocity measurements were best performed using the LDA and the droplet diameter, using PDA, excepting the size-velocity correlations. A study of the effect of the probe volume dimensions concluded that the 32/160 lens was the best compromise, based upon these findings and a knowledge of the expected droplet diameter distributions found in the literature. The fuel spray was then measured under three conditions.

The first experiments were performed in an ambient temperature and pressure, quiescent chamber. These series of tests established the fuel spray droplet diameter and spraywise velocity component over many injections. A dense spray region was shown to exist from the injector tip to approximately 8 mm along the spray axis. Measurements were therefore performed at 30, 45 and 75 nozzle diameters downstream of the nozzle. These locations were representative of values that could be viewed in the combustion chamber of the engine through the optical annulus.

The measurements across the fuel spray showed an asymmetry in the droplet velocity profile about the nozzle axis. The velocity profile revealed two sharp peaks positioned either side of the injection axis. The magnitudes of the peaks were not equal and the largest droplet diameters (and greatest droplet numbers) were observed in these two regions.
The droplet AMD was measured in the range of 6 to 20 μm. The asymmetric shape and droplet and velocity distribution characteristics were continued at increasing distances from the nozzle. However, the distribution of the magnitude of the two peaks was inversed between 12 and 20 mm. The change in distribution was due to the asymmetry in the spray induced by the rotational motion of the liquid jet exiting the swirl injector. Between the 8, 12 to 20 mm planes from the nozzle, the velocity magnitudes were reduced. The mean droplet diameters however were seen to increase from 8 mm to 12 mm and then decrease between 12 and 20 mm. The increase in droplet diameter was due to the low validation rate of measured data in the 8 mm plane using the current spherical and signal validation criteria. As a result, the measured droplet diameter distributions did not contain those larger diameter, high velocity, droplets rejected for non-sphericity. The subsequent statistical populations were skewed towards the smaller droplets. The highest data rates were recorded in each plane along the injector axis. The highest of all mean data rates was recorded at 20 mm from the nozzle. A quasi-steady state spray width was then determined using the mean, validated PDA data rate at each measurement plane. Negative velocity values were recorded along the periphery of the spray indicating the generation of re-circulating vortices between the gas and liquid phases. The LDA velocity distribution peaks were used to estimate an approximate inner cone angle of 28° at 8 mm from the nozzle and 50° for the other two planes. An estimation of the mass flux through these planes proved inconclusive for this PDA configuration. It was concluded that for this configuration, the validated LDA and PDA data rates were an additional parameter required for a complete description of the fuel spray.

In-cylinder PDA measurements were performed at the same in-cylinder positions as the 'Through Annulus' LDA air motion experiments presented in Chapter 5. In the static engine measurements, the injected fuel spray transferred momentum to the surrounding air at rest within the combustion chamber. The acceleration of the surrounding air resulted in a local pressure drop around the plume and the air flowed towards the spray and was entrained into the centre of the spray, thus forming a hollow cone shape. As the measurement position was traversed from the mid-cylinder axis towards the intake side, the measured mean, vertical velocity component increased and the delay to the first recorded droplet was reduced. At the spray axis, a pulse of data was observed that was immediately followed by a period of approximately 2 ms, which was devoid of data. This phenomenon was only observed in the time-resolved PDA measurements and was interpreted as an initial slug of fuel whose origin was in the primary phase of injection. A small amount of swirless fuel held in the injector sac volume was injected with a predominantly axial velocity component. Relatively few droplets were recorded during this phase. The droplet sizes and velocities were generally greater than those recorded in the main injection event. The same pulse of early data was observed at locations either side of the spray axis, but these did not precede a data void. More evidence was available from the photographic studies performed through the exhaust window. The image sequences clearly showed an initial slug of fuel that was followed by an asymmetrical, developing annular sheet.

Droplet measurements at the intake side cylinder wall showed relatively high negative velocities, indicating the presence of a wall-bounded vortex. This vortex was not observed in the early injection, motored studies, suggesting that the intake jet flows suppressed its formation.
A summary of the spray development was presented to show the relationship between the
temporal development of the spray structure and its relationship with the spatial distribution of the
PDA measurement locations.

