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

Steven Begg

Volume I of II

Ph.D. 2003
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Volume I of II

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A thesis submitted in partial fulfilment of the requirement of the University of Brighton for the Degree of

Doctor of Philosophy

January 2003

School of Engineering, University of Brighton

in collaboration with

Ricardo Consulting Engineers
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Abstract

An experimental study of the mixture preparation characteristics of a direct injection gasoline (G-DI) engine is presented. The airflow and fuel spray features of a top-entry, lean-burn, direct injection strategy are described by the application of advanced LASER-based experimental tools and techniques.

Two comprehensive in-cylinder airflow studies were undertaken to show the contrast between the temporal and spatial distributions of the mean and turbulent gas motions, in the mid-cylinder tumble (and cross-tumble) plane, of a conventional, side-entry, port injected (MPI) engine with flat-top piston and a top-entry G-DI engine with a bowled piston. Airflow measurements were performed using LASER Doppler Anemometry (LDA) in the axial (MPI) and axial and radial (G-DI) directions in a motored, single cylinder, research engine over consecutive engine cycles. Optical access to the combustion chambers was achieved by three different methods. In the MPI engine, a modified piston that incorporated a circular window was used in conjunction with an inclined mirror, fixed below the piston. Experimental measurements in the G-DI were performed along the axis of the spark plug and into the piston bowl and within the upper cylinder and the piston topland region using a spark plug window and glass annular section, sandwiched between the cylinder liner and cylinder head.

In the MPI engine, the principal air motion can be described as a forward tumble pattern, established 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 persisted throughout the compression stroke and up to TDC without an increase in turbulence intensity. The magnitude of the tumble plane velocity was far greater than that measured in the cross-tumble plane. Observed levels of RMS turbulence intensity were significantly greater in the conventional cylinder head towards the end of the compression stroke indicating the conservation of the reverse tumble motion in the G-DI system. During the early injection phase the airflow exhibited the requirements for a homogeneous charge mixture. 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.

Phase Doppler Anemometry (PDA) measurements were taken of a high pressure, swirl type gasoline fuel injector. Droplet size, velocity, RMS velocity and validated data rate measurements were used to quantify the spray atomisation quality and geometry under atmospheric, quiescent conditions and within a motored engine for early and late injection scenarios. High-speed LASER light sheet photography was used for visualisation. In-cylinder PDA measurements were performed at the same locations across the mid-cylinder plane, as employed for the LDA air motion study. The PDA measurements and high-speed photography showed that the fuel spray exhibited a swirling, hollow cone structure during early injection. A quasi-steady state inner cone angle for the spray was estimated.
For early injection conditions, a pre-injection, swirlless 'slug' of fuel was observed, predominantly along the spray axis, in the time-resolved experiments, that preceded the main injection and asymmetrical cone development and that contained relatively large diameter droplets with high axial velocities. 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 attributed to the nature of the in-cylinder gas properties; An ensemble-averaged, volume-weighted mean velocity was used to describe the 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 fuel spray injection axis was deflected downwards towards the piston by the momentum of the incoming air charge. The uppermost periphery of the fuel spray was subjected to flapping instabilities and partial detachment.

Observations for a suitable mixture preparation strategy are highlighted for a direct injection gasoline engine, based upon the conclusions drawn from a comprehensive experimental study. The effect of in-cylinder airflow on the fuel spray characteristics is quantified for both early and late injection operations. The stability of stratification under motored conditions is discussed in terms of the in-cylinder air motion, turbulence intensity, integral length scale of turbulence and its ability to control the distribution of the mixture and preparation of suitable conditions for initial flame kernel growth. It was concluded that models for mixture preparation based upon pre-conceptions derived from manifold injection, tumble combustion systems and high-pressure Diesel fuel spray analyses are not readily applicable to a strategy that is based upon the direct injection of gasoline into the engine combustion chamber under varying pressure and airflow conditions.
Authors Declaration

I hereby certify that this thesis is my own work except where otherwise indicated. I have identified my sources of information and in particular, have put into quotation marks and identified the origins of any passages that have been quoted word for word.

Signed: [Signature]
Date: 2nd June 2003.
Acknowledgements

I would like to take this opportunity to acknowledge the contribution of the many people who have helped in the preparation of this thesis. Firstly, I would like to thank my supervisor, Professor M. Heikal for his guidance, expert knowledge and experience. I also wish to thank Ricardo Consulting Engineers for their financial, technical and academic support without which this project would not have been possible. In particular, I would like to extend special thanks to Mr N. Jackson, Mr J. Stokes, Dr S. Edwards, Miss M. Sadler (formerly of Ricardo), Dr C. Hogg (formerly of the University of Brighton) and Dr M. Gold.

I extend my thanks to those family and friends, colleagues and students, both past and present, that have contributed in many different ways to the body of work contained within this thesis. In particular, I would like to acknowledge the technical support in the School of Engineering and particularly Mr P. Clarke, Mr T. Brown, Mr K. Watson and Mr W. Whitney.

I would like to extend my thanks to my academic colleagues within the University and within the School of Engineering who taught and advised me and who have helped me gather my thoughts and reasons behind this work. Special thanks go to Professor A. Johns, Dr. D. Kennaird, Dr. J. Kearsey, Mr R. Wood, Dr. E. Sazhina and Professor S. Sazhin, Dr. A. Bruce, Dr. M. Jones, the folk from the ‘Old Research Room’ and the folk in the ‘New Research Room’ and any other folk I’ve missed!

Aside, I would like to personally express my deepest gratitude to Dr. D. Mason for his hours and hours of kind help in the correction stages; for his discussion, critique and comment, and for his infectious desire to learn and understand. (and for his constant prodding!).

I wish to also thank Mr A. Walker of the EPSRC Instrument Loan Pool for the loan of the remarkable equipment used in the photographic studies. I would also like to thank Mr G. Hassle and Mr. R. Jaryczewski of Dantec Dynamics, for their valuable input and equipment loans in times of dire need.

Finally and most importantly, my deepest thanks go to my wife, Isabelle, for her love, support and inspiration during the difficult times. I dedicate this work to Isabelle and to our son Etienne (aged 20 and 1/2 months).
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# Nomenclature

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<tr>
<td>$C_d$</td>
<td>drag coefficient of droplet in air, $[\text{ ]}$</td>
</tr>
<tr>
<td>$d$</td>
<td>liquid droplet diameter, $[\text{m}]$</td>
</tr>
<tr>
<td>$D_a$</td>
<td>Damköhler number, $[\text{ ]}$</td>
</tr>
<tr>
<td>$D_c$</td>
<td>molecular diffusivity, $[\text{m}^2\text{s}^{-1}]$</td>
</tr>
<tr>
<td>$f$</td>
<td>frequency, $[\text{Hz}]$</td>
</tr>
<tr>
<td>$F_{c_o}, f_c$</td>
<td>in-cylinder turbulence 'Cut-Off' frequency, $[\text{Hz}]$</td>
</tr>
<tr>
<td>$i$</td>
<td>individual engine cycle, $[\text{ ]}$</td>
</tr>
<tr>
<td>$k$</td>
<td>turbulent kinetic energy, $[\text{m}^2\text{s}^{-2}]$</td>
</tr>
<tr>
<td>$k$</td>
<td>wave number vector, $[\text{m}^{-1}]$</td>
</tr>
<tr>
<td>$K$</td>
<td>volumetric turbulent kinetic energy (per unit mass), $[\text{m}^2\text{s}^{-2}\text{kg}^{-1}]$</td>
</tr>
<tr>
<td>$l_i$</td>
<td>Eulerian integral length scale, $[\text{m}]$</td>
</tr>
<tr>
<td>$l_k$</td>
<td>Kolmogorov length scale, $[\text{m}]$</td>
</tr>
<tr>
<td>$l_m$</td>
<td>Taylor Micro length scale, $[\text{m}]$</td>
</tr>
<tr>
<td>$n$</td>
<td>total number of measurements or engine cycles, $[\text{ ]}$</td>
</tr>
<tr>
<td>$N, N_c$</td>
<td>total number of engine measurement cycles, $[\text{ ]}$</td>
</tr>
<tr>
<td>$Ns$</td>
<td>number of evenly spaced intervals across engine cycle, $[\text{ ]}$</td>
</tr>
<tr>
<td>$N_t$</td>
<td>total number of measurements within a defined crank angle interval, $[\text{ ]}$</td>
</tr>
<tr>
<td>$p_f$</td>
<td>firing in-cylinder gauge pressure, $[\text{Pa}]$</td>
</tr>
<tr>
<td>$p_m$</td>
<td>motoring in-cylinder gauge pressure, $[\text{Pa}]$</td>
</tr>
<tr>
<td>$r$</td>
<td>radius of swirl, $[\text{m}]$</td>
</tr>
<tr>
<td>$R$</td>
<td>separation distance vector, $[\text{m}]$</td>
</tr>
<tr>
<td>$R_v(r)$</td>
<td>spatial or Eulerian velocity autocorrelation function $[\cdot \cdot \cdot]$</td>
</tr>
<tr>
<td>$R_v(\tau)$</td>
<td>temporal or Lagrangian velocity autocorrelation function ($R_v$), $[\cdot \cdot \cdot]$</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number, $[\text{ ]}$</td>
</tr>
<tr>
<td>$R_c$</td>
<td>cylinder radius, $[\text{m}]$</td>
</tr>
<tr>
<td>$S_L$</td>
<td>laminar flame burning velocity, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$S_t$</td>
<td>mean turbulent burning velocity, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$SRN$</td>
<td>swirl Reynolds number, $[\text{ ]}$</td>
</tr>
<tr>
<td>$t$</td>
<td>instantaneous time, $[\text{s}]$</td>
</tr>
<tr>
<td>$u, U(t)$</td>
<td>instantaneous normal velocity, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$\bar{u}$</td>
<td>mean normal velocity, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$u(t)$</td>
<td>fluctuating normal velocity component, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$u', U'_T$</td>
<td>turbulence intensity, $[\text{ ]}$</td>
</tr>
<tr>
<td>$u_{n}, u_n$</td>
<td>transit or residence time-weighted mean velocity, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$v$</td>
<td>instantaneous tangential or radial velocity, $[\text{ms}^{-1}]$</td>
</tr>
<tr>
<td>$V_p$</td>
<td>increase in measured velocity variance due to a velocity gradient, $[\text{m}^2\text{s}^{-2}]$</td>
</tr>
</tbody>
</table>
Greek Symbols

θ  crank angle ................................................................. [°]
δL  laminar flame thickness ............................................. [mm]
ε  rate of turbulent energy dissipation ......................... [m² s⁻³]
σe  standard error (in mean or RMS velocity estimates) .... [m s⁻¹]
Δt  particle transit time within LDA probe volume .......... [s]
σ  surface tension ......................................................... [kg m⁻²]
σp  standard deviation in particle velocity distribution across probe volume [m s⁻¹]
ρ  density ......................................................................... [kg m⁻³]
ν  kinematic viscosity ..................................................... [m² s⁻¹]
Φ  energy spectrum tensor ........................................... [m³ s⁻²]
Ø  phase angle, phase slot width angle with respect to crank angle [°]
ζ  vorticity ................................................................. [s⁻¹]
τ  time increment or delay ........................................... [s]
τI  Lagrangian integral time scale .................................. [s]
τL  chemical reaction time ........................................... [s]
τm  Taylor micro time scale ........................................... [s]
τT  turbulent eddy turnover time .................................. [s]
ω  swirl ratio based upon the ensemble-averaged swirl velocity [ ]
ω'  swirl ratio defined as the ratio of ω to the angular momentum of a solid body rotation ..................... [ ]
ωe  engine speed ............................................................ [rad s⁻¹]

Optical Reference Symbols

θ  light beam intersection angle ........................................ [°]
β  LASER beam angle of divergence ......................... [mrad]
Φ  light scattering angle between transmission and collection axes in the horizontal plane [°]
ψ  elevation angle of collection axis in vertical plane from the horizontal plane [°]
α  angle between surface tangent and incident light ray (in Figure 3.11.) [°]
α'  angle between surface tangent and refracted light ray (in Figure 3.11.) [°]
\( \alpha \) angle between \( \vec{V} \) and measured component of \( \vec{V}, U, \) perpendicular to the fringe pattern in the plane of the incident LASER beams \( \quad \left[{}^\circ\right] \)

\( \nu \) angle between incident ray and \( p^{th} \) order ray exiting a spherical droplet \( \quad \left[{}^\circ\right] \)

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

\( \Phi \) phase shift measured at the PDA collection optic \( \quad \left[{}^\circ\right] \)

\( s \) fringe spacing \( \quad \left[\mu\text{m}\right] \)

\( p \) light scattering designation order within a spherical droplet \( \quad \{ \} \)

\( q \) Mie size parameter \( \quad \{ \} \)

\( S \) Mie scattering function \( \quad \{ \} \)

\( m \) refractive index \( \quad \{ \} \)

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

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

\( N \) number of diffraction grating line pairs \( \quad \{ \} \)

\( N_I \) number of fringes \( \quad \{ \} \)

\( \lambda \) wavelength \( \quad \left[\text{mm}\right] \)

\( d \) line pair spacing of diffraction grating \( \quad \left[\text{mm}\right] \)

\( d_g \) Gaussian beam diameter \( \quad \left[\text{mm}\right] \)

\( d_p \) normalised particle diameter \( \quad \{ \} \)

\( r_g \) Gaussian beam radius \( \quad \left[\text{mm}\right] \)

\( r_0 \) minimum Gaussian beam radius at beam waist \( \quad \left[\text{mm}\right] \)

\( F \) transmission lens focal length \( \quad \left[\text{mm}\right] \)

\( b \) beam separation at the transmission lens \( \quad \left[\text{mm}\right] \)

\( I \) light intensity \( \quad \left[\text{Wm}^{-2}\right] \)

\( E \) beam expansion factor \( \quad \{ \} \)

\( F_s \) optical frequency shift \( \quad \left[\text{Hz}\right] \)

\( f_r \) rotating diffraction grating speed \( \quad \left[\text{rpm}\right] \)

\( f_d \) Doppler frequency \( \quad \left[\text{Hz}\right] \)

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

\( R \) refraction Fresnel coefficient \( \quad \{ \} \)

\( r \) reflection Fresnel coefficient \( \quad \{ \} \)

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

Subscripts

\( e \) \text{LASER beam extremities}\n
\( E \) \text{LASER beam centre}\n
\( EA \) ensemble-averaged\n
\( HF/LF \) high frequency / low frequency\n
\( l \) liquid phase\n
\( rel \) relative velocity between phases
**List of Abbreviations**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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</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>
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<tr>
<td>DOHC</td>
<td>Dual Overhead Camshafts</td>
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<tr>
<td>EGR</td>
<td>Exhaust Gas Recirculation</td>
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<td>EOI</td>
<td>End of Injection Timing</td>
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<td>FIE</td>
<td>Fuel Injection Equipment</td>
</tr>
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<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>
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<tr>
<td>IVC</td>
<td>Inlet Valve Closure</td>
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<td>LALLS</td>
<td>Low Angle Light Scattering</td>
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<td>LBT</td>
<td>Ignition Timing for Leanest Burn and Best Torque</td>
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<td>LDA</td>
<td>LASER Doppler Anemometry</td>
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<td>LIF</td>
<td>LASER Induced Fluorescence</td>
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<td>LASER Induced Incandescence</td>
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<td>LLS</td>
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<td>LTR</td>
<td>Linear Trend Removal</td>
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<td>MBT</td>
<td>Minimum Ignition Timing Advance for Best Torque</td>
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<td>Mean Piston Speed</td>
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<td>NOP</td>
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<td>Oxides of Nitrogen</td>
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</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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1.0. Characteristics and Mixture Preparation Strategies of Direct Injection Gasoline Spark Ignition Engines

1.1. Introduction

In this section, a survey of the present literature associated with the characterisation and development of lean-burn gasoline engine strategies and the experimental techniques associated with the measurement of in-cylinder airflow and gasoline fuel spray parameters is presented. The objective of this review is to highlight the features of a lean-burn, direct injection gasoline (G-DI) system that can best be described by the application of advanced experimental tools and techniques as applied in this thesis.

The legislative targets set by the regulatory bodies of Japan, Europe and the USA require significant reductions in exhaust emissions and fuel consumption in the forthcoming years. Figure 1.0. presents a comparison of worldwide gasoline emission legislation and indicates future values set by EURO III/IV/V (Europe), LEV/ULEV/SULEV (USA) and the Japanese 10-15 mode tests. In addition to these environmental conservation requirements, consumer demand for increased power and comparable driveability and refinement from future generations of vehicles influences the design of automotive powerplants. Recent decades have seen the successful replacement of stoichiometric, carburetted SI engines with those incorporating lean-burn and multi-point, manifold port, fuel injection (MPI) strategies, in an attempt to meet these goals.

The potential of lean-burn gasoline and high-speed direct injection Diesel (HSDI) engines to meet these objectives in the forthcoming years has pushed them once more to the forefront of automotive research. In particular, key advances in automotive ancillary technologies have provided the opportunity for ultra-lean burn gasoline fuel injection strategies, previously regarded as having an inadequate operating range for the demands of modern passenger carrying vehicles. Perhaps the single most important factor in this re-emergence has been the advent of highly controllable, high-pressure and relatively inexpensive common rail fuel injection equipment (FIE). When coupled with suitable methods of exhaust gas recirculation (EGR) and exhaust gas after treatment processes, lean-burn engines have the potential to meet the stringent emission requirements of current and future legislation. One of the main aims of researchers in this field has been to develop lean-burn engines that are on at least equal emission, fuel consumption and performance terms with modern, optimised, multi-valve, MPI gasoline engines.

The main requirement for the quantitative governing of a spark-ignition engine operation is that the incoming charge is completely and homogeneously mixed with the available fuel so that a uniform air-to-fuel ratio (AFR) is achieved. Throttling of the intake air charge is necessary to match engine load changes, which reduces the engine breathing capacity or volumetric efficiency. Diesel combustion systems are, however, controlled by the metering of fuel (variable AFR) as the engine load is varied. As such, unthrottled operation is achievable minimising the pumping losses associated with flow restrictions. Figure 1.1. shows the throttling effect upon the measured log pressure-volume diagrams for an MPI and a G-DI engine cycle at the same test point.
A more efficient engine operation is possible at part-load conditions due to the removal of the pumping work loop in the G-DI configuration. Gasoline direct injection (G-DI) engines aim to utilise this principle to achieve an efficiency gain under part-load (unthrottled) operation and as well as exploiting the full-load performance of a homogeneous mixture gasoline engine. Engines utilising direct fuel injection do not require the mixture preparation stage associated with port-injected engines. A controlled amount of fuel is injected directly into the cylinder volume at injection timings dictated by the engine speed, engine load requirement, piston position, injector characteristics and in-cylinder gas motion. The development of these strategies requires an in-depth understanding of the effects of flow behaviour and fuel spray characteristics in the combustion chamber on the combustion processes and the combustion stability. Fundamental studies of fuel-air mixing and the production of liquid and vapour fuel rich zones are required in order to achieve this understanding.

Generally, mixture preparation strategies in G-DI can be defined as stoichiometric homogeneous, lean homogeneous or lean stratified charge. The selection of a particular strategy is dependent on the engine load and speed conditions. A strategy for direct injection gasoline engine operation is shown in Figure 1.2. At or near full-load, fuel and air are prepared, as a homogeneous mixture in the induction stroke, at AFR's comparable to those of an MPI engine. The injector location, orientation, injection timing and the charge motion are instrumental in ensuring good mixture homogeneity and the prevention of fuel impingement upon the piston, open inlet valves and the cylinder liner surfaces. Under part-load conditions, a stratification of the in-cylinder products is achieved by the injection of a reduced quantity of fuel at a given point in the late induction or compression stroke. Excess oxygen ensures that the total in-cylinder AFR is lean. Air motions and/or piston interaction is then responsible for delivering fuel rich mixtures to the spark plug at the correct instant for ignition and efficient combustion under local stoichiometric conditions.

By the use of precise fuel injection control and fuel metering and a combination of the above injection strategies with selective geometrical enhancements, it is possible to maintain stable combustion at global AFR's in excess of 40:1. Load control can theoretically be considered a function of injection timing, ignition phasing and fuel injection pulse duration or fuel quantity injected. The necessity to throttle the intake charge can thus be greatly reduced. The displacement of a fresh, intake air volume by residing fuel vapour, through wall film fuel evaporation in the port is minimised, thus offering potential gains in engine volumetric efficiency. Load control through metered fuel quantity injected directly into the cylinder at wide-open throttle (WOT) improves engine transient response to driver demand (accelerator pedal) and is not impeded by the fuel and air transport lags associated with port injection. Engine cold-start with fuel enrichment is a significant source of unburnt Hydrocarbons (uBHC) emissions over the first engine cranking cycles in MPI engines. Injecting directly reduces the number of cranking cycles and has the potential to decrease cold-start unburnt hydrocarbons (uBHC) and carbon dioxide emissions.

The direct injection of a gasoline fuel spray into the combustion chamber promotes charge air cooling. Heat is absorbed from the intake air charge, which contributes to the evaporation of the liquid fuel.
An overall reduction in charge temperature (estimated at up to 30 K), and therefore charge density in late compression, reduces the octane value at which detonation will occur for a given fuel. At best, the compression ratio can be increased by up to 1.5 ratios over conventional MPI engines currently limited to approximately 12:1.

As global combustion temperatures are reduced in lean-burn stratified charge systems, the thermal cooling load is reduced and net gains in engine thermal efficiency are possible. Maximisation of these gains can lead to increased engine specific power output with reduced brake specific fuel consumption. Combustion of a compact, local stoichiometric mixture surrounded by an excess of air offers potential benefits in the reduction of exhaust emissions and soot as well as fuel consumption. However, the composition of engine-out exhaust emissions is a product of multiple factors that are inherently linked to the type of G-DI system employed. The complex relationship is illustrated in Figure 1.3. The plots show a comparison of the measured brake specific fuel consumption, hydrocarbon and NOx emissions levels between an MPI and G-DI engine. The significance of these results is discussed further in the relevant sections of the text.

The following section details G-DI combustion concepts relevant to this study. The subject of this study is top-entry, tumble motion based, direct injection gasoline engines that may be operated using homogeneous stoichiometric, homogeneous lean or lean stratified charge strategies. References to other types of lean-burn G-DI or MPI engines are included for comparison of performance, informative discussion or to examine the merits of the experimental practices employed.

1.2. Characteristics and Mixture Preparation Strategies for Top-Entry, Direct Injection Gasoline Spark Ignition Engines

Lean-burn direct injection gasoline engines can be categorised by the mechanisms that dominate the preparation of the air and fuel mixture within the cylinder. The approaches are many and varied and involve matching in-cylinder airflow with fuel delivery systems. Numerous researchers, automotive manufacturers and consultants have reviewed and proposed strategies for mixture preparation and combustion stability in lean-burn homogeneous and stratified charge gasoline engines. Examples include Moriyoshi et al., (1995), Kume et al., (1996), Fraidl et al., (1996), Arcoumanis et al., (1996), Tomada et al., (1997), Zhao et al., (1997), Harada et al., (1997), Jackson et al., (1996, 1997b), Lake et al., (1998) and Preussner et al., (1998). Methods of mixture preparation can further be classified by the sense of rotation of the air motion within the cylinder and the characteristics of the FIE employed. Preussner et al., (1998), in their work on high-pressure fuel injectors, identified three main techniques for mixture preparation within a stratified charge concept: spray-guided, wall-guided and air-guided. The classification of these different approaches are shown schematically in Figure 1.4. These models are rarely used in isolation, but nonetheless serve to highlight the salient features of modern stratified charge engines. They differ principally in their approach to the type of in-cylinder air motion and injector and spark plug location and the mechanism of fuel transport.
Previous research efforts were concentrated on establishing stratified charge concepts where air and fuel mixtures were prepared prior to introduction in the cylinder (e.g. Mitsubishi Motors Corporations' 'Three Layer Barrel Stratification', AVL's 'Tri-Flow' swirl concept and the Ricardo Consulting Engineers' 'Combustion Control through Vortex Stratification'- CCVS). Principally air-guided MPI systems, these utilised a suitable in-cylinder gas motion to maintain stratification between the port-prepared, fuel-air mixture and the remaining in-cylinder contents. Port air-fuel mixture preparations however, can introduce detrimental effects due to the deposition of liquid fuel films on port walls, valve seats and stems. Cold starting suffers from increased levels of uBHC emissions and fuel film wall stripping and evaporation induces an inherent fuel transport delay or lag, especially evident where closed-valve injection (CVI) is employed. Open-valve injection also suffers from reduced transient response and this is observed by the relative poor acceleration of such units.