In the motored studies, early injection occurred under atmospheric pressure conditions.
The location of the injector, below and between the intake valves meant that the fuel spray was
subjected to two competing jet flows in arguably the most turbulent part of the engine cycle. The
photographic studies showed that the shape of the spray plume was altered; the upper edge was
shown to become agitated and the axis of injection deflected downwards. For a SOI of 40 and 50
CA, the fuel spray was seen to impinge upon the piston crown and the exhaust side of the
chamber. The spray shape was that of a broad, hollow cone. The outline of the cone in the backlight
photographs became clearer with increased engine speed as the smaller droplets were more
effectively dispersed by the gas phase. In certain image sequences, detachment of the fuel spray
and flapping of the cone edges was observed. As the in-cylinder pressure was increased for the
case of late injection, the gas density and droplet drag force were increased. The injected droplets
were decelerated at a greater rate and momentum transfer between the phases was reduced. The
resulting spray structure showed less of the hollow cone characteristics of the early injection spray
plume. A narrow spray jet was seen to impinge upon the piston crown.

The droplet diameter probability density functions differed significantly between the static
and motored and early and late injection scenarios. The static distribution was broad and positively
skewed towards the larger droplets but the mean droplet diameter, in all but one location, was less
than in the other two cases. The motored droplet distributions were more symmetrical and showed
less flatness. For late injection, a small percentage of droplets were recorded with diameters in
excess of 70 μm. These large diameter droplets did not however constitute a significant proportion
of the droplet population during the early injection cases. For late injection, the mean droplet
diameter was always greater than those recorded during the intake stroke which were generally
greater than those recorded during the static tests. It was concluded that the difference in the mean
droplet diameters was due to the nature of the in-cylinder gas motion at the point of injection.
During early injection, the turbulent gas motion assisted the spray break-up and the smaller
droplets were dispersed in the gas phase. These droplets are carried away from the PDA probe
volume by the downward intake air motion. In the late injection case, it was only the larger droplets
that possessed sufficient momentum to reach the probe volume through the in-cylinder tumble
motion and increasing gas density. The smaller droplets were most likely to be stripped away and
carried around the chamber.

The effect of fuel injection timing upon the droplet velocity-size cross-correlation was
investigated. Under static conditions, there was no significant correlation. Under early injection, the
intake gas accelerated the liquid droplets and a positive correlation was observed. During late
injection, the diameter range was reduced and close to the spray axis, a steep positive gradient
correlation was observed. The gradient in this region was twice that of the early injection case.
From these studies, it was concluded that the secondary liquid fuel atomisation, through the
interaction with the gaseous flowfield, had become the dominant mechanism in droplet sizing and
not the injection pressure and orifice size.
The temporal evolution of the ensemble-averaged droplet diameters and velocities were compared with the LDA air motion results presented in Chapter 5 at the exact same in-cylinder locations, prior to the injection event. An ensemble-averaged, volume-weighted, mean velocity for the total number of droplet measures within each crank angle interval was calculated as a measure of the individual contribution of a droplet to the total liquid spray momentum. In this way, the significant role played by a small subset of large (and relatively high velocity) droplets in the momentum exchange between the liquid and gas phases, could be assessed. A volume mean diameter within each crank angle interval was then calculated from the ensemble-averaged, volume-weighted, mean velocity. For early injection at 120 CA, the predominant air motion was in the direction of the fuel injection and the volume-weighted velocity and volume mean diameters both increased with increasing crank angle at a rate of 0.5 ms\(^{-1}\) per CA and 0.3\(\mu\)m per CA respectively. For late injection at 290 CA and 5 mm from the cylinder axis, the air motion opposed the spray direction and there was no discernible variation in the volume-weighted characteristics.

For a SOI at 280 CA at 15 mm from the mid-cylinder location, the volume weighted droplet velocity decreased by approximately 0.5 ms\(^{-1}\) per CA. The volume mean diameters were approximately 3-4 \(\mu\)m smaller over the crank angle range than those recorded during the early injection cases. For an intermediate injection at SOI of 180 CA, the small opposing airflow led to a reduction of 0.2 ms\(^{-1}\) per CA in the volume-weighted velocity. The volume diameter did not change.

The interpretation of time-resolved PDA droplet diameter and velocity measurements of fuel sprays of this nature is complex and in most reported cases, requires the temporal decomposition of the data into specific phases of the injection event. The analysis does not provide a complete description of the fuel spray characteristics. The initial phase of fuel injection that originates from the early needle lift period imparts momentum to the gas phase resulting in a local acceleration surrounding the spray. Droplets originating later in the injection event experience less drag force and arrive behind those injected previously. These droplets will coalesce with those traveling at a slower rate in front of them, collide and break-up or pass straight through. Droplet collision and coalescence results in a droplet being recorded by the PDA measurement whose temporal origin cannot be resolved. The reference time frame is set relative to an arbitrary instant in time such as the injection trigger. The temporal measurement record however, can only log the events that occur at the PDA probe volume's fixed spatial location in chronological order.

The greatest source of potential error in the motored engine studies was that due to the correct estimation of the mean values from a limited data set. The measurements showed good repeatability. The standard error in the droplet mean velocity during early injection was \(\pm 1.0-1.5\) ms\(^{-1}\). For late injection, the standard error was \(\pm 2\) ms\(^{-1}\). The statistical uncertainty in the PDA measurements was due to low data density within each crank angle interval. This was a product of the short injection duration, spray and air interactions and the piston proximity to the probe volume. The droplets were of an indeterminable temporal and spatial origin. For these reasons, the analysis of the PDA data differs from that of the LDA data presented in Chapter 5.
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Mean Median Variance Standard Deviation

05 static 15.46 14.14 74 8.602
05_120 20.83 18.98 85.92 9.27
05_200 18.85 17.89 50.3 7.092
05_290 21.41 20.51 75.01 8.661
Figure 6.25b. Comparison of Droplet Diameter Probability Density Functions for Static and Motored Engine Operation: 5 mm from Mid-Cylinder Axis
Figure 6.25c. Comparison of Droplet Diameter Probability Density Functions for Static and Motored Engine Operation: 10 mm from Mid-Cylinder Axis

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Mean, Median, Variance, and Standard Deviation for different conditions.
Figure 6.25d. Comparison of Droplet Diameter Probability Density Functions
for Static and Motored Engine Operation: 15 mm from Mid-Cylinder Axis

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Comparison of Spatial Distribution of Mean Droplet Diameters for Static, and Motored Engine Conditions; Early versus Late Injection

Figure 6.26a. Variation of Mean Droplet Diameter with In-cylinder Measurement Location

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10 mm from Mid-Cylinder Location, Early Injection at 1000 rpm: Standard Error

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15 mm from Mid-Cylinder Location, Late Injection at 1000 rpm: Standard Error
### Table 6.0. Overlayed Circles to Determine Injector Orifice Diameter

(all measurements in μm)

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### Table 6.1. Summary of Selected Crank Angle and Pulse Width Durations for Simulated 500 rpm

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<td></td>
</tr>
<tr>
<td>6.666</td>
<td></td>
<td></td>
<td>33.2078</td>
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### Table 6.2: Summary of Dynamic Mass Flow Rate data

(Ricardo fired data provided as reference only)
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<table>
<thead>
<tr>
<th>PDA Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wavelength</td>
<td>514.5 nm</td>
</tr>
<tr>
<td>Gaussian Beam Diameter</td>
<td>0.27 mm</td>
</tr>
<tr>
<td>Beam Collimator Exp.</td>
<td>1</td>
</tr>
<tr>
<td>Beam Expansion Ratio</td>
<td>4</td>
</tr>
<tr>
<td>Beam Separation</td>
<td>32 mm</td>
</tr>
<tr>
<td>Lens Focal Length</td>
<td>160 mm</td>
</tr>
<tr>
<td>Fringe Spacing</td>
<td>2.5853 μm</td>
</tr>
<tr>
<td>Number of Fringes</td>
<td>38</td>
</tr>
<tr>
<td>Probe Volume, Δx</td>
<td>0.0975 mm</td>
</tr>
<tr>
<td>Probe Volume, Δy</td>
<td>0.0970 mm</td>
</tr>
<tr>
<td>Probe Volume, Δz</td>
<td>0.9753 mm</td>
</tr>
<tr>
<td>Transmission Polarisation</td>
<td>Parallel (0 °)</td>
</tr>
<tr>
<td>Principle Scattering Mode</td>
<td>1st Order Refraction</td>
</tr>
<tr>
<td>Phase Factor, U1-2</td>
<td>10.327 °/μm</td>
</tr>
<tr>
<td>Phase Factor, U1-3</td>
<td>5.1638 °/μm</td>
</tr>
<tr>
<td>Maximum Diameter</td>
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<tr>
<td>Angle Adjustment</td>
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<tr>
<td>Effective Scattering Angle</td>
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<tr>
<td>Lens Focal Length</td>
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<td>Ratio of RI Particle/Medium</td>
<td>1.46/1</td>
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<tr>
<td>Brewster's Angle</td>
<td>68.816 °</td>
</tr>
<tr>
<td>Max Refraction Scattering Angle</td>
<td>93.539 °</td>
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</table>