Air-guided G-DI systems are derived in general from lean-burn MPI stratified charge concepts. Following inlet valve closure, the in-cylinder induction generated flowfield maintains stratification of the mixture about an axis parallel to the cylinder axis and often in the upper part of the cylinder. Air guided G-DI systems with central or side located injectors operate under full-load homogeneous conditions by ensuring a good mixture of air and fuel whilst minimising contact with the piston surface or cylinder liner. Stratified charge operation may be effective over only a very narrow range of operating conditions where suitable injection timing allows adequate time for mixture formation and reduces fuel impingement on the piston surface.

Spray-guided systems are governed by the characteristics of the fuel spray introduced to the cylinder. They are generally categorised by a central injector and spark plug, with the plug often positioned within or on the periphery of the spray plume. Injector design governs the degree of spray penetration. Stratification can only be achieved in the vicinity of the spark plug location. However, the proximity of plug and injector reduces the time for adequate mixture preparation and often leads to plug or injector fouling.

Wall-guided concepts rely on the successful interaction of injected spray, in-cylinder air motion and piston position. The fuel spray is directed to impinge on the piston crown and bulk airflow is utilised to transport liquid and vapour fuel to the spark plug region within an appropriate timescale. Piston crown geometries are adapted to suit the in-cylinder air motion, fuel mixing and transport processes and often incorporate central or offset piston bowls like those of Kume et al., (1996), Iwamoto et al., (1997) and Jackson et al., (1996,1997). Stratification of the charge is then stabilised through the action of the induction generated bulk rotational flows. The effective operation of the engine over a wide range of conditions is achieved by varying the start of injection (SOI) and injection pulse width according to the local air velocity and instantaneous piston position. Jackson et al., (1996) investigated the effects of air motion and fuel injection on the operating limits, fuel consumption and exhaust emissions of a part-load, homogeneous and full-load, stratified charge, wall-guided G-DI concept. The results indicated that stable homogeneous operation was possible over a relatively large injection timing range. Further retarding the end of injection (EOI) resulted in only a narrow band of stable stratified charge operation. An EOI operating window of 10 °CA was observed at 1500 rpm and 1.5 bar BMEP for WOT and 'leanest-burn-best-torque timings' (LBT).
In addition, differences in in-cylinder air tumble motion were reflected in ignition instabilities at the optimum EOI timing at 1500 rpm and 4.0 bar BMEP. Both effects highlight the dependence between the correct timing of fuel injection, with an established transport mechanism of airflow and/or piston interaction and the delay required for complete mixing and evaporation.

The transition from part-load, stratified charge operation to full-load, homogeneous is, in practice, discontinuous and requires a 'stepping' strategy to ensure stable engine operation. Tomoda et al., (1997) proposed a three-stage injection strategy whereby advanced electronic engine mapping permits a combination of both early and late fuel injection during the transitional phase. Whitaker et al., (1998), in an extensive study, compared four of the most current concepts of stratified charge, fuel only G-DI combustion systems under similar operating conditions. Each system incorporated a different approach to injector location, orientation and air motion.

Three of the engines employed side-entry intake ports: a Ricardo cylinder head with a central injector and central piston bowl; a second Ricardo cylinder head with a central injector and exhaust side piston bowl and a third Toyota D4 cylinder head incorporating inlet side injector and swirling piston type bowl. A fourth Ricardo configuration employed a top-entry intake port with inlet side piston bowl and inlet side injector location. Under lean operation, injection timing ranges for the Ricardo top-entry (tumble) and Toyota D4 (swirl) engines were considerably later and reduced in width when compared with the centrally located injector configurations. Whitaker et al., (1998) suggest that for the top-entry and D4 systems, timing range was dependent on piston position, and that an air guided central injector strategy could not support stable stratification. Conversely, the air-guided concept of Preussner et al., (1998) realised a 21% fuel economy improvement over a class leading MPI, under part-load operation. Unburnt hydrocarbon emissions where however significantly higher suggesting fuel deposits on the piston or cylinder bore surfaces.

The potential of operating with an unthrottled stratified charge concept is often compromised by practical concerns in the engine design. In particular, the catalytic conversion efficiency of exhaust products is of great concern. Part-load stratified operation produces non-stoichiometric exhaust conditions. The reaction zone temperature is generally greater than that experienced in pre-mixed, lean-burn MPI engines and the generation of the Oxides of Nitrogen (NOx) in this region is elevated. The high levels of NOx present in lean exhaust cannot be adequately reduced by a conventional three-way or close-coupled catalytic converter. NOx reduction is feasible through the introduction of high levels of EGR diluent. However WOT operation is marginally compromised as EGR requires an intake manifold depression achieved through throttle control. Current research efforts center on an alternative approach utilising de-NOx type catalytic converters. For full-load, homogeneous operation, current aftertreatment technologies are effective, as exhaust composition is approximately stoichiometric.

An additional concern, noted by many research groups, is the relatively high level of uBHC emissions found in both lean and stoichiometric exhaust emissions, indicating incomplete combustion. Fuel deposited on a hot combustion chamber, cylinder liner or valve surface is the primary source of unburnt hydrocarbons. This can only be minimised through an improved understanding of the interaction between the fuel spray, air motion and engine geometrical characteristics.
1.3. In-Cylinder Airflow Characteristics of Modern Gasoline Engines

In-cylinder air motion has long been known to influence the combustion processes within spark-ignited internal combustion engines. Semenov (1963) investigated unsteady flows using Hot Wire Anemometry (HWA) in a motored, two-valve, single cylinder, CFR engine with variable compression ratio. This early study identified the structures of intake and compression gas flows responsible for the generation of turbulence. Semenov (1963) concluded that the in-cylinder flow pattern varied greatly at different points in the engine cycle and from one engine cycle to the next. After inlet valve closure (IVC) he proposed that only re-circulatory motion of the gas could be possible and that the source of fluctuation in RMS velocity measurements during compression was due to the increased velocity gradients measured during intake. Semenov (1963) concluded that the dissipation of intake-generated energy into turbulent eddies was governed by the finite time available in the compression stroke and results confirmed that the fluctuations were proportional to engine speed but independent of compression ratio.

Since the early experimental work of Semenov (1963), much research work has been conducted in this field with the aim of identifying, optimising and modelling the features of both bulk fluid motions and velocity fluctuations that can be used to refine the combustion system and its stability (e.g. Glover et al., (1988b), Hadded and Denbratt (1991), Hill and Zhang (1994), Khalighi et al., (1995b), Whitelaw and Xu (1996), and Kang et al., (1997a)). The primary benefit of introducing rotational components of motion to in-cylinder flows is their potential to positively effect the combustion rate. The level of turbulence at the point of ignition (IGN) and through the combustion process (induced by combustion compression of unburned gas) governs the burning rate of the air-fuel mix. This is predominant in the initial burn or delay angle period, (IGN-10% mass fraction burnt) and exists, to some extent, in the main burn period, (10-90% mass fraction burnt), depending on the type of motion utilised. In typical MPI engines the initial burn is a critical factor as IGN-10% mass fraction of fuel is burnt in approximately 10 °CA and 50% mass fraction of fuel is burnt by approximately 20 °CA after top dead centre (ATDC) firing. In particular, lean-burn combustion systems operate lean, dilute mixtures under differing operating conditions and often with high amounts of exhaust gas recirculation (EGR). The combustion of lean mixtures is observed to proceed with a relatively low laminar flame burning velocity. This increases the overall combustion duration and the efficiency of the conversion of thermal to mechanical energy is reduced. Ignition timing advance is not a practical solution and will only serve to encourage flame development in weaker homogeneous mixtures and result in a decrease in mechanical advantage. Minimum-advance-for-best-torque (MBT) ignition timing increases with decreasing equivalence ratio (reciprocal of AFR). A spark ignition engine's highest efficiency is obtained at its knock borderline. MBT timings' tends to be very early for G-DI concepts when compared to more conventional MPI engines. As such, lean-burn combustion is increasingly susceptible to flame sensitivity and cyclic variations. Incomplete combustion due to misfire or knock (end-gas detonation) will inevitably lead to increased uBHC emissions.

The enhancement of turbulence at ignition through intake generated flow motion is a means by which the combustion rate can effectively be increased and the heat loss to the cylinder walls reduced.
Large-scale turbulence is required to promote diffusion and mixing of unburnt, reacting, quenched and burnt gas. Smaller scale turbulence is best utilised in enhancing eddy burning (turbulent transport) in the flame front. It should be noted that excessive levels of large-scale turbulence can be detrimental to the combustion process and can lead to flame detachment from the spark plug electrode. Previous studies have shown the initial burn rate to be an approximately linear function of turbulence intensity, which could be utilised effectively if conserved through the intake and compression strokes. Hadded and Denbratt (1991) observed a near linear relationship between turbulence intensity and delay angle measured at the spark plug gap for a stoichiometric mixture with 15% EGR. The straight-line curve approximation to the results had a gradient of -23 °CA per ms⁻¹. The relationship suggests that small variations in turbulence intensity could significantly effect the delay angle. However, it should be noted that whilst the correlation was adequate for EGR, lean mixtures with excess air did not behave in the same manner due to different local mixture compositions and laminar flame speeds.

In practice, turbulence generation is achieved through the conservation of rotational bulk flow structures, generated in the induction stroke, which provide energy near top-dead centre (TDC) firing for the generation of turbulence. Suitable inlet port, inlet valve and combustion chamber geometries can be implemented to augment the turbulence level through conservation of the bulk flow. Pent-roof combustion chambers and piston crown bowl geometries have been favoured as potential solutions for effective turbulence generation in lean-burn engines. With many piston bowls, the distance that the flame can travel before quenching is effectively increased by the 'open-bowled' or re-entrant combustion chamber geometry.

Therefore, in summary, the important features of an in-cylinder flow field that must be identified if potential gains are to be realised:

- The magnitude and direction of the in-cylinder mean airflow velocity over the induction and compression strokes and particularly at the spark plug gap.
- The high fluctuation intensity of the mean airflow during the induction and compression strokes and in the vicinity of the spark gap at the time of ignition.
- The contribution of periodic, unsteady and non-stationary turbulent airflow with a potential for cycle-to-cycle variations in the mean and RMS velocities.

1.3.1. Features of In-Cylinder Mean Gas Flow Motion and Turbulent Breakdown Mechanisms

In general, two types of organised, rotational, in-cylinder charge motions act singularly or in combination to achieve optimum local flow conditions at spark ignition. These are defined by their axis of rotation in the cylinder. Rotational motion about an axis parallel to the cylinder axis is referred to as swirl motion. Air rotation in any plane perpendicular to the cylindrical axis is known as tumble motion. Figures 1.5. and 1.6. illustrate the principal in-cylinder, organised, rotational gas motions and their conventional directions as utilised in IC engines. A combination of the two bulk flows can produce effects such as inclined tumble, although in practice, a combination of both types of flow is always present.
Swirl and tumble motions are generated in the intake stroke during the period of intake valve opening. Intake flows are generally characterised by an annular jet flow through the inner valve seat area (valve curtain), which forms two toroidal vortices above and below the valve head. Steep velocity gradients with high shear are observed with zones of re-circulation behind the valve face. The in-cylinder gas motion can be generated by a variety of methods including intake valve shrouding and masking, port geometry and orientation, port directional vanes, valve deactivation, and port control valves. The port flow coefficient can however be compromised when port flow restrictions are introduced. It is often useful to define the strength or 'intensity' of swirl and tumble for the purpose of comparison. Generally a ratio of the angular velocity of inlet charge at the end of induction to the angular velocity of the engine crankshaft is used, although many interpretations exist.

A third in-cylinder gas flow, primarily used in Diesel re-entrant bowl type combustion chambers, has been employed to promote late compression turbulence generation in gasoline engines. A radial, inflow 'squish' gas flow from exhaust valve to intake valve side was effected by Kume et al., (1996) in a reverse tumble, top-entry G-DI with an inlet side spherical piston bowl, central spark plug and injector placed between the intake valves. Approaching TDC firing, the volume of gas located on the piston topland is rapidly compressed against the cylinder head and shears into the compact, spherical combustion chamber. Such squish motion interacts with the bulk motion and promotes turbulent breakdown of the conserved tumble. Evans et al., (1996) utilised 'squish' jets forced through channels integrated in the piston crown to successfully increase the burning rate of lean-burn SI combustion with natural gas. Kume et al., (1996) claimed that during the initial period of expansion following TDC firing, 'reverse squish' out of the combustion bowl could be used to enhance flame propagation towards the exhaust side of the cylinder head in their combustion system.

The effect of turbulent gas flow on combustion in IC engines has been a topic of intense research for many decades. The turbulence prediction and modelling in motored or fired engines requires an extension of the physical turbulence models to include the unsteady, periodic and non-isotropic nature of the flow (e.g. Moriyoshi et al., (1993a)). Validation of such models through experimental measurements by Hot Wire (HWA) or LASER Doppler Anemometry (LDA) can be subjective and dependent upon the analysis technique applied (e.g. Witze P.O. (1980), Glover et al., (1988a, b), Fansler and French (1992), Ruck et al., (1993)). The generation or enhancement of in-cylinder turbulence has been identified by three main mechanisms; free shear strain, wall shear strain and compressive strain (Hill and Zhang (1994)). As previously mentioned, the intake jet flows generate high shear strains during the inflow period. Steep velocity gradients, vortex shedding, vortex interaction, and flow instabilities, (e.g. Nadarajah et al., (1998)), contribute to relatively high levels of RMS turbulence intensity. Vortices shed during intake events can be transported through the gas by diffusion or convection and are likely to be carried over into the compression stroke. After IVC a coherent in-cylinder flow structure should be established and RMS velocity values are seen to decrease with increasing crank angle.

Where swirl flow systems are used, the compressive loads acting on the gas motion through the upward piston motion increase the viscous drag on the cylinder walls and reduce the angular momentum of the rotating mixture.
'Pure swirl' systems can be modelled by solid body rotation dynamics. Entraining the swirl flow in a piston bowl effectively increases its angular velocity and generally near TDC firing, a radial squish inflow is used to generate high shear strain and turbulence. A feature of swirling flows however is that some mean rotational motion is generally observed close to TDC firing.

Intake generated tumble flows generally exhibit more complex three-dimensional flow fields and are considered more effective for the generation of in-cylinder turbulence at ignition. In four-valve configurations, pairs of forward, outwardly looping, tumble vortices have been observed to descend to the piston, re-group centrally and ascend vertically as shown schematically in Figure 1.7. In certain tumble systems it is not unusual to observe secondary counter-rotating vortices which can collide with the main tumbling vortices.

The mechanism by which these coherent tumbling flow structures breakdown is not wholly clear, but is thought to be more dependent on the proximity of the piston to the cylinder head and the efficient conservation of the rotational flow, than in a swirl system. If a tumbling structure exists at bottom-dead centre (BDC), then it is likely that it will experience an increase in angular velocity through the compression stroke. The distance between the piston and cylinder head will always limit the radius of gyration of the rotational motion. Furthermore, the forced distribution of rotational energy under compression, along the axis of rotation, will be limited by the bore diameter. Constant volume, short cylinder, transient, experimental measurements, such as those of Riahi and Hill (1993) showed that mean tangential velocities and turbulence intensity were strongly dependent on chamber length to diameter ratios. Conservation of angular momentum, in the absence of strong viscous wall shear stresses will increase the tumble velocity as the radius is decreased. It is often not possible to determine the point at which this rotational vortex collapses due to the limitations of the experimental techniques employed or the ambiguities in the interpretation of measurements.

Observations and numerical studies suggest that pressure effects in late compression rapidly increase the shear stresses on the piston surface, effectively destroying the vortical structures. Decay of a tumbling structure is more rapid than that of swirl motion through the increased wall friction effects due to geometry. Additionally, turbulence generation on curved walls is considerably increased giving higher turbulence levels at TDC firing. This effect will be more pronounced in the highly curved, concave geometry of typical G-DI combustion chambers at TDC. In an experimental study, Arcoumanis et al., (1990) used a motored two-valve engine and observed reduced levels of axial velocities but increasing RMS velocity values at the piston surface at 300 °CA. In addition, radial velocities measured along the cylinder axis close to the piston surface were greatly amplified. At 330 °CA, turbulence intensity was approximately equal to the mean piston speed (MPS), whilst axial velocities tended to zero. Breakup is likely to occur by the division of large vortices into smaller vortices and eddies. Whilst the experimental results and their interpretation are varied, it would suggest that the process duration could be between 25-35 °CA depending upon the type of engine and engine conditions investigated. Jaffri et al., (1997b) observed the onset of this breakup as early as 269 °CA by the analysis of experimentally determined turbulent kinetic energy iso-surfaces in a four-valve pent-roof chamber. Their analysis identified the presence of a tumble flow, 'turbulent kinetic energy ring' towards the end of the intake stroke. Hadded and Denbratt (1991) measured a bi-modal generation of turbulence in compression in four tumble-generating cylinder heads in a four-valve pent-roof, motored engine.
A first peak was observed at approximately 110 °CA BTDC and a second, at approximately 20 °CA BTDC. Their results suggested a 'square-topped' tumbling motion was present at close to IVC as near zero velocities were recorded in the spark plug vicinity. They proposed that a limiting turbulence value would be reached for the configuration used at the point of ignition. Increasing tumble ratio served only to increase the primary bi-modal turbulence peak but not effect the secondary peak at around ignition timing. For their particular configuration, any increase in rotational kinetic energy due to increased tumble ratio was absorbed in the primary RMS velocity fluctuations at approximately maximum piston turn-around speed and the greatest rate of change of angular inertia about the tumble axis. Kudou et al., (1992) investigated the use of tumble control to improve combustion stability in a motored four-valve engine at 500 rpm using shrouded valves with varying angles of incidence. The tumble vortex centre was seen to move towards intake during induction and then towards exhaust side during the compression stroke.

As demonstrated by other experimental studies; (e.g. Hadded and Denbratt (1991), and Kang et al., (1997)), the cyclic fluctuation of the mean velocity was seen to be lower for the high tumble ratio cylinder heads in the latter stages of the compression stroke. Kang et al., (1997) assumed that a single vortical structure was present in a vertical plane through the mid-cylinder axis and expressed the nature of the tumbling flow by the mean vorticity calculated over the measurement plane. They suggest that a higher level of intake tumble generation will lead to a single peak in mean vorticity at some instant before TDC. The temporal location of this peak is a function of the rate of increase in angular velocity during compression. For higher peaks, the rate of decrease associated with bulk motion break-up is greater and the breakdown process will be seen to occur at an earlier point before TDC firing. They did not observe the bi-modal effect reported by Hadded and Denbratt (1991). As with previous work, they concluded that the rapid decay is due to the complex 3-dimensional nature of the flow, especially prevalent in four-valve configurations, as illustrated in Figure 1.7. This is often difficult to quantify experimentally due to the limitation of current simultaneous plane measurement techniques.

Experimental work by Kang et al., (1996, 1997) performed in a four valve, pent-roof, lean-burn engine, recently quantified the effects of tumble flow on combustion duration by the analysis of heat release using in-cylinder pressure profiles. They compared three intake port configurations, with entry angles of 15°, 20° and 25°, of increasing tumble ratios, with the aim of improving the lean operation of the engine. Their results showed a rapid decay of the strongest tumbling flow configuration (25°) from the end of the intake stroke to a volume averaged turbulence intensity of 0.4 times the MPS close to TDC firing at 1000 rpm. The 15° and 20° angled port configurations demonstrated a conservation of the tumbling motion through the compression stroke up to a value of 0.5 times the MPS under similar conditions. The volume averaged turbulence intensity for the 25° inclined port was initially highest during the early intake stroke but decayed rapidly during compression with the 20° port maintaining the highest level near typical lean ignition angles.

Arcoumanis et al., (1998) observed radial RMS turbulence intensity values which suggested a tumble breakdown of up to 1.5 times the MPS at 320 °CA in the spark plug region in a similar engine, again at 1000 rpm. Hadded and Denbratt (1991) reported values between 0.7 and 1.0 MPS in the direction of forward tumble motion, and values between 0.9 and 1.2 MPS in the cross-tumble plane, at the spark plug gap.
These values had been adjusted to account for the effects of cycle-to-cycle variation in the mean flow. All three studies observed the precession of a coherent tumble vortex with descending piston motion. Arcoumanis et al., (1998) noted that the instantaneous tumble centre was always lower than the instantaneous cylinder centre when observed along the cylinder axis. These studies suggest that intake generated tumble flow structures can successfully be utilised in lean-burn SI engines but may require additional geometrical enhancements to increase the efficiency of conservation of rotational angular momentum in ultra-lean stratified charge strategies.

The most common modifications to in-cylinder geometry are to the piston crown. Platts et al., (1996) successfully extended the lean limit of an open chamber stratified charge combustion system (OCCS) by replacing a flat piston with a bowled geometry. Importantly, they observed experimentally using Hot Wire Anemometry (HWA) that the tumbling vortex was sustained for a longer period of time whilst maintaining stable stratification, in the bowled piston configuration. Furthermore, the role of squish flow in turbulence enhancement in these tumble systems is generally similar to that discussed for swirl systems. Where suitable geometries are employed, squish flow can be used at TDC to enhance mean flow breakdown by providing a diametrically opposing jet flow.

1.3.2. Cyclic Variability in In-Cylinder Mean Gas Flows

The reciprocating nature of engine operation induces a cyclic variability in the mean, gas flow. These non-stationary, periodic fluctuations (as distinguished separately from 'statistically stationary' flows) can detrimentally affect combustion stability and have been the subject of much deliberation in the literature (e.g. Glover et al., (1988b), Moriyoshi et al., (1993b) and Hong and Chen (1997)). It is generally accepted that the contribution of cyclic variations to 'ensemble averaged' experimental measurements of air characteristics can be significant.