Table 6.4. Summary of the PDA Operating Parameters for the Fuel Spray Chamber
<table>
<thead>
<tr>
<th>Quiescent Spray Chamber at RTP</th>
<th>Static and Motored In-Cylinder Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spraywise instantaneous and RMS velocity components</td>
<td>Cylinder Axial instantaneous and RMS velocity components</td>
</tr>
<tr>
<td>Droplet diameter distribution</td>
<td>Droplet diameter distribution</td>
</tr>
<tr>
<td>Experimental calibration and establish measurement uncertainties</td>
<td>Limitations due to optical and geometrical constraints in the engine and synchronisation with injection timing, LASER shuttering and engine speeds</td>
</tr>
<tr>
<td>Complete spray plume traverse</td>
<td>Limited to a series of point measurements along a 'line of sight' radius of the annulus</td>
</tr>
<tr>
<td>Theoretically an infinite number of injection cycles with only small recovery periods required</td>
<td>Skip injection for fear of excessive bore wetting and window contamination</td>
</tr>
<tr>
<td>No impingement</td>
<td>Wall and crown impingement of fuel spray and interaction of piston with probe volume causing PMT overload</td>
</tr>
</tbody>
</table>

Table 6.5. Comparison of the PDA Operating Conditions between the Spray Chamber and the In-Cylinder Studies.
7.0. Summary of Conclusions and Achievements

Two comprehensive in-cylinder airflow studies have shown the contrast between the temporal and spatial distributions of the mean and turbulent gas motions, in the mid-cylinder tumble (and cross-tumble) plane, between a conventional, side-entry MPI engine and a top-entry G-DI engine. Airflow measurements were performed using LDA in the axial (MPI) and axial and radial (G-DI) directions. In the MPI engine, the principal air motion can be described as a forward tumble pattern, generated late in the intake stroke and which breaks down into turbulence close to the end of the compression stroke. In the G-DI engine, a strong, reverse tumble motion is rapidly formed early in the intake stroke. This motion persists throughout the compression stroke and up to TDC without an increase in turbulence intensity.

In the direct injection engine, the characteristics of the mean and small-scale air motions were measured through a horizontal plane across the bore and along an axis co-linear with the spark plug axis, which extended into the piston bowl. Special attention was given to the mean and turbulent components of the airflows prior to injection of fuel and spark ignition. During the early injection phase the airflow exhibited the requirements for a homogeneous charge mixture. Steep velocity gradients and the highest levels of turbulence enhance the air-fuel mixing. For late injection, the persistent coherent tumble structure produced flow stability for a stratified mixture approach.

The longitudinal integral length scale of turbulence was determined at two locations near the spark plug gap at ignition timings and TDC conditions. The indirect method and Taylor's hypothesis was used to determine the length scale from the temporal autocorrelation function estimate. The criteria for Taylor's hypothesis for stationary turbulent flow were validated for the coherent, tumbling vortex present in the late compression stroke. At crank angles close to TDC, the vortex observed a solid body rotation with minimal vortex centre precession relative to the piston crown. The turbulence intensity over the same interval showed no increase and was markedly lower in comparison with the MPI engine at the same locations. At 3 and 8 mm from the spark plug body and 330 CA, the length scale was estimated at 6.2 and 9.5 mm respectively. The longitudinal integral length scale at TDC was approximately 1/5 of the clearance height.