The instantaneous velocity, \( U(t) \), as measured by experimental techniques, can be decomposed into a mean, \( \langle U \rangle \) and fluctuating component, \( u(t) \) for stationary, turbulent flows:

\[
U(t) = \langle U \rangle + u(t)
\]

For in-cylinder fluid flows, the mean velocity is additionally dependent on time or crank angle. The mean velocity component for a given crank angle, \( \theta \) is then averaged over an 'ensemble' of engine cycles, such that instantaneous velocity in the \( i^{th} \) cycle is given by:

\[
U(\theta, i) = U_{EA}(\theta) + u(\theta, i)
\]

where \( U_{EA}(\theta) \) is the ensemble averaged mean velocity, and \( u(\theta, i) \) is the non-stationary, turbulent velocity fluctuation.
\( U_{EA}(\theta) \) is given by:

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

where \( \Delta\theta \) is the measurement window width in crank angles, \( N_c \) is the total number of measurement cycles and \( N_i \) is the total number of measurements within a crank angle window of width \( (\theta \pm \frac{\Delta\theta}{2}) \) (Hong and Chen (1997)). By the same analysis of Hong and Chen (1997), the ensemble-averaged RMS velocity fluctuation can be defined:

\[
u'_{EA}(\theta) = \left( \frac{1}{N_i(\theta)} \sum_{i=1}^{N} \left[ U(\theta \pm \frac{\Delta\theta}{2}, i) - U_{EA}(\theta) \right]^2 \right)^{0.5}
\]

The ensemble turbulence fluctuation intensity includes all high and low frequency fluctuations from the ensemble average within a given crank angle window, over a given number of cycles. Glover et al., (1988), Hilton et al., (1991), Corcione and Valentino (1991) and Hong and Chen (1997) amongst others applied filtering methods to estimate the individual, low frequency, cyclic variations in mean velocity, separate from the high frequency turbulence. The instantaneous velocity for a given crank angle, within a given cycle, is then composed of an individual mean velocity and turbulent velocity which changes from cycle to cycle and is dependent on the filtering method and arbitrary choice of cut-off frequency employed. The application of these methods to in-cylinder gas flow is covered in greater detail in Chapter 2.

Knowledge of the characteristics of cycle-to-cycle variation is particularly relevant to tumble flow strength, which has an observed relation to cyclic variations in mean gas flows. Variations can result in different gas mean flows and fluctuation intensities at the spark plug gap generated from similar quasi-steady derived bulk flow structures at IVC. Glover et al., (1988b) claimed that cyclic variation found in in-cylinder measurements was mostly attributable to the instability of the axis of rotation (tumble axis ‘tipping’) and not from random differences in intake flow structures. Hadded and Denbratt (1991) and Ando (1996) concluded that high tumble ratio ports showed reduced cross-tumble RMS velocities at IVC. These configurations were less likely to illustrate poor correlation between measured mean flow values in different cycles. As mentioned previously, part-load, stratified charge operation relies on good spatial and temporal repeatability of the air-fuel mixture. However, Hadded and Denbratt (1991) found no correlation between mean flow cyclic variations and combustion variability in a four-valve pent-roof, flat piston engine using scanning LASER Doppler Anemometry (LDA).

However their findings revealed that RMS turbulence measurements could be as much as 30% overpredicted if cyclic variations were not removed. Corrected values of turbulence intensity in the spark plug region indicated an apparent isotropy in the flow between the orthogonal planes of measurement, which otherwise would not have been evident. This is contrary to the findings of Ozdor et al., (1996) based upon a series of measured pressure-related parameters to assess motored and fired engine cyclic variability.
Motored cyclic variations in charging efficiencies were quantified as between 0.5 and 2.0% of the maximum in-cylinder pressure, with maximum values obtained at part-loads. Such cyclic variability was considered sufficient to effect the intake generated flow pattern as well as ring leakage.

1.3.3. Strategies for In-Cylinder Airflow in G-DI

Wirth et al., (1996) highlighted the partially opposing requirements of a G-DI in-cylinder flow concept. The flowfield must support the mixture preparation and also maintain the stratification of in-cylinder products. In lean, stratified charge combustion the ignition source is a singularity and is not initiated at multiple sights as in heterogeneous combustion systems. In order to achieve this, the in-cylinder air motion and fuel spray quality must maintain spatial and temporal repeatability of the air-fuel mixture from cycle to cycle. Numerical modelling and experimental investigations of the effects of such in-cylinder flow structures on combustion performance have been widely reported in the literature (e.g. Henriot et al., (1991), Hadded and Denbratt (1991), Whitelaw and Xu (1996) and Hill and Zhang (1994)). Khalighi et al., (1995a,b) reasoned that the propensity for tumble over swirl flow to break down into late compression turbulence in typical four-valve, pent-roof combustion chambers meant that it would act directly on the early stages of flame development. Swirl flow might store energy better through TDC for late burn. Their burnrates correlated best with the tumble based systems. This indicated that early flame kernel development under suitable mean convective and turbulent conditions is central to the combustion stability.

The contribution of both types of in-cylinder gas flow to combustion stability have been recognised and implemented in homogeneous, lean homogeneous and stratified charge systems. Swirl systems generally aim to envelop injected fuel in a solid central vortex whilst minimising fuel spray impingement on the piston and cylinder walls. For example, the direct injection systems of Harada et al., (1997) and Nogi et al., (1998) were designed to achieve ultra-lean, stratified, combustion at partial loads by the use of variable swirl control. Harada et al., (1997) utilised a three-stage injection strategy to overcome regime 'stepping'. A helical intake port was used to generate swirl motion while a second straight port was fitted with a swirl control valve (SCV). Partial throttling of the SCV enabled the swirl intensity to be varied according to the mode of operation of the engine. This method of mixture preparation is effective in partially disassociating the strength of the swirl motion from engine speed. A similar approach was adopted by Ohsuga et al., (1997), utilising a control valve fitted downstream of the throttle. Intake air velocity and the intensity of tumble could be varied with engine operation. Fuel spray penetration was reduced in order to maintain a rich fuel mixture in the upper part of the cylinder, below the centrally located spark plug, reducing heat transfer to the cylinder walls. Meyer et al., (1997) employed a downward air tumble approach with an internal EGR swirl valve to obtain stratification between the air-fuel mixture and exhaust gas diluent. They found that the utilisation of the swirl motion to direct exhaust gas around the cylinder wall and conserve the air-fuel mixture in a central column was limited by engine speed where the mixture tended towards homogeneous. Earlier swirl-based systems had limited operating zones where swirl momentum was proportional to engine speed. In these types of systems it was often difficult to complete combustion due to inadequate mixing.
As a result, many research teams directed their efforts towards tumble-based concepts that provided the potential for a wider range of operation. As previously discussed, tumble flow in SI engines has long been recognised as the most effective means of enhancing turbulence intensity at the spark plug gap and promoting good mixing of the incoming charge.

Tumble breakdown during the compression stroke into mean flow and smaller scale perturbations and eddies provides suitable conditions for initial flame kernel growth whilst the mean fluid motion is thought to enhance the propagation of the flame across the combustion chamber. An increase in the tumble ratio during the intake stroke has been shown to cause an earlier breakdown of the main vortex in the compression stroke leading to elevated levels of turbulence intensity at TDC firing. A stratified charge concept can only be achieved if a suitable in-cylinder gas motion is conserved throughout the compression stroke. Better stratification stability will be achieved if the mean flow can be preserved for as long as possible. In 1993, the Mitsubishi Motors Corporation proposed the 'Three Layer Barrel Stratification' system, as applied to an MPI engine, utilising such a method (e.g. Kuwahara et al., (1994). Tumble angular momentum was conserved by the use of a 'tumble control piston' that employed a surface curvature that ran parallel to the circumference of an intense tumble flow pattern. Intake port flow guides were utilised to direct intake jets in such a manner that three-layer, tumble stratification was achieved. A central rich air-fuel mixture was stratified between two outer, lean tumble layers. Motion in the direction of the tumble axis, or cross-tumble, was minimised. Late compression bulk flow was suppressed by an opposing squish flow from the intake side which, in theory, additionally aided in the generation of a uniformly distributed turbulent flow field. A similar 'three layer' numerical model proposed by Meng et al., (1996) was found to be unstable in late compression and unsuitable for supporting a stratified charge.

Subsequently, many tumble type G-DI designs utilised modified port, chamber and piston geometries to preserve the intake generated flow field. Tumble could be generated effectively through orientation of the intake ports. Generally, side-entry ports produce a forward tumbling vortex that follows the combustion chamber roof and descends the exhaust-side cylinder wall. Further motion across the moving piston surface and up the intake-side wall completes the loop and at this point, the inlet valve should be closed. The re-circulatory motion in forward tumble systems is towards the intake valve side. The tumbling air systems of Kume et al., (1996), Jackson et al., (1996), and Iwamoto et al., (1997) utilise a reverse tumbling concept whereby air is introduced to the cylinder via upright, curved intake runners and motion is directed towards engine exhaust side. Figure 1.6. illustrates the configuration used to achieve forward and reverse tumble. In both cases, intake port entry angle, valve masking or shrouding and intake port flow vanes can be used to control the induction tumble conditions.
1.3.4. Interactive Effects of Direct Liquid Fuel Spray Injection and In-Cylinder Gas Flow Structures

It is evident that the in-cylinder air motion in G-DI engine combustion systems must attain a coherent, stable and repeatable structure. This is not a factor that can be considered in isolation and the suppressive or deflective effects of direct fuel injection at high velocities on the flow pattern are not insignificant. For systems employing port fuel injection (PFI), the flow entering the cylinder is already a mixture of air and fuel. The in-cylinder flowfield is thus always due to the interaction between the two components. In a G-DI engine, an airflow field is established first and then fuel is injected into the cylinder. The direct injection of fuel from a high-pressure injector can impart significant momentum to the surrounding air and influence the preparation of the air-fuel mixture. This can be most significant where air motion is reduced as in the case of late fuel injection in the compression stroke. For injection during the intake stroke, Han et al., (1997) observed deformation and redirection of the fuel spray using computational modelling in the KIVA-3, commercial, computational fluid mechanics (CFD) code, with a central injector in a homogeneous, Ford DISI chamber geometry. This effect was most pronounced in the region of gas jet-like flow between intake valves.

Airflow and fuel spray measurements were performed by Stanglmaier et al., (1998) in a motored side-entry, modified four valve GM Quad-4 engine, with forward tumble and a central injector. Their experimental airflow results with and without fuel injection showed distinct differences in velocity and turbulence intensity magnitudes through the injection event and the middle stages of compression. As observed in previous studies, tumble velocities were seen to decrease during the compression stroke and the magnitude of tumble was greater for the non-injection case. Mean axial velocities with injection were reported 1-2 ms$^{-1}$ lower in value on the mid-centreline plane, except near the intake valve.

Han et al., (1997) had also observed a significant suppression of the intake-generated tumble by a spray-induced flow. From 240 °CA, following an intake stroke injection, velocity fluctuations increased at a greater rate than for the non-injection scenario, suggesting a ‘damping’ effect due to the spray momentum being partially transferred to the surrounding gas. The reduction in tumble intensity persisted longer into the compression stroke for the injection case. Importantly, Han et al., (1997) found that retarding SOI later than 150 °CA could be utilised to provide higher turbulence intensity at TDC firing. They reported that injection could increase turbulence intensity by up to 10%, from 0.6 to 0.7 times the mean piston speed (MPS), over a similar MPI engine. Stanglmaier et al., (1998) suggested that the injection event introduced significant out of plane flows, not readily detectable with their experimental apparatus. The high fluctuations following an injection event could not be accounted for by the cyclic variations in the mean flow but were large enough to contribute to cycle-resolved turbulence estimates.
1.4. Fuel Spray Characteristics for Direct Injection Gasoline Engines

Gasoline direct injection strategies that result in fuel injection later in the compression stroke require a FIE that can deliver high fuel pressures and a finely atomised spray. This enables the system to cope with the elevated pressures and temperatures that occur at this stage in the cycle and compensates for the reduced mixture preparation time. In addition, injection spray cone angles and penetration lengths must be optimised to ensure minimal wetting of the combustion chamber, cylinder walls and intake valves. Such systems must be both viable and ergonomically packaged and additionally should not ‘parasitically’ load the engine. Electronically controlled common rail (constant fuel line pressure) FIE and high-pressure fuel injectors can achieve these targets but the key issues of mating the injector selection with the type of combustion system adopted still need to be resolved.


G-DI fuel injection pressures between 4 and 12 MPa have been employed whereas typical MPI pressures are often found in the range of between 270 and 450 kPa. Current commercial Diesel needle opening pressures (NOP) can conservatively be estimated in the range of 50 to 140 MPa. The spray structures and atomisation qualities (droplet size and velocity) across these ranges differ greatly and are matched to the combustion strategies employed. In addition to differing injection pressures, ambient temperature and pressure conditions as well as fuel properties contribute to varying droplet drag and evaporation regimes. These features are not readily transferable to G-DI type injectors. The performance of MPI and Diesel injectors, conceived specifically for optimum full-load or part-load operation, are not capable of achieving the requirements of early or late direct injection stratified charge mixture preparation.
The following section refers to methods of describing spray structures using established experimental techniques applied to cold, non-evapourating fuel sprays. As such, the processes of heat and mass transfer are not included. Both steady state and transient fuel spray analyses have been reviewed.

Diesel fuel spray research has identified several parameters that can be used to describe a fuel spray. These can be readily applied to high-pressure, gasoline fuel sprays for use in suitable mixture preparation strategies in G-DI engines. These are often described for a Diesel fuel spray as the spray tip penetration, break-up length, angular spread or dispersal and the atomisation quality (droplet size and velocity distributions). The injection characteristics of these types of fuel sprays are generally described for either static, continuous (steady) flow or pulsed operation at near-constant ambient pressure conditions. Steady flow droplet distributions generally tend to exhibit a much wider spread of droplet diameters, with reduced mean diameters, in single hole, pintle type, fuel injector sprays (Kelly-Zion et al., (1995)). A clear distinction must be drawn between the features of steady or fully developed spray regions and transient, developing, unsteady fuel sprays.

Automotive fuel sprays are unsteady and as such there is an additional temporal dependency that must be considered when evaluating spray characteristics. The pressure differential between the instantaneous ambient air pressure and the fuel delivery pressure at start of injection (SOI) has an approximately constant ratio in each cycle. This will change shortly after SOI due to transient effects on the fuel pump delivery pressure over the injection duration, changes in the liquid volumetric flow rates with increased needle lift and the pressure history in the injector. When injection pulse widths are reduced, the initially unsteady spray may only attain a steady state for a very brief instant before needle closure. G-DI engine injection strategies often require fuel to be injected at different points in the engine cycle with varying in-cylinder pressure. The resultant spray structures are in addition dependent upon the temporal variation of the pressure differential.

Based upon the above observations, it is possible to identify the key features of G-DI fuel sprays and injector functionality that require investigation:

- Fuel spray formation, rate of formation, break-up length, spray tip penetration, spray patternation, symmetry and dispersal properties.
- Fuel injection operating pressure and its effect on fuel spray characteristics (e.g. spray pattern, liquid droplet diameters, liquid droplet velocities etc.)
- Constraints imposed by fuel injector location, orientation and inclination on a stable spray pattern of suitable geometry.
- Suitable matching to the in-cylinder air flowfield and its effects on spray characteristics.
- Fuel spray qualitative and quantitative, spatial and temporal, repeatability.
- Fuel injection stability (dynamic flow linearity) to duty cycle.
- Fuel injector fast response to reduced mixture preparation and recovery times.

For full-load, direct injection operation, a well-dispersed spray, with relatively low penetration is required to ensure good charge homogeneity and heat transfer, as well as low emissions of unburnt hydrocarbons. There is unlikely to be a restriction on the time available for mixture preparation and thus atomisation plays a less crucial role.
In contrast, and depending on the type of G-DI engine being investigated, it is generally accepted that part-load operation should best be achieved with a finely atomised, compact and deeply penetrating spray. Rapid atomisation of the liquid fuel into fine droplets will increase the total liquid surface area and hence favourably improve the rate of fuel evaporation. Both the pressure differential and the design of the nozzle will influence the injection system's ability to achieve these aims.

Many different types of injector and nozzle designs have been regarded as potentially suitable for two or three mode G-DI injection strategies. Some of these studies are presented in Table 1.0 along with the injector types and nozzle designs evaluated. Salters et al., (1996) and Tomoda et al., (1997) investigated plain-type exit orifices producing solid cone sprays. Generally, plain exit orifices require higher pressures to achieve finely atomised sprays and typically exhibit narrow spray angles. Miyamoto et al., (1996), Meyer et al., (1997) and Hoffman et al., (1997) assessed the suitability of twin-fluid, air-assisted atomisers with divergent nozzles. Fraidl et al., (1996) proposed a direct mixture injection (DMI) injector with a small mixture pre-chamber. In a comprehensive experimental and computational study, Pontoppidan et al., (1997) examined the characteristics of three types of direct fuel injection (DFI) injector nozzle designs from a manufacturer's perspective. This work was conducted in an optically accessed chamber at 0.01 MPa below atmospheric pressure and an ambient air temperature of 298 K. The injectors tested were as follows:

- a solid-cone, divergent, pintle type
- a hollow-cone, smooth pintle, swirl type
- a closed cap, smooth pintle, multi-jet injector.

The results of this investigation and of others, (e.g. Wirth et al., (1996) and Tomoda et al., (1997)), concluded that a hollow-cone, pressure-swirl type of injector could best be adapted to a stratified charge application, with low penetration velocities and reduced droplet mean diameters.

Experimental evidence indicates that the distribution of large droplets (which are undesirable) tends to be along the spray centre axis in solid-cone, pressure-swirl injectors. Smaller droplets are found along the spray periphery. This is contrary to that observed in hollow-cone spray structures, where the majority of droplets are concentrated within the annular ring that forms downstream from the nozzle exit. The lower momentum of such a spray when compared to the solid cone type would suggest that using this type of injector would increase the dependency on the in-cylinder air motion to deliver optimal air-fuel mixtures to the spark plug. Pontoppidan et al., (1997) and others concluded that such an approach to mixture preparation would be dependent on injector and combustion chamber design. Better spray stability properties were achieved by the use of retracting or inwardly opening needle designs opposed to outward opening types.

These injectors are sometimes regarded as exhibiting the characteristics of a multi-jet injector with an infinite number of injection holes, whereby fuel is distributed evenly over the surface of a cone (Zhao et al., (1997)).

The following review relates to the characteristics of high-pressure FIE and single-fluid, hollow-cone, pressure-swirl type injectors as used in the experimental work described in the following Chapters. Figure 1.8. illustrates the swirling liquid flow pattern and angular distribution for a typical hollow-cone, swirl-type, injector nozzle geometry with an inwardly retracting needle design. For the purpose of this thesis, the parameters that can best be used to describe high-pressure, non-evapourating, hollow-cone, swirling sprays in G-DI combustion chambers can be placed into two categories: those that pertain to the spray distance travelled or spray tip penetration in simultaneous axial and radial directions (sometimes referred to as the global parameters in the literature) and those that indicate the spatial and temporal 'quality' of the spray pattern (local parameters). It should be noted that both categories are intrinsically linked and cannot be considered in isolation.

1.4.1. Spray Tip Penetration
The spray characteristics of relatively low pressure MPI injectors have been reported in the literature, but as of yet little experimental data exists on the spray penetration characteristics of high-pressure, swirl-type injectors. As such, this field has been identified as a new and important research area (Wirth et al., (1996), Zhao et al., (1997), and Comer et al., (1998)).

The spray tip velocity or penetration is a complex function of injector nozzle design (typically the length to diameter ratio of the nozzle), injection pressure (volumetric flow rate) and ambient air pressure. These in turn affect the injector internal flow profile, liquid fuel break-up length and droplet dispersal, as observed by Hiroyasu et al., (1995) in two plain-type orifice nozzles with water. Atomisation of liquid fuel is unlikely to occur at the nozzle, but through a droplet generation mechanism downstream of the injection orifice. The experimental fuel spray studies of researchers such as Pitcher and Wigley (1992a) and Cousin et al., (1998) suggest that fuel exits the nozzle initially as a continuous liquid column with only the jet periphery finely atomised, as a developing multiphase mixing layer, through the action of strong interfacial gas shear forces. Sinuous distortion of the jet region, through the effects of gas shear flow and the propagation of internal radial forces within the liquid, initiate 'wave-like' dilatational instabilities in the surface of the liquid. The mode and wavelength of the interfacial waves is governed by the film thickness, density and surface tension (Pitcher et al., (1990) and Dementhon and Vannobel (1991)). Wavy atomisation or Rayleigh break-up is dependent on the ambient pressure, relative velocity, viscosity, fuel injection rate and the quantity of fuel injected. The wavy structures are stretched and through necking, stripped from the jet core under the action of shear, leading to the formation of elongated ligaments of fluid, holes in the liquid core and individual satellite 'parcels' of droplets. At a given point downstream, the ligaments will finally break-up into droplets through gas shear as the inertial and viscous forces exerted on the liquid overcome the surface tension forces. This distance is characteristically termed the 'break-up' length and is dependent upon the nozzle geometry. Individual droplets are then formed into a fully-developed spray pattern as the jet core becomes thinned, unstable and unable to sustain a coherent form.
Evidence from the experimental studies of Pitcher and Wigley (1991a) suggest that wavy atomisation was responsible for superimposing a wavy structure on the main droplet velocity and size profiles in the near-nozzle region of a Bosch DSLA type, direct injection, single hole, Diesel injector. Four main 'wave peaks' were observed in droplet concentration under steady state operating conditions, atmospheric ambient pressure and a NOP of 220 bar. Measurements performed in the combustion chamber of a modified Jenbach engine (unsteady conditions) also exhibited a wavy pattern.

Along the envelope of the spray plume, liquid accelerates the local gas boundary layer from a laminar to turbulent state. The rapid transition in boundary layer thickness creates local regions of gas re-circulation and a toroidal vortex is often formed, propagating from the near nozzle region with the spray tip leading edge. These opposing gas velocity gradients are responsible for entraining relatively small liquid droplets and enhancing gas flow back into the spray plume (Iwamoto et al., (1997)). Pontopiddan et al., (1998), Park et al., (1998) amongst others, observed local regions of counter-rotating backflow in the outer parts of a gasoline spray using shadowgraphy and LASER light sheet photography techniques. In a hollow-cone, fuel spray, this entrainment mechanism is thought to contribute to the collapse of the cone structure and can be seen to start close to the orifice exit. The preceding, vortical structures entrain more gas and strip liquid from the outer surfaces of the cone. The enlarged entrainment vortex rejoins and interacts with the spray plume, disrupting the hollow, conical form and dispersing droplets. Hoffman et al., (1997) claimed that the collapse of the hollow-cone occurred below the top re-entry point of the toroidal re-entrainment vortex ring. Briefly the toroidal ring can be seen to exist in isolation before complete dispersal. The subsequent spray plume was described as a dispersed cylindrical plume. Hoffman et al., (1997) and Comer et al., (1998) suggest that the vortex may contribute to the collapse of the cone through widening of the spray walls, even though the spray cone angle remains approximately constant.

Transient fuel sprays have been divided into three regions in order to correlate data from both experimental and computational investigations. Most of the experimental techniques employ fixed position measurements and record the temporal passage of the spray through a given point. In their simplest form, these aim to divide the spray up into a leading edge (or spray head or tip), a central section (spray body) and a trailing edge (spray wake or tail). The characteristics of each section are often considered in isolation and importantly, differ spatially throughout the spray (Brunello et al., (1991), Pitcher and Wigley (1991a), and Long et al., (1996)). The spray head is typically characterised by very high velocities, unsteadiness, reduced droplet number densities and a broad droplet size range. However, PDA measurements performed near the nozzle exit have shown an absence of discrete droplets in many experimental studies (Park et al., (1998)). It is likely that the lack of data indicates the presence of elongated ligaments or non-spherical droplet parcels in the liquid jet that are not readily detectable as valid PDA signals. The high levels of photomultiplier tube (PMT) activity observed support evidence for this (Pitcher and Wigley (1991a)).

During the next stage, most of the fuel is injected and the flow is seen to reach a quasi-steady state. The droplet diameter range becomes much narrower. The trailing section shows a relatively slow decay of droplet velocities with both small and large sized droplets in large numbers.
The smaller droplets may have originated in the spray body, but their reduced momentum means that their arrival is delayed. Brunello et al., (1991) suggested that the larger droplets were generated during late injection where fuel delivery pressure was reduced. It should be noted that close to the nozzle orifice there is considerable deviation from the above, simplified model.