In-cylinder PDA measurements were performed under stationary and motored conditions, at the same locations across the mid-cylinder plane, as described for the LDA air motion study. The PDA measurements and high-speed photography showed that the fuel spray exhibited a swirling, hollow cone structure that was most evident at high engine speeds. A quasi-steady inner cone angle for the spray was estimated at 28° close to the nozzle and 50° further downstream. The in-cylinder time-resolved analyses highlighted the presence of a swirlless 'slug' of liquid fuel that preceded the main injection and asymmetrical cone development and that contained large diameter droplets with high axial velocities. A dense spray region was identified over 8 mm from the nozzle along the spray axis. In-cylinder static PDA measurements showed the presence of a wall-bounded vortex on the intake side of the chamber below the injector that was suppressed during the motored studies. For late injection, the mean droplet diameters were larger than those of the early injection and static tests and the fuel spray formed a narrow jet. The difference in the mean properties was due to the nature of the in-cylinder gas properties; either assisting spray break-up and dispersion or impeding spray penetration. The static PDA droplet probability density functions were much broader and less symmetrical than those for motored conditions. An ensemble-averaged, volume-weighted mean velocity was used to describe the individual contribution of a droplet to the total spray momentum exchanged between the gas and liquid phases. It was concluded from the comparative studies of early and late fuel injection, that the droplet size distribution was dominated by secondary atomisation through interaction with the gaseous flowfield when the fuel spray was injected into high gas velocities in the spray-wise direction. The gradient of the size-velocity cross-correlations during late injection were approximately twice those observed during early injection.
The fuel spray injection axis was deflected downwards towards the piston by as much as 10° at 1500 rpm due to the momentum of the incoming air charge. Furthermore, the uppermost periphery of the fuel spray was subjected to flapping instabilities and partial detachment.

7.1. Recommendations for Future Work

Future work can be divided into two categories. The first category is drawn from a review of the results presented within this thesis and includes suggestions where the current work could be extended or further validated (the analysis of air motion in Chapter 5 is used to validate a CFD model in Faure et al., (1998)). In the first instance, the current study could be extended to include an investigation of differing types of fuel injector, alternative combustion chamber geometry's and further investigation and development of the optical techniques employed with an aim to improving data acquisition rates and measurement in two or more simultaneous directions.

The indirect determination of the integral length scale of turbulence based upon Taylor's hypothesis used the method of single point LDA and an evaluation of the temporal autocorrelation function. Further work must include validation of such a hypothesis by the direct estimation of the integral scale using two-point LDA and integration of the spatial autocorrelation function. In addition, an evaluation of the effects of lag range and data filtering, implicit in the method, should be reviewed. In some reported cases, the deviation of the autocorrelation estimate from unity has been attributed to a slow, time-varying contribution from the cycle-to-cycle variation in the mean flow (tumble vortex centre jitter) that is manifested as an apparent turbulent contribution. Further analysis of the results is required to determine the frequency content of the turbulence fluctuations and hence derive that attributable to cycle-to-cycle variations. The data acquisition rate should be improved to enable an in-cycle data analysis comparison with the ensemble-averaging procedure. A similar data analysis can be performed in the cross-tumble plane.

Further analysis of the spray data under motored engine conditions would allow the extraction of useful data about the post-impinged droplet diameters and velocities required to validate spray impingement models. These subsets of the total droplet population could be separated by comparison of the fuel spray statistics measured at elevated pressures in conditions where impingement did not occur. The impingement study could also be extended to include impingement upon realistic (hot) surfaces to investigate the effects of evaporation upon the post-impinged droplet population. In addition, there exists evidence to suggest that the magnitude of the tumble motion is significantly decreased during the compression stroke when fuel is centrally injected into the chamber of a forward tumble, side-entry, direct injection engine (e.g. Stanglmaier et al., (1998)). As such, the in-cylinder LDA airflow study could be extended to include an analysis of the momentum exchange effects induced by the direct injection of fuel.

In the current study, no quantitative analysis of the higher order statistical moments was undertaken. The static and motored droplet distribution statistics (e.g. skewness and kurtosis) could be used to enable suitable curve fits of common statistical distributions that would be applicable to computational modelling. In addition, a further research area is in the validation of droplet dispersion models.

The second category involves the next step in the characterisation of a suitable operating strategy for G-DI. At present, the fundamental properties of the in-cylinder air motion, throughout the engine cycle and prior to injection and ignition, as well as the characteristics of a fuel spray under these conditions, have been quantified. The next stage is to apply measurement techniques, such as LIF, that can provide knowledge of the interaction between the air and fuel; the fuel liquid and vapour distribution in the combustion chamber, its cyclic stability, homogeneity or degree of stratification. This in turn must be related to an appropriate measure of combustion stability. A combustion study would include analysis of the rate of flame propagation and flame geometry (flame photography); the nature of the airflow ahead of the flame front (fired LDA); an identification of the chemical species distributed within the flame (LIF/Spectroscopy); pressure-related combustion parameters such as IMEP and combustion stability and engine-out emissions sampling.
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    **Lecture 5: Multi-Dimensional Modelling II, ‘SI Engine Combustion and Emissions’**  
    Gosman, A. D.
List of Publications by the Author


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Appendix A: Temporal Characteristics of Instantaneous Velocity, Ensemble-Averaged Mean and RMS Velocity across the Engine Cycle for the SW Measurement Point Series

### Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position Sweep 1

- **EVC** (End of Compression Stroke)
- **IVC** (Intake Valve Closure)
- **EVO** (End of Expansion Stroke)
- **EV Peak** (Early Valve Timing)
- **IVO** (Intake Valve Opening)

### Ensemble Averaged Mean Velocity against Crank Angle

- **Ensemble Averaged (EA)**
- **Smoothed EA**

### Ensemble-Averaged RMS Turbulence Velocity

### Data Counts per Ensemble-Averaging Window

*Figure A1. LDA Data Set Sweep Location 1*
Figure A2. LDA Data Set Sweep Location 2
Instantaneous Velocity against Crank Angle in a Four-stroke, Part-roof Gasoline Engine
Position : Sweep 3

Ensemble Averaged Mean Velocity against Crank Angle

\[
\begin{align*}
\text{Ensemble Averaged (EA)} & : \quad \text{Smoothed EA} \\
90 & , 180, 270 \\
\end{align*}
\]

Ensemble Averaged RMS Turbulence Velocity

\[
\begin{align*}
\text{Ensemble Averaged (EA)} & : \quad \text{Smoothed EA} \\
90 & , 180, 270, 360 \\
\end{align*}
\]

Figure A3. LDA Data Set Sweep Location 3
Figure A4. LDA Data Set Sweep Location 4
Figure A5. LDA Data Set Sweep Location 5
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position Sweep 6

Ensemble Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure A6. LDA Data Set Sweep Location 6
Figure A7. LDA Data Set Sweep Location 8
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position Sweep 9

Ensemble Averaged Mean Velocity against Crank Angle

Figure A8. LDA Data Set Sweep Location 9
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-rod Gasoline Engine
Position Sweep 11

Ensemble Averaged Mean Velocity against Crank Angle

---

Ensemble-Averaged RMS Turbulence Velocity

---

Figure A9. LDA Data Set Sweep Location 11
Appendix B: Temporal Characteristics of Instantaneous Velocity, Ensemble-Averaged Mean and RMS Velocity across the Engine Cycle for the SP Measurement Point Series

![Graph: Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position Spark Point 1 Ensemble Averaged Mean Velocity against Crank Angle](image)

![Graph: Ensemble-Averaged RMS Turbulence Velocity](image)

*Figure B1. LDA Data Set Spark Location 1*
Figure B2. LDA Data Set Spark Location 2
Figure B3. LDA Data Set Spark Location 3
Figure B4. LDA Data Set Spark Location 4
Figure B5. LDA Data Set: Comparison of Ensemble-Averaged Mean Velocity at Three Spark Locations
Appendix C: Temporal Characteristics of Instantaneous Velocity, Ensemble-Averaged Mean and RMS Velocity across the Engine Cycle for the T-Measurement Point Series

![Graphs showing instantaneous velocity, ensemble-averaged mean, and RMS velocity against crank angle.]

**Figure C1.** LDA Data Set TN15P00
Figure C2. LDA Data Set TN30P00
Figure C3. LDA Data Set TP00P00
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position TN15P00
Ensemble Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure C4. LDA Data Set TN15P00
Figure C5. LDA Data Set TN30P00
Figure C6. LDA Data Set TP00P00
Figure C7. LDA Data Set TN15P00
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position TN30P00
Ensemble-Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure C8. LDA Data Set TN30P00
Figure C9. LDA Data Set TP00P00
Figure C10. LDA Data Set TP15P00
Figure C11. LDA Data Set TP00P00
Figure C12. LDA Data Set TN15P00
Instantaneous Vsbclty against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position TP00P00

Ensemble Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure C13. LDA Data Set TP00P00
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine
Position TP15P00
Ensemble Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure C14. LDA Data Set TP15P00
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine
Position TP30P00
Ensemble Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure C15. LDA Data Set TP30P00
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine
Position TN15P00
Ensemble Averaged Mean Velocity against Crank Angle

Ensemble-Averaged RMS Turbulence Velocity

Figure C16. LDA Data Set TN15P00
Instantaneous Velocity against Crank Angle in a Four-stroke, Pent-roof Gasoline Engine Position TP00P00
Ensemble Averaged Mean Velocity against Crank Angle