In a pressure-swirl injector, the rotating liquid exits the orifice in the form of a cylindrical jet but rapidly forms an approximately axisymmetric, hollow, conical sheet. The observed spray cone angle immediately following SOI is very small when compared to needle or orifice angles (Yamauchi et al., 1996). The characteristics of this liquid sheet (thickness and velocity components) at the exit orifice will define the spray break-up length and atomisation quality. These parameters will vary with needle lift from SOI. Cousin et al., (1998) realised that with reduced injection times in G-DI engines, steady state operation should be achieved ideally before full needle lift. For high SOI mass flow rates, their model, based on a relative mass flowrate term, showed the formation of large droplets in the jet regime. Cousin et al., (1998) concluded that for the reduced fuel injection pulse duration’s commonly required in G-DI duty cycles, the volume of fuel injected during the unsteady regime was not negligible and large droplets could potentially be detrimental to HC emissions. This was supported by the experimental studies of Pontopiddan et al., (1997) and Stanglmaier et al., (1998) who concluded that the volume of fuel supplied during the initial liquid jet stage could not be considered insignificant.

The break-up length can be considered of crucial importance to the successful operation of G-DI engines with compact combustion chambers, where injector to chamber or cylinder wall distance may be reduced through constraints imposed by the injector location and orientation or by geometrical modifications to combustion chambers and piston crowns. Wirth et al., (1996) claim that typical pressure-swirl injector break-up lengths are in the order of 25 mm from the nozzle exit. Penetration velocities of up to 50 ms⁻¹ were to be expected. Fraidl et al., (1996) stated that for efficient transport of the air-fuel mixture, the penetration velocity of the fuel spray should be at least of the same order of magnitude as the in-cylinder air velocity. Where penetration of the spray tip is too great, impingement of liquid fuel on hot combustion surfaces is likely to occur. This can have a significant effect on the spatial distribution of the liquid fuel and therefore on the preparation of the air-fuel mixture.

For injection early in the intake stroke there is minimal resistance to spray penetration and dissipation of the high-pressure energy is required. This is commonly achieved through the use of a ‘swirler’ tip or swirl chamber placed upstream of the nozzle. A ‘swirling’ nozzle can ensure that a proportion of the available pressure energy is imparted to the liquid fuel as a tangential momentum component before nozzle exit and will not contribute to the axial penetration. Fuel sprays from swirl type injector’s exhibit relatively small break-up lengths as a consequence of the relatively high internal ‘start of needle lift’ (SONL) Reynolds number. The subsequent flat, turbulent, fluctuating and rotating jet profile distorts rapidly.
A rotating or swirling spray will penetrate in both the axial and radial directions from the injector tip at a rate governed by the geometry and angle of 'swirler' tip and the nozzle cone angle. A distinction is made between expressions used to describe spray angles as the definitions are often loosely termed. For cases where continuous injection is not permitted, the transient nature of the spray, from an initial unsteady state to fully-developed, will vary the observed spray angles. Generally, the injector nozzle angle refers to the angle between the injector axis and the orifice inner seat or needle tip.

This is generally similar to the spray cone angle that is measured directly at the nozzle exit at SOI. Sprays with nominal SOI cone angles of less than 50° are generally classed as narrow and those in excess of 50°, as wide-angle sprays. Other spray cone angles, usually derived from experimental spray images, are based on an angular sweep of the spray that encompasses all perturbations found along the spray periphery when a quasi-steady state is achieved. For hollow-cone type fuel sprays this can produce misleading interpretations of droplet distributions. Whilst boundary defined spray angles may remain constant with changing operating conditions, no importance is attached to the possible changes in thickness of the cone walls (Hoffman et al. (1997).

For a given in-cylinder pressure condition, Meyer et al., (1997), Nogi et al., (1998) and Park et al., (1998) have shown that the spray structure is sensitive to the injector nozzle angle or fuel spray cone angle. Smaller spray angles generally lead to the development of more compact sprays whereas greater angles promote increased air entrainment. Should sufficient air entrainment exist, a 'hollow-cone' spray structure is observed. This can be utilised to promote faster evaporation and mixing. If the spray angle is too large during early injection then at full-load conditions, liquid fuel impingement can occur on the combustion chamber, piston crown, cylinder liner or intake valves depending on the injector location and inclination. Nogi et al., (1998) found that at an injection pressure of 9 MPa the spray angle could influence engine torque in a swirl type combustion chamber. If the spray cone angle is considered to be a function of in-cylinder pressure, then a spray angle that is too small can result in a compact, near-cylindrical spray pattern under part-load, late injection conditions, with excessive piston or chamber liquid fuel impingement.

Kume et al., (1996) using a LASER light sheet illumination technique, experimentally observed a 45° angle between the vertical and horizontal droplet velocity components on the spray periphery of a hollow-cone, swirl-type injector after SOI. They suggested that both the radial and axial penetration components of velocity in that region were approximately equal. Fuel injection into higher ambient pressures, typically greater than 20 bar for part-load operation, was seen to effectively reduce the cone angle with relatively more penetration observed along the spray axis than in the radial direction. Park et al., (1998) observed a reduction in spray cone angle with injection into 5 bar ambient pressure. A toroidal ring vortex was captured using LASER light sheet photography, but the spray pattern had not developed into a hollow-cone shape. The increased density of the charge and greater drag on the fuel droplets resulted in a more compact spray pattern than that observed at or below atmospheric pressure with reduced overall penetration distances. It should be noted that for non-swirl type injectors the opposite is often the case.
With increasing charge density, axial droplet retardation is the dominant force and the spray angle is seen to increase. These studies would support the analysis of Fraidl et al., (1996) who claimed that the fuel supply pressure had a minor effect on the spray cone angle for constant ambient pressure conditions. Increasing the ambient pressure from 5 to 15 bar effectively halved the spray cone angle.

Injector nozzle orifice, needle and seat angles must be carefully selected. Iwamoto et al., (1997) investigated different types of injector orifices for use with a pressure-swirl injector. A tangential slot was preferred to a conical or an axial-helical type orifice based on the application of a simple Swirl Reynolds Number (SRN) to the optimum minimal spray tip penetration distance:

\[ SRN = \frac{u r}{\nu} \]

where:
- \( u \) velocity in the swirling grooves [\( \text{ms}^{-1} \)]
- \( r \) radius of swirl [\( \text{m} \)]
- \( \nu \) kinematic viscosity of the fuel [\( \text{m}^2\text{s}^{-1} \)]

For an injection pressure of 5 MPa, spray tip penetration decreased with increasing SRN until an asymptotic minimum penetration value was reached. This was approximately 50 mm at an ambient pressure of 0.5 MPa (approximately in-cylinder, late injection conditions) for a SRN of 3 x 10⁴. Under these conditions, the spray cone angle showed little dependence on the SNR. No details of the type of needle or needle tip are given. Fraidl et al., (1996) showed that increasing the fuel supply pressure, for a given ambient condition, would lead to increased penetration velocities. For a given set of operating conditions, mean droplet diameters were asymptotic to a minimum value when plotted against spray tip velocity. Penetration velocities between 40 and 50 ms⁻¹ were observed for injection pressures in the range of 50 to 70 bar. The droplet velocities were approximately twice that of the maximum in-cylinder airflow and as such, convective transport or fuel spray deflection by the main flowfield would be minimal. They concluded that penetration could only be usefully increased to aid atomisation quality up to a certain limit. Preussner et al., (1998) applied the theoretical expressions of Brodkey (1967) to illustrate the relationship between droplet size and penetration observed by Fraidl et al., (1995). For a non-evapourating, spherical droplet moving in inviscid 'air' at rest, they proposed that droplet motion was governed by the following expression:

\[ \frac{d u_{rel}}{dt} = -\frac{3}{4} C_d \left( \frac{\rho_{air}}{\rho_{liquid}} \right) \left( \frac{1}{d} \right) u_{rel}^2 \]

where:
- \( u_{rel} \) magnitude of the droplet velocity relative to the gas [\( \text{ms}^{-1} \)]
- \( \rho \) density of phase [\( \text{kgm}^{-3} \)]
- \( d \) liquid droplet diameter [\( \text{m} \)]
- \( C_d \) drag coefficient of droplet in air [\( \text{c} \)]
The coefficient implies that neighbouring drops in the spray do not interfere with each other.

Upon substitution and rearrangement:

\[
\frac{du_{rel}}{dt} \propto \frac{1}{d^{0.5}}
\]

It should be noted that this analysis is concerned with individual droplets and no coalescence or momentum transfer has been considered in this approach. For comparable nozzle exit velocities, smaller sized droplets are rapidly decelerated and contribute little to overall penetration. For a given range of droplet sizes, axial retardation follows an exponential decay. The theoretical approach is limited over a broad droplet size range by the validity of the empirical expressions derived. The penetration rate and hence the likelihood of fuel impingement (with the consequence of poor emissions) in such fuel sprays could therefore be dictated by the number of over-sized droplets. In order to achieve the required spray penetration length whilst producing smaller droplets sizes, the injection pressure must be increased.

An approach to minimising fuel impingement and retaining sufficient spray momentum to reach the spark plug region will rely on the choice of a critical spray cone angle. For measured spray angles of between 55° and 90°, Nogi et al., (1998) found a critical angle of approximately 80° was suitable for their swirl concept. It is likely however, that tumble flow based systems employing similar operating pressures with late injection conditions will require smaller spray cone angles to achieve adequate penetration momentum for fuel transport to the spark plug region. These spray angles are likely to be as diverse in range as the types of G-DI concepts implemented. A 60° injector cone angle was considered optimum in the experimental study of Park et al., (1998). However, the experimental study of Meyer et al., (1997) showed that neither a 60° or 90° injection angle could successfully facilitate a stratified charge concept. In both cases fuel was seen to impinge on the cylinder walls. The choice of a nominal spray cone angle must be suited to the type of G-DI concept employed and be capable of adapting to injection regimes at different charge densities.

This approach to optimising spray penetration may often be too simplistic for realistic, fired engine application. Han et al., (1997) and Stanglmaier et al., (1998) realised that the effect of intake generated jet airflow, between the valve curtains, was detrimental to the spray penetration when injecting early into the cycle from a central injector at full-load conditions. The deflection effects were most prevalent where injection occurred near maximum mass flowrate conditions at inlet valve opening and were scaled with increasing engine speed (Stanglmaier et al., (1998)). Their simulation using the commercial, KIVA-3, computational fluid dynamics (CFD) code predicted that between 10-18% of total injected fuel mass could be deposited on the cylinder liner by this method.
For such an approach, the phasing of valve opening and injection timing are an additional element to be considered when assessing the fuel spray and gas interactions required for adequate mixture preparation.

1.4.2. Spray Atomisation Quality

The term 'spray quality' as applied to gasoline fuel sprays in this context refers to the spatial and temporal distribution, within a given spray structure or pattern, of the liquid fuel droplet quantities of:

- Droplet diameter distribution
- Instantaneous or ensemble-averaged droplet velocity
- Droplet velocity fluctuation
- Spray droplet concentration

Additionally, the spray quality must demonstrate good repeatability for efficient engine operation and reduced emissions. The majority of the data collected in this area of investigation has largely been obtained through the application of non-intrusive, optically based, experimental techniques. The validity of such data is dependent on the spray density (Ruff and Faeth, (1995)): a sensible and experienced application of the technique, a measured and rational interpretation of the results and knowledge of the limitations of the apparatus employed. The characterisation of a fuel spray by the analysis of dropsize distributions obtained by such methods can be misleading. The measured results are often subsets of the total flux quantities passing through the point of interrogation. In Chapter 3, the experimental techniques employed are described and the terms used to describe spray quality are discussed. As such, no reference to the validity of directly measurable and non-measurable spray terms is made in this section.

A requirement for the successful preparation of air-fuel mixtures through the direct injection of gasoline fuel into the combustion chamber is that the liquid is finely atomised. MPI engines rely on the secondary atomisation of liquid fuel droplets impacted upon and stripped from port walls, valve seats and valve stems to promote mixing and evaporation. Fuel injection by this means, especially against a closed valve, gives rise to a broader range of droplet sizes (Vannobel, 1992, 1993). This is a stage of the mixture preparation process that is not available in G-DI engines. A stratified charge, late injection strategy, where mixture preparation periods are much reduced, will require small droplet diameters for good air motion transport, mixing and evaporation. Suitable means of inducing fine atomisation of the fuel is required without the detrimental effect of low momentum droplets on the spray penetration. This can be achieved through injector selection, increased injection pressures and the augmentation of turbulent flow that acts on the droplet to overcome surface tension and viscous forces.

Atomised fuel mean droplet diameters vary approximately with the inverse square of the pressure differential between the in-cylinder air and fuel supply line pressures. However, increased operating pressures can lead to asymmetry in the fuel spray pattern and injection cycle-to-cycle irregularities as the operating loads on components is increased.
The atomisation of droplets is governed by the droplet Reynolds (Re) and Weber (We) numbers,

\[ We = \frac{\rho u_{rel}^2 d}{\sigma} \]

where:
\( \rho \) = gas density \( [\text{kg/m}^3] \)
\( u_{rel} \) = magnitude of the droplet velocity relative to the gas \( [\text{m/s}] \)
\( d \) = droplet diameter \( [\text{m}] \)
\( \sigma \) = surface tension \( [\text{kg/s}^2] \)

The droplet Weber number \( (We) \) represents a balance of forces and is defined as the ratio of the droplet inertial energy to the droplet surface energy. For \( We \) numbers greater than a critical value, atomisation of liquid droplets will occur. The resultant smaller, droplets have less inertia and are more readily slowed by the gas phase. The \( We \) numbers then become very small and secondary atomisation cannot proceed. It should be noted that although larger droplets possess more inertia, if they are injected into low ambient pressure conditions, the resultant droplet \( We \) number could be too small (less than critical) for atomisation to occur also. Critical \( We \) numbers and droplet break-up mechanisms are simplified in this approach. The processes involved are more complex and is currently being researched by a large number of workers.

To enable the characterisation and comparison of spray droplet diameter distributions, a series of generalised mean diameter terms are employed in the literature. These are detailed more specifically in Chapter 2. One such term that is commonly applied to fuel sprays in combusting environments is the Sauter Mean Diameter (SMD) or \( D_{32} \). The SMD is the ratio of droplet volume to surface area within a given population. It is often found useful for describing a droplet distribution in computational models where fuel evaporation and combustion characteristics are to be simulated. A broad range of SMD's have been recorded experimentally or observed through simulation, for hollow-cone gasoline fuel sprays.

In addition to pressure differential effects and the selection of injector geometry, a given spray structure exhibits SMD values that differ both spatially across the spray plume and with time from SOI. Droplet sizes are seen to be functions of injection pulse width and frequency (duty cycle). Kelly-Zion et al., (1996) observed decreasing droplet size distribution for increasing pulse width at constant frequency. The same tests applied at a constant pulse width but increasing frequency, recorded decreasing droplet diameters.

Average values of SMD are commonly erroneously reported across complete planes of spray plumes ('line-of-sight' SMD). Zhao et al., (1997) suggest that the SMD may not be sufficient to characterise the droplet diameter distributions in hollow-cone, gasoline fuel sprays. In transient, gasoline fuel sprays of this nature, smaller droplets in a broad diameter population will contribute less to the total fuel mass than those much larger. The larger droplets will remain as liquid for longer periods and ultimately lead to an increase in unburnt HC emissions.
Where SMD's are to be applied in transient fuel sprays, it is more representative to ‘weight’ individual ‘averaged’ SMD values by the number flux of droplets collected at that point (Ren et al., 1998).

Tomoda et al., (1997) compared the Diesel fuel spray swirl penetration expressions of Wakuri et al., (1960) with hollow-cone, gasoline fuel spray measurements. Experimental results using the LASER light diffraction technique confirmed that whilst the spray droplet liquid volume from SOI was always greater than that of a plain orifice injector, penetration was reduced by up to 40% for a given injection pressure and volume flow rate. This is indicative of a much smaller range of SMD’s achieved through swirl atomisation. Typically, pressure-swirl injectors produce a narrower distribution of droplet sizes than plain exit orifices. Increasing injection pressure further reduced the SMD to approximately 15 µm at 12 MPa and 10 µm at 20 MPa when injecting into atmospheric pressure conditions. It should be noted however that although the droplet diameter range remained approximately constant, there were instances when large sized droplets at increased velocities were observed. Pontoppidan et al. (1998) experimentally observed SMD’s, using Exol/CSL2 test fluid, in the range of 20-28 µm, 15 mm downstream of the nozzle exit, on the mid-plane of a spray under identical injection and ambient pressures. For a pintle-type, hollow-cone, pressure-swirl injector, droplet velocities were measured in the range of 17-112ms⁻¹, with a mean value of approximately 60 ms⁻¹ over the injection duration of 2 ms. For the three nozzles tested by Pontoppidan et al., (1998), an apparent difference of approximately 30% in the SMD values was observed across the complete spray plume at a distance of 15 mm from the nozzle exit. Park et al., (1998) observed that with increasing distance from the nozzle of a pressure-swirl injector, measurements of SMD and velocity tended towards a uniform distribution. SMD values were recorded within the range of 15-30 µm towards 60 mm from the nozzle with a 60° injector cone angle. With injection pressures increased from 20 to 70 bar, the SMD value decreased across the spray from 20 to 17 µm.

SMD values obtained experimentally using the LASER diffraction method by Tomoda et al., (1997) with a PZT pressure-swirl injector were seen to rapidly reduce with increasing injection pressure. At 5 MPa injection pressure SMD’s were recorded at 20 µm, at 15 MPa, 15 µm and at 20 MPa, 10 µm. However, the range in fuel droplet diameters remained approximately constant between 0-50 µm. Iwamoto et al., (1997) measured SMD’s of less than 15 µm in the spray periphery but those greater than 30 µm along the spray centre axis. Han et al., (1997) computed average SMD’s of approximately 30 µm across an approximately 55° spray cone angle gasoline spray, injected at 5 MPa line pressure into atmosphere.

At an increased pressure of 7 MPa and with a spray cone angle of 80°, Nogi et al., (1998) measured an ‘average’ SMD of approximately 15 µm in 0.1 MPa ambient pressure, at 50 mm from the nozzle. Kelly-Zion et al., (1996) support inferences from the above experimental investigations that the size distribution changes most significantly at measurement points in the spray periphery. The droplet size distribution decreased however with increasing temperature as the rapid evaporation and increased entrainment of small droplets took place. Dodge et al., (1996) recommended mean droplet diameters of approximately 15 µm for adequate evaporation. Ren et al., (1998) assumed that small sized droplets in the spray centre region would evaporate more effectively and that the recirculatory gas flow would carry vapour to the centre of the spray.
The experimental study of steady state and pulsed, chilled, gasoline sprays by Kelly-Zion et al., (1996) indicated increased liquid core lengths and reduced atomisation quality when the gas temperature was lowered. Vannobel et al., (1993) found a poor correlation between measured droplet sizes between hot and cold flows in a MPI engine. Typically, 'cold' droplet diameters in MPI engines have been observed up to 200 μm for injection pressures of between 270-450 kPa (Carabateas et al., (1996)). Under realistic MPI engine operation conditions, Vannobel et al., (1996) experimentally observed 'direct spray' droplets (open-valve injection) with an SMD of 90 μm. These were consistently smaller than those measured in a closed-valve injection (CVI) case which ranged up to maximum values of 120 μm SMD.

The spray quality of a hollow-cone pattern differs spatially and temporally within the envelope of the spray plume. Both the experimental results of Yamauchi et al., (1996) and Ren et al., (1998) indicated that small droplets, more influenced by the surrounding airflow, migrated to the central region of the fully-developed spray. PDA measurements of droplet diameters and droplet number densities in a 70° spray cone angle, swirl injector were performed by Yamauchi et al., (1996). These indicated that from 0.4 ms after EOI, the smaller and less densely populated diameter droplet region (0-10 μm) was surrounded by a broader small and medium sized droplet region that extended outwards to the spray periphery. Along the periphery and at the spray centre axis immediately following injection, larger droplets were present in the range of between 25-58 μm.

LASER light sheet photography was utilised by Park et al., (1998) in a spray visualisation study of in-cylinder fuel spray impingement under motored engine conditions. Different piston crown geometries and airflow tumbling motions were investigated for both central and side injector positions. Spray visualisation revealed piston crown impingement and deflection under cold conditions. However, no quantitative results for spray quality post-impingement were recorded in this investigation. Much of the research undertaken in fuel spray impingement has been limited to water or Diesel fuel sprays injected under atmospheric pressure conditions onto flat cold or hot plates. Quantitative gasoline fuel spray data after impingement on complex piston crown geometry's are required for evaluation of the transport of liquid and vapour fuel phases to the spark plug through the action of the bulk in-cylinder air motions. Post-impingement Phase Doppler Anemometry (PDA) analyses of MPI gasoline fuel sprays have been carried out by Dementhon and Vannobel (1991) and Vannobel et al., (1992). They used an imaginary, idealised Lagrangian approach to maximise the PDA instrumentation data validation rate of droplet detection. This was performed in a steady state flow rig at varying volumetric flow rates. Importantly, they concluded that the largest droplet diameters, post-impingement, were obtained where air and liquid velocities were approximately equal.

Experimental results by numerous research groups including Salters et al., (1996), Wirth et al., (1996), Kelly-Zion et al., (1996), Parish and Farrell (1997), Comer et al., (1998), Nogi et al., (1998), Stanglmaier et al., (1998) and Ren et al., (1998) have shown the presence of an initial 'slug' or narrow core jet of liquid fuel that precedes the main injection (or cone edge) along the central axis of the spray. The origin of these large droplets is in this liquid core. They are formed during the period of initial pintle retraction when the nozzle internal flow conditions are unsteady. Experimental droplet sizing techniques such as PDA illustrate very low discrete droplet detection rates within this region (Pitcher et al., (1990) and Shrimpton et al., (1992)).
This phenomenon effects both the spray atomisation quality and spray tip penetration. These can be considered separately from the effects of injection needle bounce, which leads to undesirable secondary injections and density irregularities in the spray pattern. Secondary injections are often characterised by larger droplets of lower velocities. It is likely that this represents the displacement of a dead volume of fuel held in the injector sac volume that is not at fuel line pressure. The displaced sac volume can effectively reduce the amount of atomisation through retardation of the main fuel volume arrival resulting in large droplets traveling at increased velocities. It is also possible that detachment of liquid parcels issued from the tip of the preceding liquid jet column at break-up could contribute to the observed measurements of initial leading fuel mass 'slugs' along the central axis (Cousin et al., 1998). Yamauchi et al., (1996) referred to the effect as the 'coarse droplet phenomenon' when they observed experimentally many large droplets near the spray axis and on the spray tip, with a pressure-swirl injector. Wirth et al., (1996), Preussner et al., (1998) and Comer et al., (1998) assumed that the injector was acting as a simple plain orifice injector during the early jet stage, with the ejected liquid having little or no swirl component of velocity. Pressure energy was equated to the initial acceleration of the flow in this phase. Evidence to support this is found in the experimental work of Salters et al., (1996). They utilised a Zexel type, high-pressure, solid-cone injector (spray cone angle of approximately 50°) with a plain-type exit orifice. Under both fired and static engine conditions, they identified an initial fuel 'slug' that penetrated before the arrival of the main spray cone. Large droplets, possibly up to 100 μm in diameter and traveling at velocities of up to 40 ms⁻¹, were recorded in the period from SOI to approximately 1.5 ms after injection. At the same in-cylinder location, the main spray cone arrived with velocities of approximately 13 ms⁻¹. Piston crown impingement was observed using LASER light sheet illumination and a 2 camera CCD-based imaging system with SOI set at 80° ATDC.

In all cases, center axis AMD or SMD values are seen to rise sharply due to the arrival of the leading fuel slug in fixed position, point measurement, experimental techniques. The following spray tip, formed during start-of-needle-lift (SONL), causes a rapid decrease in the SMD values. Additionally, this would appear to be related to the duration of the injection pulse width. Hoffman et al., (1997) recorded small amounts of mass flux with a spray patternator, along the spray centreline, from 0-0.4 ms after SOI for a 7 MPa pressure-swirl atomiser. For short injection periods, mass flux through the orifice is at a maximum immediately after SONL and more fuel is concentrated along the central axis (Zhao et al., 1997).

Stanglmaier et al., (1998) observed pre-slugs of fuel at up to 1.4 ms from SOI, using LASER light sheet droplet imaging under static conditions and in a motored engine at 750 and 1500 rpm. A Zexel, high-pressure, hollow-cone pressure-swirl injector, similar to that of Kume et al., (1996) and Iwamoto et al., (1997), was used for the experimental investigation. The main body of the spray did not reach the plane of investigation until 3.4-3.6 ms after SOI at 750 rpm. The pre-slug made contact with the piston at approximately 1.9 ms after SOI, where injection pulse width was fixed at 2 ms. The effect was reduced at increased engine speed. With a static engine, the greatest density of 'measurable' fuel drops was concentrated around the cylinder axis.
1.5. Conclusions of Literature Survey

The reduction in exhaust emissions from automotive vehicles imposed by future worldwide legislation requires an increased knowledge of the key factors that can best be utilised to meet these stringent limits. Fundamental to this understanding are the margins that can be gained through advancements in the efficient preparation of a suitable air and fuel mixture within the combustion chamber that is dependent upon the engine load and speed requirements. Gasoline direct injection technology and fuelling strategies offer a potential solution.

Many approaches to the preparation of an air and fuel mixture within the combustion chamber of a direct injection engine have been proposed in the literature. Homogeneous charge operation is achievable over a relatively large range of SOI timings, whereas stratified charge operation is dependent upon the correct timing of the fuel delivery. Differences in the in-cylinder air tumble motion have been reflected in ignition instabilities. As such, the current research direction has involved matching the in-cylinder airflow and fuel delivery systems to ensure combustion stability and to broaden the operating range under stratified charge conditions.

The primary benefit of adding a rotational component to the in-cylinder airflow within a spark ignition engine is to promote the combustion rate. Large-scale bulk motion structures improve mixing and transport, whereas smaller scale motions enhance diffusion and mixing between the unburnt and burnt gases. Turbulence can be augmented at TDC through the conservation of the energy contained within the bulk flow induced during the intake stroke. This is affected by the combustion chamber geometry. In a modern, MPI combustion chamber, the increased pressure and pent-roof chamber geometry increase the shear stress and the vortex structures are destroyed during late compression. A direct injection strategy, however, must rely upon the conservation of these bulk flow structures during late compression to provide stratification of the in-cylinder charge. Turbulence enhancement is provided by the squish flows entering the combustion chamber at TDC. The requirements of a G-DI in-cylinder airflow are therefore partially opposing; that is it must support both mixture preparation and maintain stable charge stratification. This highlights the importance of both the spatial and temporal repeatability of the coherent, mean air motions. In addition, the suppressive or deflective effect of momentum transfer brought about by the direct injection of liquid fuel into the chamber must be considered.

For a direct injection strategy, the fuel spray must be optimised to the combustion system with regards to atomisation quality, geometry (cone angle) and penetration length. This is especially important where mixture preparation times are greatly reduced. The majority of fundamental liquid atomisation research relates to Diesel injectors. The correlations developed in these studies are not readily applicable to the determination of a G-DI fuel spray suitable for such mixture preparation criteria. Automotive fuel sprays are unsteady and therefore exhibit a temporal dependency that must be considered during the evaluation of the spray characteristics over short injection durations and in locations that are most effected by the intake jet flows. Current research suggests that a hollow-cone, pressure-swirl type of injector can best be adapted to meet the requirements for both homogeneous and stratified operation. There are, however, reported cases with these types of injectors, where a ‘slug’ of fuel has been observed that preceded the main injection event and proved detrimental to the uBHC emissions.
1.6. Outline of Thesis

Chapter 1: The first chapter is an introduction to the concept of direct injection gasoline engines and the numerous approaches to fuel-air mixture preparation for both homogeneous and stratified charge engine operation as identified in the literature. In particular, close attention is drawn to the application of in-cylinder air motions and fuel injection technologies that address the problems inherent to a dual-mode engine operating strategy. The nature of the problem is investigated, along with a discussion of the benefits of such systems and some of the solutions found in the literature are outlined.

Chapter 2: This chapter aims to introduce the types of measures and scales that can be used to experimentally quantify those engine characteristics pertinent to a direct injection strategy. The experimental methods and analysis of the results are selected with respect to their specific application to in-cylinder IC engine gas flows and fuel sprays. The intrinsic link between the experimental techniques and the careful interpretation of the results is discussed in detail.

Chapter 3: The third chapter details the experimental techniques utilised in the series of studies. The theories and types of data analysis are drawn from the literature and from experimental observation and are presented with reference to IC engine studies. The interpretation of the results and assessment of experimental uncertainties is included. A comprehensive knowledge of these data acquisition techniques and the subsequent correct interpretation of the results are of fundamental importance in any experimental study and as such, a complete chapter is devoted to this area.

Chapter 4: The fourth chapter details a preliminary experimental study of the airflow characteristics in the cylinder and combustion chamber of a modern, pent-roof, indirect injection, gasoline engine. This study was instrumental in establishing the operating method, system sensitivity and practical errors for the LDA technique within a motored engine. Importantly, the study established an experimental baseline with which to compare the characteristics of the G-DI engine.

Chapter 5: The fifth chapter describes a comprehensive study of the air flowfield characteristics in the combustion chamber of a four stroke, G-DI engine with pent-roof cylinder head geometry and a ‘bowl-in-piston’ crown. The bulk, mean airflow and smaller-scale turbulent structures are presented throughout the four-strokes using LDA. Particular attention is drawn to the features of the in-cylinder airflow during the periods of early and late fuel injection. The scale of the turbulent structures at typical ignition timings and close to TDC are presented. The turbulent nature of the G-DI airflow at TDC is compared with the MPI baseline study investigated in Chapter 5.

Chapter 6: The sixth chapter involves the characterisation of the gasoline fuel spray in an ambient, quiescent chamber and under static and dynamic engine conditions for differing speed and load strategies. Fuel liquid droplet size and velocity measurements are performed using PDA. Visualisation of the fuel sprays is performed using high-speed photography, shadowgraphy and LASER light sheet illumination. The effect of the in-cylinder air flowfield upon the fuel spray, prior to fuel injection, as measured in Chapter 5, is discussed and compared with the static, control tests.

Chapter 7: The final chapter summarises the conclusions drawn from the body of work and proposes several recommendations for future work.
KEY

**HC**  Hydrocarbons
**NMOG**  Non-methane organic gases
**NOx**  Oxides of Nitrogen

**JAPAN 10-15 MODE**  Japanese Combined Emission Test Cycle
**ECE 15 + EUDC**  Stationary and Transient European Emission Test Cycle
**FTP 75**  United States Federal Test Procedure for Car and Light Truck Emissions
**EURO I-V**  European Union Tailpipe Emissions Standards
**LEV/LEVII**  California Low Emission Vehicle Standard (LEV until 2003)  (LEVII extends from 2004 to 2010)
**ULEV/ULEVII**  California Ultra Low Emission Vehicle Standard
**SULEV**  California Super Ultra Low Emission Vehicle Standard

*Figure 1.0. Comparison of Worldwide HC-NOx Emissions Legislation (Jackson, 1997a)*
Figure 1.1. Comparison of Log Pressure-Volume Indicator Diagrams between an MPI and a G-DI Engine (Stokes, 1998).

Figure 1.2. Strategies for Gasoline Direct Injection Engine Operation (Stokes, 1998).
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Wall-guided Air-guided

- Exact positioning of spray and spark plug required
- High stress on spark plug

Actual development to wall-/air-guided systems

- Fuel transport to spark plug due to internal flow
- Specification of spark plug remains standard

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Table 1.0. Experimental and Computational Studies of Gasoline Injectors for a Direct Injection Fuel Injection Strategy
2.0. Fundamental Principles of In-Cylinder Air Flows and Fuel Sprays of Internal Combustion Engines

2.1. Introduction

In-cylinder turbulence is defined as non-stationary across the engine cycle (statistically unsteady), stochastic and invariably anisotropic. Its primary role in internal combustion (IC) engines is that of promoting momentum, heat and mass transfer and chemical reaction through mixing. The strong correlation between turbulent in-cylinder air motion and combustion performance in IC engines has long been known. For homogeneous, stoichiometric spark ignition, high turbulence intensity but low mean velocity is required at the spark gap such that burning is promoted and the flame kernel is not detached from the spark plug gap. The in-cylinder fluid dynamics supports two important mechanisms: small or micro scale turbulent fluctuations in the fluid are essential in promoting efficient mixing of the pre-burn air and fuel vapour. Following ignition, they augment the burning velocity through diffusive mixing in the early flame kernel and induced wrinkling of the propagating flame front. Turbulent gas fluctuations approximately scale with increasing engine speeds. This leads to increased turbulent burning velocities that can be utilised for minimum advance of ignition timing for best torque (MBT). Large scale or bulk mean flow is utilised in the transport of the mixture plume to the combustion zone and the convective transport of the flame front from the spark plug gap. In G-DI engine configurations its role is additionally to support stable charge stratification. This can be achieved by increasing the preservation of the mean flow. Both types of flow possess different physical characteristics that vary in magnitude from cycle to cycle, within a given stroke and with in-cylinder location.

Some of the physical manifestations of in-cylinder turbulence generation, dissipation and decay have been reported in Chapter 1. This section will address the features of in-cylinder turbulent flows and introduce some of the important measures and expressions used to define them. In addition, it will discuss the important role of pre-combustion turbulence on flame development and burn rates in automotive spark ignition combustion systems. Finally, the significance of MBT and LBT ignition timings and injection swings on torque response and emissions will be introduced for a G-DI combustion system.

The first section of this chapter introduces the important fluid dynamic expressions commonly utilised as the starting point for more advanced studies of in-cylinder flow characteristics. These incorporate terms that describe the nature of turbulent fluid flows and expressions that can be used to quantify the spatial and temporal distributions of these features. An effort has been made to limit these discussions to those areas most pertinent to the analysis of in-cylinder gas and liquid flows. The more advanced analyses in future chapters will be based upon these foundations.

Furthermore, this chapter describes the features of gas flows and fuel sprays in IC engine combustion chambers that merit the special requirements of the LDA or PDA techniques. Central to this, is the intrinsic link between the description (or descriptors) of turbulent flows and the tools that yield the measurements.
No study of turbulent flows using the experimental techniques of LDA or PDA is complete without a careful description of what can be measured as mean and flow turbulent properties. From the advent of these developing techniques, long employed in turbulent flow conditions, there emerged the necessity for new analysis methods to describe the unsteady and non-stationary nature of in-cylinder flows. As yet, there exists no unambiguous method of decomposing measured instantaneous velocity vectors into their turbulent and non-turbulent parts. Some of the proposed decomposition techniques are set out in this chapter. Table 2.0a and b. summarises 18 experimental in-cylinder LDA flow investigations found in the literature from 1988.

The general features of flows in IC engines are as follows:

- Multi-dimensional, non-stationary turbulent flow due to the reciprocating motion of the piston at all engine speeds and for all combustion chamber geometries.
- The spatial flow boundaries are continuously time-varying.
- The mean flow is periodically in phase with the engine cycle with high mean velocity fluctuations over the engine cycle.
- Cycle to cycle variations in the mean flow exist throughout the engine cycle and vary with temporal location in the cycle and spatial location within the chamber.
- The turbulence intensity at top-dead centre (TDC) firing can reach the same order of magnitude as the mean flow velocity.
2.2. A General Description of Turbulence in Spark Ignition IC Engines

Turbulent flows may be classified as either Boundary layer or Recirculating flows. In the former case, the flow is uni-directional and intense shear is experienced over thin regions near boundary walls or interfaces with other fluids. In the latter case, there is no predominant flow direction and regions of flow separation from boundaries can be found. In practice, most flows exhibit features from both classes.

The equations of motion that describe an incompressible, Newtonian, viscous fluid are the Navier-Stokes conservation equations. These can be derived from a control volume analysis of an infinitesimal element of fluid governed by the continuity equation. For the control volume illustrated in Figure 2.0, of constant mass and fluid density, far from a free surface, fluid must leave the volume at the same rate that it enters. For an infinitesimal volume, \( dxdydz \), with orthogonal, perpendicular instantaneous velocity components, \( U, V \) and \( W \), in the \( x, y \) and \( z \) directions respectively, mass conservation is observed when

\[
\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z} = 0 \tag{2.0}
\]

If the same analysis is applied to the rate of change of velocity of the fluid as it passes through the control volume in time, \( t = dx/U \), an expression can be derived for the component in the \( x \)-direction of the acceleration following the motion of the fluid (similarly for the other two components)

\[
\frac{\partial U}{\partial t} + U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} + W \frac{\partial U}{\partial z} \tag{2.1}
\]

From Newton's second law of motion, the rate of change of momentum (the product of mass and acceleration) is equal to the sum of the forces acting on the fluid. Ignoring body forces, there are two others. Pressure gradients, generated by the motion of the fluid exert a net force on the fluid element between any two parallel faces of the control volume. Stress gradients, caused by deformation of the fluid, produce a difference in forces between the opposing faces (constant stresses do not produce a net rotation or acceleration of the fluid). The stress is proportional to the rate of strain (\( \partial U/\partial x \)) in Newtonian, viscous fluids. The pressure gradient acting on a fluid element of pressure, \( P \) density, \( \rho \) and kinematic viscosity, \( \nu \), between the faces perpendicular to the \( x \)-direction, will give rise to a net force given by

\[
\left( -\frac{\partial P}{\partial x} \right) dx dy dz \tag{2.2}
\]

The net force in the \( x \)-direction produced by normal and shear forces acting on the surface is given by

\[
\frac{\partial}{\partial x} \left( \mu \frac{\partial U}{\partial x} \right) dx dy dz + \frac{\partial}{\partial y} \left( \mu \frac{\partial U}{\partial y} \right) dx dy dz + \frac{\partial}{\partial z} \left( \mu \frac{\partial U}{\partial z} \right) dx dy dz \tag{2.3}
\]

Per unit mass, the Navier-Stokes equation for the acceleration of a fluid element in the \( x \)-direction is equal to the sum of the pressure and stress gradient terms.

Substituting for kinematic viscosity, the expression becomes
Equations for $y$ and $z$ momentum can be obtained through cyclic substitution. There are four coupled partial differential equations (including the pressure equation) and analytical solution is very rarely possible and experimental measurement methods are required.

In effect, the above equations relate the net flux of some quantity out of the infinitesimal fluid element surfaces to the net sources of the same quantity within the control volume. The quantity may be transported in or out of the volume by the mean flow, by turbulent diffusion (including pressure fluctuations) or by diffusion at a molecular level. These turbulent diffusivities are not determined by the mean gradient of the transported quantity, are not usually constant, but depend on the preceding history of the flow which carries the turbulent eddies. There exist both productive and destructive processes within the volume. A quantity may be produced within the volume through the action of body forces or pressure gradients or by exchange with another quantity (e.g. production of turbulent energy at the expense of mean flow kinetic energy). Alternatively, a quantity may be destroyed either by further exchange processes (e.g. dissipation of turbulent kinetic energy into thermal internal energy) or absolute removal (e.g. conductivity leading to the destruction of temperature fluctuations).

Turbulence produces additional fluxes of quantities through inducing turbulent stresses that are generally significantly higher than those encountered through viscous effects. In turbulent flow, the velocity fluctuates about the mean value, $U$, with a time-dependent component, $u$. The time-averaged value of $u$, denoted by $\bar{u}$, being zero by definition. Now the momentum flux in the $x$-direction through face $dydz$ is equal in mean to

$$\rho U^2 dydz \quad \text{or} \quad \rho U^2 dydz + \rho u^2 dydz$$

as $\bar{u}$ is zero and $\bar{u}^2$ is the mean value of $u^2$. Therefore the velocity fluctuation, $u$ produces a mean momentum flux of its own, proportional to the mean square of the fluctuating velocity. From Newton's second law, this results in an extra apparent stress, $-\rho u^2$ normal to the face $dydz$.

In a similar way to the above analysis, extra normal stresses are found in the $y$ and $z$ directions and extra shear stresses (e.g. $-\rho uv$ on the face $dxdz$ for $x$-momentum passing the face $dxdz$) on all faces of the control volume. These six extra turbulent stresses are referred to as the Reynold's stresses and control the rate of generation of turbulence.

Random motion eddying and oscillations of a range of wavelengths and frequencies occur in turbulent flows. The largest eddies are only limited by the system boundaries whereas the smallest extend into the region of molecular diffusion. Homogeneous turbulence is defined as a turbulent flow whose velocity fluctuations in the system are random and yet the time-averaged characteristics are independent of position in the fluid (no spatial gradients).

Where the velocity fluctuations are additionally independent of the direction of reference, the turbulent flow field is described as isotropic (no preferred direction).

This randomness implies that turbulence quantities can be interpreted by statistical distributions of a stochastic function; the 'central limit theorem' states that the probability
distribution of a continuous variable that is the sum of a large number of independent variables is approximately Gaussian. Generally, however there is often a small departure from the Normal distribution in most turbulent flows marked by the skewness and kurtosis of the statistical distributions.

The variables in the Navier-Stokes equations refer to instantaneous values at a point under consideration. In turbulent flows, their direct application yields little useful information where individual values vary to a significant degree. Reynolds modified the Navier-Stokes equations so that the variables could be treated as time-averages. For steady turbulent flows, the flow can be treated as a time-averaged mean flow on which is superimposed a turbulent time-varying component. This component is defined as the fluctuating velocity component, \( u' \) and is given by the RMS value of the fluctuation about the mean over a given period of time. The turbulence intensity is defined as \( u'_T = \frac{u'}{U} \).

2.2.1. Instantaneous Velocity Decomposition in Non-Stationary Turbulent IC Engine In-Cylinder Flows

In-cylinder turbulence is non-stationary throughout the engine cycle (statistically unsteady), stochastic and invariably anisotropic. The mean values of fluctuating quantities are not independent of time or spatial location. There is a non-stationary variation of the mean flow across the engine cycle and additionally a variation from cycle to cycle in the bulk fluid flow caused by instabilities in the intake and exhaust processes. Recently, Naradajah et al., (1992) and St Hill et al., (2000) highlighted significant jet flapping instabilities in a modern pent-roof gasoline engine with dual intake ports. Under such conditions it is often difficult to obtain reliable estimates of parameters to define turbulence components.

Typical reciprocating engine in-cylinder turbulent shear flows are periodically non-stationary or unsteady and the flow pattern changes continuously throughout the strokes of the cycle. There exists both boundary layer and recirculating type viscous flows that exhibit a broad range of large ('macro-scale') and small ('micro-scale') turbulent motions. Macro-scale eddies decay into micro-scale turbulence where energy is then rapidly dissipated. These turbulent flows can be characterised by their temporal resolution (turbulence intensity) and their spatial resolution (integral length scales). These definitions provide a quantitative description of a distance characteristic to the flow pattern or a measure of the speed of turbulence over a characteristic distance. Typical experimental measurements of in-cylinder flows yield instantaneous velocity vectors at a given location in the chamber at a known crank angle. Decomposition of these measurements is required to ascertain the turbulent and non-turbulent contributions. Within the experimental techniques available (for data acquisition) there is no single, readily applicable (or wholly justifiable) method for such a decomposition. Regions of in-cylinder flows that exhibit a marked departure from the classical Reynolds turbulence model often further complicate such analyses.

The following sections are drawn from the experimental in-cylinder studies available in the literature since 1988 as summarised in Tables 2.0a and b. These focus on the characterisation of
in-cylinder turbulent gas, vapour and liquid flows that can be described by and within the limitations of the experimental methods utilised. Reference is made to the inherent relationship formed between these methods and a general description or useful measure of turbulence.

2.2.2 Ensemble-Averaging

One decomposition technique most often applied to motoring engine studies involves the ensemble or phase averaging of instantaneous values recorded over many cycles. In stationary, turbulent flows, it is valid to use the classic Reynold's model for turbulent velocity and divide the instantaneous velocity, \( U \) into a time-averaged (ensemble) mean velocity, \( \bar{U} \) and a statistically random fluctuating velocity, \( u \) about that mean velocity.

\[
U = \bar{U} + u
\]

The root-mean-square (RMS) value of the fluctuation velocity is termed the fluctuation intensity, \( u' \).

\[
u' = \sqrt{(U - \bar{U})^2}
\]

This technique assumes that the mean flow does not differ appreciably at a given location and for a given crank angle from one cycle to the next. If the mean velocity is assumed to remain constant, then any cycle-to-cycle variation in the mean flow will be assumed to be a contribution to the turbulence. This is not wholly adequate for the definition of in-cylinder flows where appreciable differences between the mean flow of each individual cycle are observed (such as in fired studies). The contribution of cycle-to-cycle variation to turbulence levels will also be significantly higher in regions of the flowfield less affected by directed flows (e.g. Liou et Satavicca (1983), Glover et al., (1998a,b), Fansler and French (1992), Mahmood et al., (1996), St. Hill et al., (2000)). As such, the cycle-to-cycle variation in the mean velocity at a fixed point can be defined as the difference between the time-averaged mean velocity (arbitrary time interval) in a given single cycle and the ensemble-averaged mean recorded over many consecutive cycles. The instantaneous velocity is a function of crank angle and of the particular engine cycle.

2.2.3 Cycle-Resolved Analyses

Heywood (1988) used Figure 2.1a. and 2.1b. to graphically illustrate the velocity components that can contribute to an ensemble mean average measurement. Figure 2.1a. shows a typical ensemble average trace where cycle-to-cycle variation in the bulk flow is low. In this case, the ensemble-averaged mean velocity and the fluctuation velocity about that mean can be used to produce a reasonable estimate of the instantaneous velocity. In figure 2.1b., the cyclic variations are appreciable and the ensemble average is a poor estimation; both over and under predicting the instantaneous velocity profile. Instead, the individual cycle mean velocity is a better approximation. In both Figures 2.1a. and b., the dark circle denotes an individual measurement of instantaneous velocity.

Then, for a given crank angle, \( \theta \), the instantaneous velocity in the \( i \)th cycle is given by the sum of the time-averaged mean in that cycle, referred to as the in-cycle bulk velocity (Liou and
Savaticca, (1983), Corcione and Valentino (1991)) and the turbulent fluctuation about that mean, such that

$$ U(\theta, i) = \overline{U}(\theta, i) + u(\theta, i) $$  \hspace{1cm} 2.8

and the ensemble averaged mean of these instantaneous velocity's taken over $n$ total cycles at the same point is given by

$$ \overline{U}_{EA}(\theta) = \frac{1}{n} \sum_{i=1}^{n} U(\theta, i) $$  \hspace{1cm} 2.9

such that for $n$ cycles,

$$ U(\theta, i) = \overline{U}_{EA}(\theta) + u(\theta, i) $$  \hspace{1cm} 2.10

It should be noted that the in-cycle bulk velocity is a function of crank angle and cycle number whereas most practical measurements yield the ensemble mean over many cycles for a given crank angle. Data is normally collected over $n$ cycles in order to obtain a statistically significant data set. The number of cycles, $n$ varies with the spatial location within the combustion chamber. The practical constraints of time and data storage usually limit the maximum number of cycles used. Hong and Chen (1997) showed that for a sufficiently high data rate (approximately 40kHz) and sample set (400,000), the ensemble averaged velocity and the ensemble averaged in-cycle bulk velocity were of the same order in the tumble plane of their two-valve conventional combustion chamber. However, the integral length scales estimated from the ensemble averaged data at TDC were markedly greater than those from the cycle resolved analysis. It was suggested that these were unaffected by low frequency cyclic variations. This value would appear to be the minimum data rate favoured amongst other research groups for cycle-resolved measurements (e.g. Glover et al., (1988a,b), Arcoumanis et al., (1990), Hadded and Denbratt (1991), Moriyoshi and Muroki (1995), Baby and Floch (1997), St. Hill et al., (2000)). In most cases, the choice is often arbitrary with little justification offered. The type of measurement system employed seems to make little difference to the situation. It is unlikely that this assumption is valid throughout the complete cycle particularly since sample sizes have been seen to be strongly dependent upon spatial location.