Ensemble Averaged RMS Turbulence Velocity

Figure C17. LDA Data Set TP00P00
Appendix D: Interpolation Scheme Generated Contour Plots for the Horizontal Mean Velocity Component in the Intake and Compression Strokes

Figure D1. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Ensemble Averaged Mean Velocity at 60CA at 1000 rpm

10 m/s

Ensemble Averaged Mean Velocity at 80CA at 1000 rpm

10 m/s

Figure D2. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Figure D3. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Figure D4. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Reconstruction
Figure D5. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Ensemble Averaged Mean Velocity at 220CA at 1000 rpm

Ensemble Averaged Mean Velocity at 240CA at 1000 rpm

Figure D6. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Figure D7. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Reconstruction
Figure D8. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Figure D9. Horizontal Scalar Velocity Contour Plot based upon Interpolated Velocity Field Re-construction
Appendix E: Interpolation Scheme Generated Extracted Velocity Scalar Plots for the Horizontal Mean Velocity Component in the Intake and Compression Strokes

Figure E1. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E2. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E3. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E4. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E5. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E6. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Ensemble Averaged Mean Velocity at 260CA at 1000 rpm

Ensemble Averaged Mean Velocity at 280CA at 1000 rpm

Figure E7. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E8. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Figure E9. Extracted Velocity Horizontal Scalars based upon Interpolated Velocity Field
Appendix F: Instantaneous and Ensemble-Averaged Mean and RMS Velocities in the Tumble Plane along the Spark Plug Axis in a Top-Entry, Gasoline, Direct Injection Combustion Chamber

![Graphs showing instantaneous and ensemble-averaged velocity data against crank angle encoder counts for a direct injection gasoline engine.]

Figure F1. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 01, Tumble Plane
Figure F2. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 02, Tumble Plane
Figure F3. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 03, Tumble Plane
Figure F4. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 04, Tumble Plane
Figure F5. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 05, Tumble Plane
Figure F6. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 06, Tumble Plane
Figure F7. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 07, Tumble Plane
Appendix G: Instantaneous and Ensemble-Averaged Mean and RMS Velocities in the Cross-Tumble Plane along the Spark Plug Axis in a Top-Entry, Gasoline, Direct Injection Combustion Chamber

Instantaneous Velocity Against Crank Angle Encoder Counts
1500 rpm WOT/Cross-Tumble 01 / 33 mm Depth

Ensemble Averaged Velocity Data Against Crank Angle Encoder Counts 1500 rpm WOT/Cross-Tumble 01 / 500 bins

Figure G1. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 01, Cross-Tumble Plane
Figure G2. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 02, Cross-Tumble Plane
Figure G3. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 03, Cross-Tumble Plane
Figure G4. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 04, Cross-Tumble Plane
Figure G5. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 05, Cross-Tumble Plane
Figure G6. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 06, Cross-Tumble Plane
Figure G7. LDA Results for Direct Injection Gasoline Engine
Spark Plug Location 07, Cross-Tumble Plane
Appendix H: Instantaneous and Ensemble-Averaged Mean and RMS Axial Velocities in the Mid-Cylinder Plane at z=10 mm in a Top-Entry, Gasoline, Direct Injection Combustion Chamber

Figure H1. Velocity Measurements at z=10 mm, Location R05F00
Figure H2. Velocity Measurements at z=−10 mm, Location R00F00
Figure H3. Velocity Measurements at z=10 mm, Location N05F00
Figure H4. Velocity Measurements at z=-10 mm, Location N10F00
Figure H5. Velocity Measurements at z=10 mm, Location N15F00
Figure H6. Velocity Measurements at z=10 mm, Location N20F00
Figure H7. Velocity Measurements at z = 10 mm, Location N25F00
Figure H8. Velocity Measurements at z=-10 mm, Location N30F00
Instantaneous Velocity Against Crank Angle
500 rpm WOT/N35F000 / 10 mm Depth

Ensemble Averaged Velocity Data Against Crank Angle 500 rpm WOT/N35F000 / 500 bins

Ensemble Averaged Velocity Data Against Crank Angle 500 rpm WOT/N35F000 / 500 bins

Figure H9. Velocity Measurements at z=10 mm, Location N35F00
### Appendix I: Operating Conditions for the Kodak EktaPro HS Motion Analyser, Model 4550

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