From the above section and Figure 2.1., the cycle-to-cycle variation in the mean velocity, $\hat{U}(\theta, i)$, in the $i^{th}$ individual cycle is defined

$$ \hat{U}(\theta, i) = \overline{U}(\theta, i) - \overline{U}_{EA}(\theta) $$  \hspace{1cm} 2.11

and therefore for completeness

$$ U(\theta, i) = \overline{U}_{EA}(\theta) + u(\theta, i) + \hat{U}(\theta, i) $$  \hspace{1cm} 2.12

Importantly, this defines the turbulence intensity in relation to the individual cycle mean velocity.
The RMS fluctuation intensity in the ensemble average mean velocity, \( u'_{EA} \) for a given crank angle over \( n \) cycles is given by

\[
u'_{EA}(\theta) = \left( \frac{1}{n} \sum_{i=1}^{n} [u(\theta, i)]^2 \right)^{\frac{1}{2}} = \left( \frac{1}{n} \sum_{i=1}^{n} \left( U(\theta, i) - \overline{U}_{EA}(\theta) \right)^2 \right)^{\frac{1}{2}} \tag{2.13}
\]

In the review of the literature, Heywood (1988) remarks that during compression, the cycle-to-cycle mean flow variation can be of a comparable magnitude to the ensemble-averaged turbulence intensity and is seen to scale with piston speed. Where cycle-to-cycle variations are of this order, the ensemble-averaged fluctuation intensity will not represent the ‘true’ RMS turbulence intensity. From their experimental study, Hong and Chen (1997) have shown that in the tumble plane of a conventional SI chamber, the total ensemble-averaged fluctuation intensity, \( u'_{EA} \) can be considered as composed of the ensemble-averaged turbulence intensity and the fluctuation in the in-cycle bulk velocity given by the relationship

\[
u'_{EA} = \sqrt{u'^2 + \overline{U}^2_{RMS}} \tag{2.14}
\]

2.2.4. Crank Angle Intervals

As discussed earlier, high data acquisition rates are often needed in order to obtain a statistically significant value for the mean velocity at a particular location in an individual cycle. With LDA and PDA measurements it is often difficult to achieve such rates. A common solution to this problem is to define a range of crank angles, or window, over which the data can be acquired. This leads to a problem in defining the size of the window and care has to be taken to avoid large velocity gradients across the window, which leads to an effect known as crank angle broadening. The width of window will then determine the maximum frequency contribution to be included in the estimate of the mean velocity. Conversely, if the selected window is too narrow, impractically large data sets will be required for a statistically acceptable mean. For a crank angle window of width \( \Delta \theta \), equally about crank angle, \( \bar{\theta} \), Equation 2.29 becomes

\[
\overline{U}_{EA}(\bar{\theta}) = \frac{1}{n_i} \sum_{i=1}^{n_i} \sum_{j=1}^{n_j} U_{i,j}(\bar{\theta} + \frac{\Delta \theta}{2}) \tag{2.15}
\]

and the ensemble-averaged velocity fluctuation intensity

\[
u'_{EA}(\bar{\theta}) = \left( \frac{1}{n_i} \sum_{i=1}^{n_i} \sum_{j=1}^{n_j} \left[ u_{i,j}(\bar{\theta} + \frac{\Delta \theta}{2}) \right]^2 \right)^{\frac{1}{2}} = \left( \frac{1}{n_i} \sum_{i=1}^{n_i} \sum_{j=1}^{n_j} \left[ U_{i,j}(\bar{\theta} + \frac{\Delta \theta}{2}, i) - \overline{U}_{EA}(\bar{\theta}) \right]^2 \right)^{\frac{1}{2}} \tag{2.16}
\]

where \( n_i \) is the number of velocity measurements recorded in the window during the \( i \)th cycle, \( n_k \) is the total number of cycles, and \( n_i \) is the total number of measurements.
2.3. Analytical Techniques for Turbulence Intensity Estimation for an In-Cylinder Turbulence Definition

Many decomposition techniques and data smoothing methods have been proposed in the literature in order to extract the mean flow component in each engine cycle and at each location from the measured instantaneous velocity. For all methods, a 'high' data rate is required. Most techniques can be classified into one of three categories, namely;

1. Cycle-Resolved or Ensemble-Averaged Data Filtering.

This requires a form of low-pass filtering with the assumption that above some given cut-off frequency, all flow is considered turbulent (high frequency) and that below it, the contribution is from cycle-to-cycle variations in the mean flow (low frequency). The choice of this frequency threshold will have a significant influence upon the amount of low frequency turbulence that is considered as high-frequency cyclic variation. In addition, the choice of cut-off frequency and its effect on the partitioning can be greatly affected by piston location (crank angle). Often the 'cut-off' frequency is stated explicitly or, as is more often the case, it is implicit within the type of averaging intervals or data 'manipulation' used.

2. Autocorrelation Estimators.

Decomposition is achieved by the comparison of the experimentally measured autocorrelation functions with those of modelled 'expected' autocorrelation functions.


This technique requires the determination of the turbulent component by the use of an additional parameter (event or pattern) that can be assumed to be directly affected by the in-cylinder flowfield (e.g. rate of flame propagation, burn angles).

A précis of the significance of these techniques to the experimental measurement of in-cylinder flows is presented in the following section. Far more detailed analyses and texts devoted to the more advanced methods are given in Hilton (1992), from which this section has been summarised.

2.3.1. Cycle-Resolved or Ensemble Averaged Data Filtering.

Hilton (1992) divided cycle-resolved data filtering techniques into four categories based broadly upon the criteria utilised to distinguish a meaningful non-stationary component of the instantaneous velocity field. The notation used therein is followed throughout this section.

1. Non-Stationary Time-Averaging (NTA)

This approach assumes that over some arbitrary time interval within the engine cycle, the turbulence can be considered stationary and that the classic Reynolds decomposition is valid over that interval. The instantaneous velocity comprises an ensemble-averaged mean velocity, an in-cycle time-averaged mean velocity (arbitrary time interval) and a fluctuating velocity component. The non-stationary contribution is then evaluated from the difference between the instantaneous velocity and ensemble-averaged velocity over the specified time interval.
2. Time-Average Filtering (TAF)

Within this technique, data from a given cycle (or commonly over many cycles), distributed in equal crank angle intervals, is averaged across that interval. A mean velocity curve is then ‘fitted’ to the averaged crank angle window values at the mid-interval point using a curve-fitting routine. The choice of curve-fitting routine or the amount of data ‘smoothing’, ‘weighting’ or data ‘padding’ is subjective. The estimated turbulence intensity within that cycle is then computed from the difference between the measured and ‘fitted’ mean values.

3. Inverse Fourier Transform (IFT)

The procedure follows that of the TAF method. In-cycle velocity measurements are averaged across suitable crank angle intervals. The frequency spectra of the velocity measurements across the cycle is obtained by taking the Fourier transform of the mean values. Subsequently, a ‘cut-off’ frequency is assigned to separate low and high frequency fluctuations. This is often stated explicitly, based upon the analysis of typical frequency spectra for a single cycle or ensemble average from another fully-developed, stationary, characteristic turbulent flow (e.g. Moriyoshi et al., (1993a) and Brereton and Kodal (1994)) or from the analysis of a combustion related timescale (Dimopoulos et al., 1997)). The upper limit of this frequency spectra is generally taken as the ‘cut-off’ value. All values within the frequency spectrum that lie above this threshold are then set to zero and the inverse transform is applied to yield the in-cycle bulk velocity. As with the TAF techniques, the turbulence intensity is estimated from the difference between the bulk velocity characteristic and the instantaneously measured value within that cycle. An ensemble-averaged variation in the turbulence intensity (at a point in the cycle) can be estimated from the ensemble-averaged turbulence intensity curves for all cycles.

4. Linear Trend Removal (LTR)

This procedure requires the choice of a suitable time interval within a given engine cycle where the measured time-averaged mean velocity remains constant and can be considered as the mean value. A LTR procedure is then utilised to compute the difference between the mean and instantaneous velocities within this period. The computed velocity fluctuation averaged over the time interval represents the time-averaged turbulence intensity and can be ensemble-averaged over the total number of cycles.

In practice, most recent reported analyses of experimental in-cylinder measurements utilise a combination of the cycle-resolved filtering techniques outlined above. Often, ‘cut-off’ frequencies are stated with little or no justification or the criteria for filtering is subjective and not based upon an identifiable physical process. Alternatively, the cut-off frequency is varied until a linear variation is observed in the mean velocity at a point (or number of points) with engine speed.

Moriyoshi and Muroki (1995) calculated the mean flow velocity using the ensemble-averaging technique (excluding a period of ‘data-hold’) and the turbulence intensity, by the ‘moving-average’ method, with a cut-off frequency of 100 Hz. The cycle-to-cycle variation in turbulence intensity was estimated from the power spectrum analysis of Moriyoshi et al., (1993a). Corcione and Valentino (1991) and Dimopoulos et al., (1997) estimated the in-cycle mean velocity by means of a square, ‘moving-average’ window in the time domain shifted through the whole cycle. The width of the averaging window is dictated by the choice of cut-off frequency.
A weighted sum is computed over \( N \) total engine cycles for all instantaneous velocity values at equi-spaced angular crank angle intervals \((\Delta \theta = 0.2\, ^\circ)\). Using the above notation, the filtered velocity in the \( i^{th} \) cycle is given by,

\[
U_i(\theta, i) = \sum_{k=-L}^{k=L} U(\theta - k, i) w(k) \tag{2.17}
\]

for \( N \) total cycles where \( N = 2L + 1 \) and \( k \) is an integer value. The rectangular weighting function is given by:

\[
w(k) = \begin{cases} 
1 & \text{for } k \in (-L, L) \\
0 & \text{otherwise}
\end{cases} \tag{2.18}
\]

The length, \( L \) of the weight (smoothing) function determines the 'cut-off' frequency, \( F_c \). Corcione and Valentino (1991) used the end of the first decay of the ensemble power spectral density function with a Fast Fourier Transform and square window function technique. This procedure effectively partitions the flowfield between the filtered in-cycle velocity and a high frequency fluctuation term relative to the cut-off frequency. Within an ensemble-averaged approach, the intensities of the ensemble high frequency fluctuations are given, as before, by the RMS about the ensemble-averaged filtered velocity.

For the end of the compression stroke, Baby and Floch (1997) used 1500 Hz, corresponding to an 8° CA 'moving-average' window. Kudou et al., (1992) utilised 300 Hz, for a window of 10° CA in a similar combustion system. Corcione and Valentino (1990) observed a greater dependency between the cyclic variation and the choice of 'cut-off' frequency in a Diesel swirl chamber suggesting that critical appraisal of this partition was crucial to all studies. St. Hill et al., (2000) used a low-pass Hamming window filter with cut-off frequency of 250 Hz. Filtered and unfiltered mean velocity's in the vertical direction, near the cylinder centre axis and within the intake jet region showed very close agreement. This suggested that in this particular case, the choice of cut-off frequency was valid as it had no great effect on the bulk flow features in these regions and over these ranges. Measurements were available in the compression stroke up to 60 CA BTDC firing. The ensemble-averaged RMS, cycle-to-cycle variation in the mean flow and the cycle-resolved turbulence remained approximately constant in both regions. The magnitude of the ensemble-averaged RMS, in some spatial locations, was twice that of the cycle-resolved 'true' turbulence.

Dimopoulos et al., (1997) were first to propose a 'cut-off' frequency based upon characteristic scales of physical or chemical subprocesses of the combustion system. Those terms that related to 'low' frequency motion or 'high' frequency turbulence were effectively defined by the time-scales of the real processes chosen (time to ignition, rate of flame propagation and time for largest droplet evaporation). By utilising the time for premixed flame propagation in SI combustion chambers, they estimated that an appropriate 'cut-off' frequency was of the order of twenty times the engine frequency. They applied 'cut-off' frequencies of 200, 333, 500 and 1000 Hz to LDA data obtained in a two-valve, pancake chamber with variable swirl and tumble. The results clearly show the effect of appropriate determination of the 'cut-off' frequency and indicated a degree of anisotropy in the filtered turbulence intensities in the axial, radial and tangential directions that was not apparent in the ensemble-averaged turbulence intensities.
The axial component was higher in each case by as much as 30% of the radial and 40% of the tangential values. This however, was not apparent at 1000 rpm until at least a 'cut-off' frequency of 350-400 Hz was used. They concluded that below this boundary, the observed higher tangential ensemble turbulence intensities (i.e. with 'cut-off' frequency of zero) must include a component-specific, low frequency contribution from the cycle-to-cycle variation. In addition, the high frequency turbulence intensities did not increase linearly with engine speed in contrast to the intensities of the fluctuations about the ensemble-averaged mean velocity.

The above studies represent only a few of the many and varied approaches. Generally, the proposed model for instantaneous in-cylinder velocity,

\[ U(\theta, i) = U_{EA} + u(\theta, i) \]  \hspace{1cm} 2.19

is replaced by

\[ U(\theta, i) = U_{EA} + [u_{LF}(\theta, i) + u_{HF}(\theta, i)] \]  \hspace{1cm} 2.20

where \( u_{LF} \) and \( u_{HF} \) represent the in-cycle low and high frequency velocity fluctuations respectively about some arbitrary cut-off frequency. The filtered, in-cycle, mean velocity is that due to those 'lower frequency' terms about the ensemble-averaged mean,

\[ U_{LF}(\theta, i) = U_{EA}(\theta) + u_{LF}(\theta, i) \]  \hspace{1cm} 2.21

It has been proposed (e.g. St. Hill et al., 2000), that the 'true' turbulence intensity is then

\[ u_{1HF}^{2} = \sqrt{\left(u_{HF}^{2}\right)^{2}} = \sqrt{\left[U(\theta, i) - U_{LF}(\theta, i)\right]^{2}} \]  \hspace{1cm} 2.22

and the cycle-to-cycle variation in the mean as that given by

\[ u_{1LF}^{2} = \sqrt{\left(u_{LF}^{2}\right)^{2}} = \sqrt{\left[U_{LF}(\theta, i) - U_{EA}(\theta)\right]^{2}} \]  \hspace{1cm} 2.23

### 2.3.2. Autocorrelation Estimators.

This technique, also known as the phase locked model, makes use of a model for 'perceived' or 'expected' in-cylinder velocity measurements to differentiate between turbulence and cycle-to-cycle variations in the instantaneous velocity measurements. The expected flow measurements are described by a non-stationary stochastic process and a function determined from a physical property suggested from prior knowledge of the flow (from derivation of the autocorrelation function of the measured flow). Based upon this discrimination, an expected autocorrelation function is derived and compared with that obtained experimentally. In such an analysis it is assumed that the turbulence has no phase stability and that the cycle-to-cycle variation is deterministic and approximately phase-locked with the engine cycle. A small phase offset may be used for some points in the engine cycle. In summary, a simplified version of the phase-locked model of Glover (1986) and subsequently Hilton (1992) is given by:

Instantaneous velocity = Turbulence (i) + Cyclic Variation (ii) + Ensemble mean (iii)
The terms in the above equation are given by:

(i) turbulence intensity is modelled in a given cycle by a zero mean, unit standard deviation, non-stationary stochastic process. *(deterministic function \times non-stationary, stochastic process)*

(ii) cyclic variation is modelled in all cycles by a number of phase-locked, deterministic functions that vary with temporal location within the cycle and a zero mean, unit standard deviation, random number multiplier for each cycle. *(deterministic functions \times random number)*

(iii) mean velocity is modelled by a deterministic function. *(deterministic function)*

The model description of velocity measurements is complete and the expected value of the autocorrelation function of the turbulence can be computed for given turbulence and cyclic variations. Estimates of the autocorrelation function of the experimental measurements are derived by the ensemble averaging procedure. It is then possible to alter the amount of given turbulence and cyclic variation in the model that will yield the best approximation to the estimated autocorrelation function. Since the processes that describe each are different, a minimisation routine can be used to converge the model function to the estimated function. The partitioning of turbulence is thus achieved by comparison and the assumption that cyclic variation can be discriminated from turbulence in a given cycle because of its phase dependence. Application of this method to in-cylinder flow measurement using LDA have been performed by Glover et al., (1988b), Hadded and Denbratt (1991), Hilton et al., (1991), and Hilton (1992).

A highly detailed study of algorithms for estimating in-cylinder turbulent flows, methodology and treatment of autocorrelation based analyses and the implementation of autocorrelation based analysis is given by Hilton (1992).

### 2.3.3. Conditional Sampling

The sampling of experimental data for in-cylinder fluid motions, with a view to identifying the turbulent component, has been attempted using a measurable, physical parameter of the combustion system. Where possible, it is preferable that this parameter be acquired simultaneously with the flow measurements. Analysis of the results can identify 'families' of cycles that exhibited similar features. Such identifiable parameters include maximum pressure, flame front velocity and burn angles. Ozdor et al., (1996) used a series of measured pressure-related parameters to assess motored and fired engine cyclic variability.

The cyclic variability was considered sufficient to affect the intake generated flow pattern. Dimopoulos et al., (1997) presented a study on the choice of appropriate 'cut-off' frequencies for cycle-resolved LDA velocity measurements based upon combustion related parameters for both SI and Diesel combustion systems. They chose the flame propagation in a pre-mixed charge through the combustion chamber as a key 'measure' of the thermal and chemical processes in turbulent engine combustion. The reciprocal of time required for the flame to propagate through the combustion chamber was used to estimate a characteristic frequency.
They assumed that any turbulent eddies with lifetimes greater than the flame propagation time would act in 'drawing-out' or convecting the flame and would not significantly influence the convolution and mixing of the flame front. Hence frequencies above this threshold could be considered as those due to 'true' turbulence in a cycle-resolved analysis.

2.4. Frequency and Wave Number Spectra

Turbulent motion can be considered as a complex pattern of random, eddying, oscillations of a range of amplitudes and frequencies. The distribution of the amplitudes of the fluctuations over the different frequencies is termed the spectrum. The analysis of the fluctuating energy may be carried out in the time domain (frequency spectrum) or the space domain (wave number spectrum) depending upon the type of correlation chosen. Applying a Fourier transform is a means by which the complex, random waveform of turbulent motion can be represented by the sum of sine waves of various amplitudes and frequencies. The transform of the correlations established previously describe the distribution of the energy of turbulence over the frequencies of fluctuations, i.e. identification of those frequencies which contribute more or less to the total energy spectrum.

Generally, Fourier series are used to represent a periodic, continuous (or discontinuous) function as an infinite trigonometrical series in sine and cosine terms. Turbulence is a non-periodic function and the Fourier integral transform must be used with infinite limits for an exact solution. The Fourier integral transform of the covariance term used in the Eulerian space correlation coefficient is called the energy spectrum tensor given by

\[
\Phi_y(k) = \left(\frac{1}{8\pi^3}\right) \int_{-\infty}^{\infty} u_i(x)u_j(x+r)e^{-j(k \cdot r)} dr
\]

\[
u_i(\tilde x) = \int \Phi_y(k)e^{j(k \cdot r)} dk
\]

where \( j \) in the exponent is \( \sqrt{-1} \) and \( k \) is the wave number vector, analogous to \( r \) in 'Eulerian space', but in 'wave number space' (Brodkey, 1967). The term \( dk \) is an infinitesimal element of volume in wave number space about the vector, \( k \). The components of wavelength and frequency of the wave number vector are given by,

\[
k_i = \frac{2\pi}{\lambda_i} = \frac{2\pi f_i}{U_i}
\]

The transform is referred to as the wave number spectrum and for this particular example, it is the \( x \)-component wave number spectrum of the \( u \)-component velocity. The three-dimensional wave number spectrum can be defined in a similar way. From the physical model of vortex stretching and energy cascading, it can be assumed that turbulent energy is transferred from low wave numbers (large wavelengths) to high wave numbers (small wavelengths). Low wave numbers represent the larger energy carrying eddies, whereas the high wave number range is concerned with dissipation into internal energy.
A number of research groups have employed wave numbers and wave number space to evaluate energy partitioning between turbulence at high wave numbers and cycle-to-cycle variations thought to be represented by lower wave numbers. In more advanced studies, use is made of power spectrum profiles as observed in nearly isotropic flow fields where cycle-to-cycle variations do not occur (Moriyoshi et al., (1993a,b)). This is illustrated by the turbulence power spectrum model of Hinze (1975) as shown in Figure 2.2. The profile at one spatial and temporal location has one frequency peak (the energy containing eddy) and decays logarithmically in the frequency range above and below this threshold. It is then assumed that the high Reynolds number, in-cylinder flow fields at TDC are governed by the Kolmogorov $^{-5/3}$ power domain region of the power spectrum in the high wave number region. The values of $\epsilon$, the dissipation rate of turbulent energy and $k$, the turbulent kinetic energy, are calculated from the turbulence power spectrum, without the need for a defined cut-off frequency and observing the above Kolmogorov relationship. Figure 2.3. shows the calculated power spectrum of the instantaneous velocity observed over 10 CA's (94-84 CA BTDC) at one point, 15 mm from the cylinder axis and 10 mm from the gas face. The turbulence decay would appear to follow the $-5/3$-power line. The turbulence power (at a given frequency) and Equation 2.26 (to convert to wave number space) are used to derive the dissipation terms by rearrangement of the Kolmogorov relationship. A more comprehensive discussion of this theory, its limitations and comparison with more conventional 'cut-off' frequency analyses is given by Moriyoshi et al., (1993a).

St. Hill et al., (2000) presented normalised power spectra which appeared to exhibit significant peaks at harmonics of the camshaft frequency during the intake stroke in the intake jet and along the cylinder axis in the vertical component of velocity. Where measurements of velocity are taken at two points in space, a spatial cross-correlation function of the two velocity records can be estimated for various separations. Benak et al., (1993) reviewed several estimators for cross-correlation functions as applied to discontinuous LDA measurements to obtain the cross-spectral density estimates. In all cases, however, the spectral analysis of LDA data is complicated by the random sampling of LDA data (non-stationary mean, RMS, frequency content and data rate). It is therefore necessary to employ more advanced techniques to extract the power spectral density function estimate directly or to estimate over very small crank angle intervals where steady behaviour can be assumed. Broadly, these are categorised by Grosjean et al., (1997) as either:

1. Computation of the autocorrelation function using the 'slotting' technique as originally proposed by Roberts (1976), (e.g. Glover et al., (1999b), Hilton et al., (1991), Hilton (1992)).
2. Direct transform methods and stochastic modelling (e.g. Roberts and Hilton (1999) and Brereton and Sasak (1999))
3. Prediction model based upon power spectrum analyses. (e.g. Moriyoshi et al., (1993a,b))
2.5. Intake Generated Tumble Flows

Multi-dimensional in-cylinder airflows are generated in the intake phase of the four-stroke engine cycle. Intake port and combustion chamber geometry's are utilised to establish a defined bulk flow motion. Spark ignition combustion chambers generally rely on a tumble flow to effectively fill the chamber with charge, mixing and to promote suitably turbulent conditions towards the end of the compression stroke. These large vortical structures are present throughout the intake stroke and exhibit both translational and rotational motion. A measure of the rotation of the structure about an axis perpendicular to the plane of rotation is given by the vorticity, $\xi$. Where no distortion of the fluid element occurs (i.e. rate of shear strain is zero), vorticity is defined as twice the angular velocity of the element.

From the control volume element $dxdydz$, of Figure 2.0., with solid body rotation in the $xy$-plane, vorticity is defined as

$$\xi = \left( \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right)$$

2.27

If the fluid element is then subjected to a rate of strain in the $z$-direction, the element will be stretched in this direction and its cross-sectional diagonals in the $xy$-plane will be reduced as the element gains height. The conservation of angular momentum applies and thus during the stretching process the kinetic energy of $w$, the $z$-component velocity gives up some energy to the kinetic energy of rotation. This will decrease the degree of the distortion in the $xy$-plane. The velocities of the $x$ and $y$ components have been increased and the length scales decreased in the $xy$-plane with the application of a rate of strain in the $z$-direction. The increased velocities in the $x$ and $y$ directions then, in turn, stretch other rotating elements of fluid (or vortices) in their respective directions and the cycle repeats. At each stage, the length scale decreases and the motion in the other two components is intensified. The energy transfer continues with smaller and smaller eddies until the molecular fluid viscosity eventually dissipates the remaining energy into internal energy as heat. At this point, turbulence is isotropic throughout the fluid with equal amounts of eddies in all directions. Kolmogorov introduced eddy length and time scales for these small eddies, relating kinematic viscosity with the rate of energy dissipation per unit mass.

The process of vortex stretching occurs within the cylinder from the instant the intake valves open. The intake conditions, particularly the maximum piston speed and maximum inlet valve lift opening (~75 -105 °CA), control the generation of these flows. The magnitude of the bulk fluid motions is proportional to the flow velocity through the area between the valve and seat and scales with mean piston speed or engine speed where ordered mean motions are observed (Liou and Santavicca (1983), Heywood (1988)). In tumble flows, the size of the vortices induced by the intake ports are limited by the piston position at inlet valve closure (IVC) and by the cylinder bore. Likewise, three-dimensional vortex stretching is bounded with large scale vortex stretching limited in certain directions. It is these large scale structures that can best interact with the mean flow and contribute most to the Reynolds's stresses that govern the rate of supply of kinetic energy to the turbulence.
Turbulence in intake-generated flows is predominantly anisotropic. The turbulence intensity rises to a maximum value at maximum valve lift before decaying rapidly until valve closure. The nature of the mean flow at the onset of the compression stroke is dictated by the large-scale structures defined by the intake flowfield. Swirl motion generally sustains rotation up to TDC and is more effective than tumble in suppressing cyclic variations as observed experimentally in many engine studies (e.g. Reuss et al., 1989; Moriyoshi and Muroki, 1995; Dimopolous et al., 1997; Jaffri et al., 1997; and Baby and Floch, 1997). Conventional tumble flows however generally decay rapidly during the intake phase and few large-scale structures persist during compression. It is not until the late compression stroke that the magnitude of the turbulence intensity increases. There is a rapidly decreasing volume for competing eddies and distortions are induced in the fluid by the motion of the piston. Reuss et al., (1989) observed high strain rate in shear layers formed between competing flow regions and smaller incursions in the mean, spatial flow field. These features were randomly, spatially distributed from cycle to cycle and evident at up to 12 °CA ATDC. As the piston approaches TDC, radial, inward squish flows enhance the turbulence intensity and further turbulence is generated through the piston interacting with the boundary layer established on the cylinder liner wall.

In the absence of a bulk swirl motion, experimental studies often report that the turbulence at TDC can be described as homogeneous and isotropic. Heywood (1988) confirms the consensus of many research groups; for open combustion chambers without swirl, the ensemble averaged turbulence intensity fluctuation at TDC is approximately equal to one half of the mean piston speed. The mechanism of vortex breakdown would appear to be most affected by the initial tumble ratio, the rate of vortex compression and by the combustion chamber geometry (Arcoumanis and Whitelaw, 1995).

As yet, there is no presented experimental data for the high tumble chambers commonly found in top-entry G-DI systems. Pre-conceptions about the optimum flow characteristics for ignition in stratified charge combustion systems, based upon existing pre-mixed, homogeneous combustion models, require reconsideration. The modifications to the geometry of the intake ports and the combustion chamber are required to conserve the mean tumble pattern up to and beyond the point of fuel injection. Turbulence intensity values will then increase through the action of shear, as the conservation of angular momentum will require the angular velocity of the large-scale flow structures to increase. The degree of isotropy and turbulence homogeneity will control the fuel/air mixing and the burning rate. It is then assumed that the turbulent flow field in front of the propagating flame front is apparently unaffected by compressibility if the pressure fluctuations within the turbulence are small compared with the absolute pressure.

2.6. Space and Time Correlations

In practice, most in-cylinder fluid flows cannot be described as homogeneous. That said, in all turbulent flow situations it is necessary to make assumptions in order to establish time and length scales that can best quantify the eddy size and lifetime of the eddy. Such characterisation of in-cylinder turbulence at this level is fundamental to the future development of theoretical models.
The largest eddies are limited by the system boundaries (e.g. port jet flow thickness, maximum valve lift and valve inner seat area) and the smallest extend into the regions of molecular diffusion. It was Taylor, (1938) who first suggested that a high degree of correlation exists between the same fluctuating quantities measured at two different points in space, if the separation between the points is small compared to the diameter of the eddy. There exists a classical statistical procedure often applied to IC engine flow analysis to define the degree to which these turbulent fluctuations are random or deterministic. The dependence of two random variables (the velocity as a function of position, measured at each point) can be measured by the covariance and a coefficient of linear correlation, (Wackerly et al., (1996), Chapter 5). For a random variable, such as velocity, at two points separated by a distance vector, \( r \) (points \( x \) and \( x+r \)), then the covariance of the \( u \)-component is defined as \( u_i(x)u_j(x+r) \) in Cartesian coordinates. A linear coefficient of correlation, the Eulerian correlation function (spatial velocity autocorrelation function), is related to the covariance and defined as:

\[
R(r) = \frac{u_i(x)u_j(x+r)}{\sqrt{u_i^2(x)u_j^2(x+r)}} \text{ or } \frac{u_i(x)u_j(x+r)}{u_i'(x)u_j'(x+r)}
\]

which can be shown to satisfy the relationship \(-1 \leq R(r) \leq 1\), where the bounds represent a perfect correlation with all points falling on a straight line. Figure 2.4. illustrates the form of such a spatial velocity autocorrelation as a function of separation distance. Simplified expressions can be derived for correlations in homogeneous and isotropic turbulence in both the longitudinal (same direction as the space vector, \( \mathbf{j} \)) and lateral (normal to the space vector, \( \mathbf{j} \)) directions (Brodkey, (1967)).

Numerical approximations rely on a large number of measurements, \( n \) being made simultaneously at two points in the flow. The Eulerian or spatial autocorrelation function is then estimated by:

\[
R(r) = \frac{1}{n-1} \sum_{i=1}^{n} \frac{u_i(x)u_j(x+r)}{u_i'(x)u_j'(x+r)}
\]

For isotropic turbulence, the RMS fluctuations are all equal by definition and the equation reduces to

\[
R(r) = \frac{u(x)u(x+r)}{u^2}
\]

The sign of the correlation coefficients at two points indicates the 'direction' of the variables dependence. Positive values of \( R(r) \) from 0 to \( r_{\text{max}} \) suggest some degree of correlation or decay of the flow over a long period. Negative values of \( R(r) \) suggest that the distance \( r \) (or delay time in Lagrangian terms) for which fluctuations are correlated is that at which \( R(r) \) has a minimum. The integral of the spatial velocity correlation coefficient is termed the 'integral' or Eulerian length scale, \( l_i \) and is a measure of the largest scale structures in the flow field,

\[
l_i = \int R(r) \, dr
\]
Both longitudinal (with the direction of the flow) and lateral, or transverse (normal to the direction of the flow) length scales can be defined for homogeneous, isotropic turbulence. In practice, measurements of this kind are often difficult to realise and a time scale is more readily attainable.

Glover (1988), Fansler and French (1988) and Hadded and Denbratt (1991) used an exponential function as an approximation to the spatial autocorrelation function from which to estimate the integral length scale. Measured integral length scales have been shown to scale with clearance height and relatively independent of engine speed. Limited by the system boundaries, they decrease to approximately one fifth of the clearance height at TDC. For typical IC engine geometry's, this is of the order of several millimetres. Table 2.1. illustrates the range of direct integral length scales estimated by a number of research groups using experimental measurement techniques in different combustion systems. The number of studies is limited by the complexity of obtaining such measurements with fidelity.

The above Taylor hypothesis makes use of the Eulerian approach to fluid flow analysis; that is, the spatial correlation function, $R(r)$, is a correlation between the instantaneous velocity fluctuations separated by a distance, $r$. In contrast, the Lagrangian system of co-ordinates follows the path of a particle and the velocity fluctuations of a fluid particle are correlated at two different times along its path, such that for some time increment or delay, $\tau$, the correlation coefficient is given by

$$R_y(\tau) = \frac{\langle v'_y(t)v'_y(t+\tau) \rangle}{\langle v'_y(t) \rangle^2}$$

or for isotropic turbulence, $R(\tau) = \frac{\langle v'(t)v'(t+\tau) \rangle}{v'^2}$

The function can be numerically estimated in a similar manner to that of the spatial correlation coefficient. The Lagrangian integral time scale, $\tau_i$, is defined

$$\tau_i = \int_0^{\infty} R(\tau) d\tau$$

The integral time scale is often referred to as an indication of the lifetime of an eddy rigidly convected through the fluid. Alternatively, it can be considered as a measure of the time it takes a large eddy to pass a point without mean motion and where the turbulence is relatively weak. Integral time scales measured in IC engine combustion chambers are estimated to be of the order of several milliseconds at engine speeds of 1000 rpm at the end of the compression stroke. These scales decrease for higher engine speeds. Several Lagrangian length scales can be defined based upon the time scale and a Lagrangian mean or rms velocity. It is however important to note the $v'$ in the Lagrangian system, is the rms velocity fluctuation of many particles averaged over their respective paths, and not a time-averaged component at one given point.

Taylor’s hypothesis suggests that in turbulent flows, the rate of change of shape of eddies is slow compared to the mean velocity, such that $u'$ is much less than $\overline{U}$. The eddies are ‘rigidly convected’ or transported by a ‘convection velocity’. The turbulent eddies do not appreciably change their shape as they pass a given point (homogeneous, steady state). If an Eulerian space correlation had a separation related to time, $x = \overline{U}t$ in the x-direction then an autocorrelation exists where the space co-ordinate can be replaced by an equivalent time (Brodkey, 1967)).
This is given by

\[ f(r) = \frac{u(x)u(x+r)}{u^2} = \frac{u(t)u(t+\tau)}{u^2} = f(\tau) \text{ and } l_T = \overline{U_T} \]  \hspace{1cm} (2.34)

It is important to note that the above expression relates to an average of the product of the velocities, \( u \) taken at two times from one point (an Eulerian time correlation). Although of similar form, the Lagrangian coefficient is the average of the product of the velocities, \( v \) taken at two times along particle paths (the velocities along those paths).

The above isotropic turbulence expression lends itself readily to experimental non-intrusive optical measurement techniques where disruption to the flow is eliminated and is a good measure of the accuracy of the above hypothesis. However, typical in-cylinder flows exhibit intense shear flows with high turbulence intensity and it is the departure from the Taylor hypothesis that becomes of particular interest. Eulerian space correlations will become asymmetrical in a given direction should the largest turbulent eddies be changing appreciably in that direction (Bradshaw, (1971)). For stationary turbulent flows, the autocorrelation is always an even function for positive and negative time delays. It is those smaller eddies (fed by the continual breakdown from those of the larger scales) that exhibit more isotropy as they respond more readily to bulk flow pattern changes. Further correlations introduced to the text will be based upon the above fundamental space and time correlations.

### 2.6.1. Taylor Microscales

Taylor defined the Taylor micro length and time scales by assuming that the rate of strain of a fluid in a turbulent flow field was proportional to its turbulence intensity

\[ \frac{\partial U}{\partial x} = \frac{u}{l_m} \]  \hspace{1cm} (2.35)

where \( l_m \) is the Taylor length scale determined from the gradient of the spatial autocorrelation curve at \( x=0 \) as indicated in Figure 2.4. The micro time scale, \( \tau_m \) is determined from the temporal autocorrelation function such that (Hinze (1975))

\[ \tau_m^2 = \frac{2}{\left( \frac{\partial^2 R(t)}{\partial t^2} \right)_{t=0}} \]  \hspace{1cm} (2.36)

For homogeneous, isotropic turbulence,

\[ l_m = \overline{U_T} \tau_m \]  \hspace{1cm} (2.37)

Experimental measurements have confirmed similar trends to those observed in the integral time and length scales. The Taylor micro time scale is reduced with increasing engine speed whilst the Taylor micro length scale shows no significant change. The time scale has been estimated at TDC firing to be of the order of tenths of milliseconds at 1000 rpm. Under the same conditions, the length scale is of the order of 0.5-2.0 millimetres.
2.6.2. Kolmogorov Length Scale

From the above definitions of spatial and temporal scales in turbulent flow, it is possible mathematically or empirically to establish useful relationships between the scales from the momentum diffusion scale of Kolmogorov, to the microscale of Taylor, to the integral scales. Kolmogorov related the rate of energy dissipation per unit mass, $\varepsilon$, and the kinematic viscosity, $\nu$, to the Kolmogorov length scale, $l_k$.

$$l_k = \left( \frac{\nu^3}{\varepsilon} \right)^{1/4} \quad 2.38$$

Kolmogorov length scales have been reported in the order of 0.01 mm for TDC conditions at 1000 rpm in IC engine combustion chambers.

It has been assumed that the large scale eddies (that govern the integral scale region) must carry by far the largest part of the energy distribution. The kinetic energy per unit mass of these eddies must be proportional to the square of the fluctuation intensity.

A typical time taken for these eddies to give up energy (turnover time) is dependent on $l_i/\nu'$. Therefore for conservation of energy per unit mass, the rate of energy supplied by the turnover of an eddy of the integral scale must be approximately equal to the rate of dissipation of that energy into internal heat at a molecular level

$$\varepsilon \approx \frac{u'^2}{t} = \frac{u'^2}{l_i/\nu'} = \frac{u'^2}{l_i} \quad 2.39$$

and

$$\frac{l_k}{l_i} = \left( \frac{u' l_i}{\nu} \right)^{3/4} = Re_T^{-3/4} \quad 2.40$$

where $Re_T$ is defined as the turbulent Reynolds number (based upon the integral length scale). For homogeneous, isotropic turbulence

$$\frac{l_m}{l_i} = 3.873 Re_T^{-0.5} \quad 2.41$$

and

$$\frac{l_m}{l_k} = (225 Re_T)^{1/4} \quad 2.42$$

2.6.3. Evaluation of Direct and Indirect Turbulence Length Scales from Flow Measurements

2.6.3.1. Indirect Integral Length Scale Estimation

The aforementioned analyses of ensemble averaged or cycle resolved velocities and velocity fluctuations could be used to evaluate the integral length scale of in-cylinder turbulence.
Often, the methods of evaluation are divided into two distinct groups dependent on the type (and interpretation) of the experimental measurements used. The most common of these reported in the literature refers to the indirect evaluation of length scales from the integral time scale using numerical integration of the autocorrelation function determined from single point measurement techniques. For a flow that exhibits no mean motion, the time scale defines the lifetime of the largest eddy. Where a mean motion exists, the time scale describes the eddy transit time or the time taken for an eddy to be transported past a point by the mean flow. This requires several assumptions to be made about the type of flow. Generally, Taylor's hypothesis is utilised and the flow field must obey the following criteria:

1. The turbulent flow field is considered isotropic and homogeneous.
2. The fluctuations in the velocity are very small compared to the mean flow velocity.
3. The mean flow velocity is considered stationary.

In practice, this last criterion is not readily applicable to in-cylinder flows of reciprocating engines and often a phase factor is introduced to take into account the non-stationary nature of these flows. Corcione and Valentino (1990, 1991) and Hong and Chen (1997) amongst others, proposed that indirect integral length scale could be evaluated from the following re-defined Eulerian temporal autocorrelation function of Equation 2.29 in Section 2.6, under the Taylor hypothesis, such that the space co-ordinate is replaced by an equivalent time:

\[ R_T(\tilde{\theta}, \phi) = \frac{1}{n} \sum_{i=1}^{n} \frac{u(\tilde{\theta}, i)u(\tilde{\theta} + \phi, i)}{u(\tilde{\theta}, i)u(\tilde{\theta} + \phi, i)} \]

where \( \phi \) is the phase angle with respect to the crank angle, \( \theta \). From Section 2.6., the integral time scale is defined:

\[ \tau = \int_{0}^{\phi_{\text{max}}} R_T(\tilde{\theta}, \phi) d\phi \]

where \( \phi_{\text{max}} \) is the maximum phase angle for which the velocity fluctuations are no longer correlated. This value is often difficult to predict in studies of in-cylinder flows, especially at crank angles close to those where the piston velocity is near its maximum.

The integral length scale, \( l_i \), is then evaluated from the expression

\[ l_i(\tilde{\theta}) = \left| U_{EA}(\tilde{\theta}) \right| \tau(\tilde{\theta}) \]

In order for the above expression to be valid, a strong mean flow must exist that is quasi-steady over the time interval considered. This case is not likely in conventional SI engines but may be more applicable to air-guided concepts employed in G-DI combustion chambers.

Corcione and Valentino (1990, 1991) defined the temporal autocorrelation function in each cycle in terms of the fluctuation about that cycle's mean:

\[ R_T(t, i) = \frac{u(\tilde{\theta}, i)u(\tilde{\theta} + t, i)}{u'(\tilde{\theta}, i)u'(\tilde{\theta} + t, i)} \]

where

\[ u(\tilde{\theta}, i) = U(\tilde{\theta}, i) - U_{EA}(\tilde{\theta}) \]
If the cycle is divided into $N_s$ evenly spaced windows or time bins;

$$u(\bar{\theta}, i)u(\bar{\theta} + t, i) = \frac{1}{N_s - t} \sum_{\delta = 0}^{\delta = N_s - t - 1} u(\bar{\theta}, i)u(\bar{\theta} + t, i)$$  \hspace{1cm} 2.48

$$u'(\bar{\theta}, i) = \left[ \frac{1}{N_s - t} \sum_{\delta = 0}^{\delta = N_s - t - 1} u^2(\bar{\theta}, i) \right]^{1/2}$$  \hspace{1cm} 2.49

$$u'((\bar{\theta} + t, i) = \left[ \frac{1}{N_s - t} \sum_{\delta = 0}^{\delta = N_s - t - 1} u^2((\bar{\theta} + t, i) \right]^{1/2}$$  \hspace{1cm} 2.50

Hence, the ensemble autocorrelation function comprises the sum of each cycle's temporal autocorrelation function divided by the total number of cycles, $n$:

$$R_T = \frac{1}{n} \sum_{i=1}^{n} R_T(t, i)$$  \hspace{1cm} 2.51

Again, the integral time scale is found from the integral of $R_T(t)$ from 0 to $t_{\text{max}}$ for positive values of $R_T(t)$. For negative values, the delay time over which the fluctuations remain correlated is that at which $R_T(t)$ is at a minimum. The integral length scale is then found from the time scale and the ensemble-averaged velocity component.

For a conventional disc or pent-roof combustion chamber, the experimental data of many researchers (e.g. Glover 1988a, Corcione and Valentino (1990, 1991), Dimopolous et al., (1997), Hong and Chen (1997), Baby and Floch (1997), Arcoumanis et al., (1998), St. Hill et al., (2000)) show that throughout the intake and compression strokes, the magnitude of the mean velocity and the turbulence intensity can be comparable. This would suggest a departure from Taylor's hypothesis and that the indirect method can only be used as a preliminary estimation in these systems. Indeed, Corcione and Valentino (1991) found significant discrepancies between indirect and direct length scale estimation approaches in both straight-sided and re-entrant type combustion bowls which they attributed to the assumption of Taylor's hypothesis. Their Diesel combustion system utilised variable swirl and squish flows to promote better air utilisation. The re-entrant bowl chamber was seen to conserve in-bowl swirl until after TDC. The re-entrant bowl had a higher integral time scale at TDC as compared to the straight-sided bowl. They concluded that at the periphery of the bowl, the turbulent kinetic energy dissipation process, in relative terms, was more persistent, thus aiding the air-fuel mixing process. However, at engine speeds of 600 rpm, indirect length scales at TDC, 4 mm down from the cylinder head gas face and 20 mm from the mid-cylinder axis, were more than twice that calculated using the direct length scale method.

### 2.6.3.2. Direct Length Scale Estimation

Where experimental data can be collected at two separated points in the flow simultaneously, it may be possible to estimate a direct integral length scale. This however involves complex optical methods of data acquisition and hence a limited number of published studies are available. Table 2.1. shows the results of some of these studies where orthogonal axes are defined in the lateral (cross-tumble plane) and the longitudinal (tumble plane) directions as indicated.
In this instance, the separation distance between the measurement points defines the bounds of the direct Eulerian integral length scale:

$$l_i(\overline{\theta}, x) = \int_{0}^{r^*} R(\overline{\theta}, r) dr$$

where $R$ is the lateral or longitudinal Eulerian spatial correlation function, $x$ is the position of the fixed point and $r^*$ is the distance between the changeable point and the fixed point that satisfies $R=0$. When $r > r^*$, the fluctuation velocity is no longer correlated (Hong and Chen (1997)). Typically, this procedure is performed over a fixed width crank angle window. The function then becomes the average of each correlation function taken for each of the total number of valid data points in that window. Corcione and Valentino (1991) used 1000 validated velocity values per crank angle time bin. Hinze (1975) and then Glover (1988) and Fansler and French (1988), amongst others, fitted an exponential function as an approximation to the data. Again, the comparisons between direct length scale estimation and those obtained using Taylor's hypothesis indicated that only under certain spatial and temporal conditions could the hypothesis be considered valid.

Hong and Chen (1997) presented an experimental study of in-cylinder turbulence comparing integral length scales estimated by both the direct and indirect methods in a pancake combustion chamber of a Megatech MK3 engine. They measured both lateral and longitudinal scales in the mid-cylinder plane at 13 mm from the cylinder head roof and at 500 rpm. The cylinder head was of the conventional two valve and side ported type. When plotted against crank angle, the direct and indirect methods exhibited different magnitudes and trends. They suggested that the indirect method underestimated the length scale during the closed period in the cycle yet over estimates during the open period. However, they point out that at TDC firing, both quantities are of the same magnitude. Importantly, they conclude that the indirect method of integral length scale evaluation offers a good, preliminary quantitative estimation under these conditions. Direct integral length scales were shown to peak at IVC of approximately 200CA. In this instance, the lateral scale was 1 mm greater than the longitudinal scale estimated at 7 mm. Towards TDC, the spatial correlation coefficient was seen to increase slightly and the longitudinal scale was estimated at 7 mm, whereas the lateral scale had dropped to 5 mm. In Table 2.1., lateral integral length scales are shown to be consistent across the studies but due to the complexity of obtaining such measurements with confidence, it is not possible at this point in time to make valid predictions about integral length scale estimations based upon such a small data set.
2.7. Turbulent Energy

The distribution of turbulent energy is an important factor in the analysis of in-cylinder fluid flows. From the analysis of Reynolds stresses, the sum of the principal stresses (defined about a set of principal axes to ensure only normal stresses exist) is \(-\rho \bar{q}^2\), where \(\bar{q}^2 = \bar{u}^2 + \bar{v}^2 + \bar{w}^2\) as defined in the previous notation. The turbulent (kinetic) energy per unit volume is defined, \(\frac{1}{2} \rho \bar{q}^2\).

This is the total turbulent energy associated with all energy transport, source and sink terms in the control volume \(dxdydz\) of Figure 2.0. The sum of the transport terms into and out of the control volume (spatial gradients of mean turbulence quantities) is equal to the sum of source and sink terms for conservation. For completeness, these can be briefly identified:

- **Mean energy transport (Advection)**

  The rate of transport of the mean \(\bar{q}^2\), in the \(x\)-direction is \(\bar{q}^2 \frac{U}{\bar{q}^2} + \bar{q}^2 \frac{U}{\bar{q}^2}\). The net rate at which \(\frac{1}{2} \rho \bar{q}^2\) is leaving the control volume is the sum of the \(x, y,\) and \(z\) terms, of which the \(x\) term is given by

  \[
  \rho \frac{\partial}{\partial x} \left(\frac{1}{2} \bar{q}^2 U + \frac{1}{2} \bar{q}^2 U\right) dxdydz
  \]

- **Turbulent energy transport**

  The work done in transporting an element of fluid through a region of fluctuating pressure results in a loss of turbulent energy from the control volume.

- **Turbulent energy production / loss terms**

  The transfer of energy from the mean flow by the turbulence. This is governed by the rate at which work has to be done against the \(x\)-component of Reynolds normal stress such that an element of fluid \(dxdydz\) is stretched in the \(x\)-direction at the rate \(\frac{\partial U}{\partial x}\).

- **Turbulent energy viscous dissipation**

  At the smallest scale, this refers to the transfer of energy between the turbulence and the molecular motion by viscosity. Turbulent kinetic energy is dissipated into internal thermal energy. It should be noted that dissipation of mean flow energy, where velocity gradients are much smaller than those of the turbulent fluctuations, is significantly reduced in magnitude.

- **Viscous transport of turbulent energy**

  These are considered negligible for all flows except near a solid surface.

  Of the five terms, turbulent energy production (work done against Reynold's stresses) and viscous dissipation are generally the largest although mean and turbulent energy transport (diffusion and advection) become more significant near the edge of a shear layer. This will occur in the high shear region formed when intake flows mix in dual intake valve engine configurations. Hascher et al., (1997) evaluated turbulent kinetic energy in a four valve SI engine based upon comprehensive, three-dimensional, LDA measurements performed at 600 rpm.
The volume-averaged turbulent kinetic energy was calculated over the effective in-cylinder volume assuming that the 3-D measurement grid defined a volume cell about each measurement point. Importantly, they concluded that the decay of volumetric turbulent kinetic energy (per unit mass) in the crank angle region of 60CA-270CA could be represented very closely by an exponential function $K$, of crank angle, $\theta$, given by

$$K = 90e^{-0.0198\theta + 1.3}$$

The combustion system observed was that of a conventional four-valve pent-roof gasoline engine with a predominantly tumbling air motion. Some swirl was observed as tumble axis ‘tipping’ and tumble centre precession that contributed to increased turbulence levels and cyclic variability (turbulent kinetic energy generation terms).

The experimental results of Kudou et al., (1992), Jaffri et al., (1997), Hascher et al., (1997) and Nadarajah et al., (1998), suggest that within the range of observed crank angles and for this type of ‘high’ tumble cylinder head; the turbulent kinetic energy is predominantly governed by the transfer of energy from the mean flow by the turbulence as work done against the components of the Reynolds stresses and less so by the viscous dissipation term in the presence of such high intake generated velocity gradients. The tumble study of Kudou et al., (1992) observed the transfer of turbulent energy to the spark plug region from 40-10 CA BTDC (in proportion to the turbulence intensity). In this region, the tumble ‘breakdown’ process and the initial burn periods occur. Through enhancing the tumble intensity, they claimed a reduced initial burn period due to an earlier observed maxima in the turbulent kinetic energy and a more rapid decay with time. They claimed that this type of turbulent energy promotion, whilst reducing the initial burn period, would not be to the detriment of the main burning velocity and that it was the rate of decay of turbulent kinetic energy that was a fundamental key to combustion controlling strategies in lean burn engines.

However the findings of Hadded and Denbratt (1991) reveal, to the contrary, that within four-valve, pent-roof configurations, the increase of tumbling motion does not result in turbulence enhancement in the late part of the compression process due to early decay of the turbulence. This is believed to be due to the interaction between intake valve jet flows resulting in secondary vortex generation. Such a combustion system clearly showed a bi-modal distribution of turbulence with initial vortex breakdown at approximately 110 CA BTDC resulting in reduced turbulence generation at TDC. Peak turbulence intensity was observed closer to TDC for the pent-roof chamber (20 CA BTDC) opposed to a disc shaped chamber with tumble flow.

### 2.8. Measurable Characteristics of an In-Cylinder Turbulent Fuel Spray for a G-DI Engine

The investigation of turbulent gasoline fuel sprays in G-DI combustion systems represents an important current area of research (Zhao et al., 1997). To date, the literature contains extensive data devoted to the experimental and computational studies of ‘direct injection’ sprays limited to Diesel engines.
However despite many well-developed and validated models, these studies are not readily applicable to gasoline engines where the physical properties and injection requirements and chemistry are fundamentally different. In addition, the knowledge gathered from the numerical and experimental studies of closed-valve and open-valve manifold injection regimes is difficult to transfer to G-DI and may only be used as a tentative starting point for G-DI studies.

Central to characterising high-pressure gasoline injectors and the effect of air-fuel mixing is the understanding of the significance of the local airflowfield upon the structure of a homogeneous or stratified mixture through an accurate description of the interaction between the continuous (air) and dispersed (fuel) phases present. Air interaction with the two phase turbulent jet flow of liquid and vapour fuel plays a fundamental role in the preparation of a suitably combustible mixture. Turbulence helps to disperse the liquid droplet phase and it is the residence time of these droplets in the gas phase eddies that will define the rate of evaporation of the injected spray. In addition, the injection of a liquid mass imparts momentum to the surrounding gas phase with the local generation of a turbulent jet stream formed along the accelerated boundaries between the two phases. This effect leads to the entrainment of air into the spray plume along its periphery and a higher degree of air-fuel mixing.

Chapter 1 identified the features of sprays for G-DI concepts and the physical terms used to describe them. This section therefore aims to briefly introduce the currently accepted measures and quantifiable features that can be experimentally measured predominantly by the PDA technique. Figures 2.5. and 2.6. summarise the requirements of a fuel spray for a G-DI engine.

The central terms in Figure 2.5. play an important role in the development of the spray growth and the subsequent air-fuel mixing process. These are dependant upon a developed air flowfield at the time of injection and upon the requirement for increased injection pressures to provide better atomisation. The pivotal, central terms are ranked in increasing dependence upon either the developed airflow or increasing injection pressure. LASER Doppler Anemometry (LDA) and Phase Doppler Anemometry (PDA) are experimental techniques that can be used to directly measure the most important of these categories; that is, the local airflow characteristics and the fuel droplet size distribution. The spray penetration and shape are most generally measured using high-speed photography or LASER based techniques such as Particle Image Velocimetry (PIV) and LASER Induced Fluorescence (LIF). The measures most reported in the literature that are used to describe fuel spray quality by the mean droplet size distributions, are given in Table 2.2. for reference. The choice of mean droplet diameter distribution is often dependent upon its applicability to theoretical calculation or numerical modelling. Some example applications are shown in Table 2.2. It should be noted that the moment means are analogous to centres of gravity of the respective distributions. These calculations do not require knowledge of the numbers of particles measured. In general fuel spray measurements however, the number of particles (per second) is often required for statistical analysis and as an additional means of characterising the spray in the combustion chamber.
2.9. The Role of Pre-combustion Turbulence on Early Flame Development and Burn Rates in Pre-mixed SI Combustion.

In modern spark ignition engines the predominant large scale structures exist in the tumble plane. Generated in the intake stroke and by the piston boundary, they can be seen to persist, through conservation of angular momentum, well into the late compression stroke and often remain intact at the time of ignition at around 20-30 CA BTDC. The more turbulent the air motion, the greater the degree of fuel evaporation and air and fuel mixing prior to ignition. Higher flame speeds will generally lead to more rapid and complete combustion. After approximately 50 µs following spark breakdown, the flame front then propagates through a pre-mixed fuel, air and burned gas mixture until extinguished on the combustion chamber wall. This process can be sub-divided into four stages: spark ignition, early laminar flame kernel development phase, turbulent burning flame propagation and flame termination.

Following spark ignition, the high mean velocity across the spark plug gap acts on the developing flame kernel, drawing it out in a thin, conical form from the point of ignition. The rate of burning increases due to the increase in flame frontal area available for reaction. The growth of the flame front varies from cycle to cycle as a result of the varying local mixture concentration gradient dictated by the non-stationary nature of the mean fluid motion at the spark plug gap. The initial phase of early flame development is often referred to as the laminar kernel phase of combustion and is associated with a laminar flame burning velocity, \(S_L\). The rate of growth of the flame kernel (shape and propagation speed) is controlled by the flame stretching effect induced by the mean flow, rate of strain, randomly distributed and fluctuating vorticity, local mixture composition and the combustion chamber geometry. In quiescent and swirl type combustion chambers, the flame front then propagates in a spherical manner, in the absence of a mean flow, which is governed by the location of the spark plug and the bounds of the piston, cylinder head and cylinder liner walls. The surface of the developing kernel is thin and smooth. This phase will continue until the piston approaches TDC firing and the point at which eddy breakdown commences. If the early flame kernel is acted upon by a flow field with a high strain rate, then its growth is inhibited and hence the phasing of the in-cylinder peak pressure will be retarded, which leads to cyclic combustion instability.

Empirical studies suggest there is a linear relationship between the ensemble-averaged pre-combustion turbulence intensity, \(u_t'EA\), and the average turbulent burning velocity, \(S_T\). In such systems, the flame speed ratio is defined as

\[
\frac{S_T}{S_L} = a + b \left( \frac{u_t'EA}{S_L} \right)^{1/c}
\]

where \(S_L\) is the laminar flame speed for a flame that propagates into a quiescent, pre-mixed mixture. This is illustrated by Figure 2.7., which plots the variation of the ratio of turbulent to laminar burning speeds under fired conditions for a range of spark timings against turbulence intensity for motoring engine operation over a range of engine speeds.
In this case, a factor is introduced (firing pressure, \( p_f \)/motoring pressure, \( p_m \)) to compensate for the additional compression effects upon the ensemble-averaged, RMS velocity fluctuation. For quiescent or pure swirl type combustion chambers, the constants \( a, b \) and \( c \) are close to unity. Tumble based systems with pent-roof geometry's do not exhibit such a relationship. Ensemble-averaged turbulence estimates are given in the range of between 0.4 and 0.6 times that of the mean piston speed but do not scale with engine speed.

Another common measure is the laminar flame thickness, \( \delta_L \), governed by the molecular diffusivity of the mixture, \( D_L \) and the laminar propagation velocity through the mixture:

\[
\delta_L = \frac{D_L}{S_L}
\]

Larger values of \( \delta_L \) increase the overall flame surface area available for reaction. Typical estimates of \( \delta_L \) are of the order of a tenth of a millimetre. Borghi (1985) introduced the theoretical Borghi diagram, shown in Figure 2.8., which illustrates the effects of the ratio of turbulence integral length scales and laminar flame thickness to turbulence intensity and laminar burning speeds in premixed, turbulent flames; that is the influence of turbulence scales on the flame structure (thickness and propagation). If the laminar flame thickness is significantly less than the Kolmogorov length scale, then the flame front is likely to undergo local stretching with increased wrinkling of the flame front. At higher turbulent Reynolds numbers, the point at which this occurs is associated with greater turbulence intensity, reduced laminar flame speed, reduced flame thickness and increased integral length scale. However, Taylor (1993), remarks that these regimes are yet to be established experimentally.

The second combustion phase coincides with break-up of the main vortices through the mechanism of cascading turbulent vortex stretching as described in Section 2.5. For tumble systems, the turbulence intensity term, generated from vortex breakdown, is affected by the chamber geometry, the initial tumble ratio and by the rate of vortex compression. The laminar turbulent transition phase leads to an increasingly wrinkled, fast burning (pre-mixed) turbulent flame. It is the turbulence intensity and turbulent length scales (Integral and Kolmogorov), \( I_t \) and \( I_z \), just ahead of the flame front at this point that crucially dictate the flame shape, propagation speed and combustion performance. The initially thin and moderately wrinkled reaction sheet soon develops into a thick turbulent 'brush' through the direct action of the turbulent flowfield. The simple interconnecting sheets between burnt and unburnt gases become highly convoluted, multiply interconnected and increasingly finely wrinkled with increasing turbulence intensity and decreasing integral length scale (with increasing engine speed). Overall turbulent flame 'brush' thicknesses have been reported of approximately 1 cm from front to back in a conventional SI combustion chamber (Heywood 1988) and are an order of 100 times greater than the laminar flame thickness, \( \delta_L \), observed in pre-mixed, quiescent chambers. Within the convolutions of the turbulent burning flame, local regions are thought to proceed at the laminar burning speed.
Experimental LDA measurements gathered from before ignition until after flame arrival in a disc shaped chamber, close to the cylinder axis at 300 rpm and a mean piston speed of 0.83 ms\(^{-1}\), show that the ensemble-averaged mean velocity, in-cycle rms velocity fluctuation and turbulence intensity are observed to increase linearly throughout the initial laminar burning phase up to TDC (Figure 2.9). However, Hadded and Denbratt (1991) and Gosman (1995) both reported insensitivity of turbulent length scales to turbulent burning flame speeds in tumble-based, pent-roof, SI engines. In certain cases, the turbulent flowfield ahead of the flame front is inhomogeneous and there exists cycle-to-cycle variations in the mean flow. It is however of use to assess the effect of a homogeneous and isotropic turbulence field on a flame propagating with a laminar flame speed (i.e. without the influence of a mean flow).

From the above spatial and temporal correlation analysis, a characteristic eddy turnover time is defined from the integral time scale; that is those eddies that carry the largest proportion of available energy. The turbulent eddy turnover time, \( \tau_T \) is

\[ \tau_T = \frac{l_T}{u'} \quad 2.57 \]

For a laminar flame propagating into a pre-mixed, quiescent mixture with laminar burning speed, a characteristic chemical reaction time, \( \tau_L \) is given as:

\[ \tau_L = \frac{\delta_l}{S_L} \quad 2.58 \]

The ratio of both terms is defined as the Damköhler Number, \( D_a \) often quoted as the inverse measure of the influence of turbulent flow on the chemical processes occurring in the flame Heywood (1988)

\[ D_a = \frac{\tau_T}{\tau_L} = \left( \frac{l_T}{\delta_l} \right) \left( \frac{S_L}{u'} \right) \quad 2.59 \]

Both bulk motion and turbulence play an important role in achieving the desired combustion phases. These are often related to the delay angle, defined as the number of crank angle degrees from ignition to 10% mass fraction burnt and the burn angle, defined as the number of crank angle degrees from 10% to 90% mass fraction burnt. The delay angle includes both laminar kernel and laminar turbulent transition phases. The burn angle incorporates the rapid burning phase of the fully-developed turbulent burning flame as it propagates throughout the complete chamber volume. Propagation is halted by the chamber boundaries and combustion continues briefly consuming all the unburned fractions remaining behind the convoluted flame front. Control of both of these phases will dictate the rate of flame propagation and hence the duration of the delay angle. Shorter delay angles produce better combustion stability with lean or dilute mixtures. Longer burn angles give large reductions in part load NO\(_x\) and uBHC emissions but to the detriment of fuel consumption. At full load, octane requirement and noise are effected by burn rates. It is therefore of paramount importance to improve the understanding and accuracy of predictions of mean and fluctuating velocity components in IC engine cylinders.
In lean burn combustion systems, the weak mixture often penalises the combustion process through a reduction in the speed of flame propagation. In addition, this is more problematic at higher engine speeds. This can be seen in Figure 2.10.; both the delay and burn angles are dependant upon engine speed. The results here are presented for a stoichiometric mixture and spark ignition timed at 30 CA BTDC. A solution is to advance ignition to increase the time available for combustion. However, this method often promotes initial flame development in less homogeneous mixtures thereby enhancing the likelihood of combustion instabilities. This lack of combustion reproducibility from cycle to cycle in lean-burn combustion systems is most effected by inhomogeneity close to the spark plug gap, in regions where the flow field experiences cyclic variability. In addition, it is also provoked by differences in the total amounts of fuel, air and end gas in the cylinder within each cycle.

Experimental LIF studies of air/fuel ratio at the spark plug gap compared to pressure-related heat release calculations were carried out by Neij et al., (1996). They observed that for stratified charge operation, cyclic variations in the early flame development were directly related to variations in the AFR in the vicinity of the spark plug gap and that the burn rate scaled with increasing AFR. They concluded that the mean fuel concentration was an adequate identifier of combustion instability under these conditions and that small-scale inhomogeneities had little influence over the process.

The four-valve cylinder head, tumble airflow motion study of Hadded and Denbratt (1991), using LDA, revealed a linear relationship between measured turbulence intensity at the spark plug gap at ignition with early burn delay angle (ignition to 1% burnt and ignition to 10% burnt). Importantly, they observed that this relationship was maintained for dilution through EGR but not for lean mixtures with excess air. Moreover, an ensemble analysis of the RMS of the mean flow cycle-to-cycle variations showed no clear correlation with combustion variability and a cycle-resolved analysis was proposed.

More recent studies of concepts for G-DI combustion systems (part and full load) that have involved local AFR sampling at the spark plug gap by a mechanical valve (Sadler et al., (1997) and quantitative LIF of the in-cylinder AFR (deSercey et al., (2001) and Gold et al., (2001)) have highlighted the sensitivity of burn angles to air flow and fuelling mechanisms. Stokes (1997) presented Figure 2.11. for a top-entry, reverse tumble, close-spacing, intake-side injector system. As can be seen from the figure, there is a rapid departure from stable combustion (COV, coefficient of variation in the IMEP) between the two modes of operation; from near stoichiometric to ultra lean burn. This illustrates the complexity of ascertaining a meaningful measure of combustion criteria, based upon current knowledge, suitable for a G-DI concept.
2.10 Conclusions of Chapter 2

In-cylinder turbulence is periodic and non-stationary and is independent of its temporal and spatial location in the engine cycle. The turbulence is random and turbulence quantities are described by distributions of a stochastic function; that is one whose variables are distributed according to the Gaussian or Normal distribution. Within IC in-cylinder engine flows, there exists a broad range of large and micro scale gas motions. These turbulent flows are characterised by their temporal resolution (turbulence intensity) and by their spatial resolution (integral length scales).

Across any given engine cycle there is a variation in the mean flow properties. In addition, there is a variation from one cycle to the next. As such, the experimental estimation of the flow properties is performed over finite crank angle intervals, either during a single cycle, where the data rate is high, or more commonly, ensemble-averaged over many engine cycles. The detail of the flow scales then requires partitioning of the flow into a mean, fluctuating turbulent and cycle-to-cycle contribution. This often requires many assumptions to be made about the flow. Several approaches have been adopted in the literature; ensemble-averaged data filtering, autocorrelation function estimation and conditional sampling. Each technique however relies upon an arbitrary choice of ‘cut-off’ frequency in the turbulent spectra.

An alternative approach is to utilise the Taylor hypothesis where flow conditions permit. In a G-DI engine, the conservation of the bulk flow mean velocities, with low RMS fluctuation or cyclic contribution, is required for stable charge stratification. The flow could be considered quasi-steady as the piston decelerates towards TDC. As such, analysis of each air motion may lend itself to the indirect estimation of the integral length scale from the integral time scale using single point LDA.

Four-valve, intake generated, tumble flows are complex and 3-dimensional. High mean gas velocities, steep velocity gradients and the highest levels of turbulence intensity govern the intake period of the stroke. These structures start to decay close to BDC unless a predominant tumble or swirl flow has been established. Tumble centre precession and tumble axis ‘tipping’ are observed as a contribution to an increase in measured turbulence levels. As of yet, there is no presented experimental data through the intake and compression strokes for the high tumble, top-entry, bowled-piston, combustion chambers commonly employed in G-DI engines. Preconceptions about the optimum flow characteristics for ignition in stratified charge combustion systems, based upon existing, pre-mixed, homogeneous combustion models require reconsideration for G-DI.

Both the bulk and turbulent air motions play an important role in achieving the desired combustion phases. In the turbulent, in-cylinder air motion, the eddy turnover time governs the rate at which the chemical reaction can take place through action upon the wrinkled, propagating flame front. The average turbulent burning velocity shows a linear relationship with ensemble-averaged, pre-combustion, turbulence intensity. The laminar thickness of the flame, during the early combustion phase, is defined by the ratio of the molecular diffusivity to the laminar propagation velocity through the mixture. The mean bulk flow motions act upon the flame, drawing it across the combustion chamber. As such, the mean and turbulence characteristics in the spark plug region are of the greatest importance to the first stages of combustion and the early flame growth. The importance of their roles is further compounded in the promotion of combustion in G-DI engines, where the spatial stability of a lean, stratified mixture is required.
Figure 2.0. Infinitesimal Fluid Control Volume

Figure 2.1a and b. The Influence of Cycle-to-Cycle Variations upon Ensemble-Averaged and Cycle-Resolved, Mean and Instantaneous Velocity Measurements for a Typical In-Cylinder Analysis (Heywood (1988))
Figure 2.2. Expected Model of Power Spectrum Profile
(Moriyoshi et al. (1993a,b))

Figure 2.3. Derived Power Spectrum of Instantaneous Measured Velocity
(Moriyoshi et al. (1993a,b))
Figure 2.4. Profile of the Spatial Velocity Autocorrelation Function defining the Integral ($l_i$) and Micro ($l_{\mu}$) Length Scales (Heywood (1988))

Figure 2.5. Summary of the Requirements of a Gasoline Fuel Spray For a G-DI Engine
Figure 2.6. Description of the Requirements of a Gasoline Fuel Spray for a G-DI Engine

Figure 2.7. Variation of Burning Speed with Turbulence Intensity for a Range of Engine Speeds and Spark Timings (modified from Heywood (1988))
Figure 2.8. The Borghi Diagram for Turbulent Pre-mixed Combustion with $\delta_L S_L = 10 \nu$ (modified from Taylor, (1993))

Figure 2.9. LDA Measurements from Start of Ignition to After Flame Arrival in a Disc Shaped Combustion Chamber at 300 rpm (Heywood (1988))
Figure 2.10. Effect of Engine Speed on Delay and Burn Angles for a Stoichiometric Mixture and Spark Timing Set at 30 CA BTDC (Heywood (1988))

Figure 2.11. The Effect of End of Injection Timing Ranges upon the CoV of IMEP for a G-Di Engine at 1500rev/min and 1.5bar BMEP, Top-entry and Reverse Tumble (Stokes (1997))
<table>
<thead>
<tr>
<th>No.</th>
<th>Research Group</th>
<th>Combustion System</th>
<th>Engine Specification</th>
<th>Piston</th>
<th>Engine Speed</th>
<th>Air Motion</th>
<th>Operation</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Yasue et al. (1989)</td>
<td>non-turbulent gas jet</td>
<td>4 valves, single fuel injection</td>
<td>Hot top</td>
<td>rotary</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
</tr>
<tr>
<td>2</td>
<td>Stirling and Bursell (1989)</td>
<td>swirl-chamber gas jet</td>
<td>4 valves, dual fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Fiala and Frisch (1989)</td>
<td>impinging opposed jet</td>
<td>Single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Ansari et al. (1989)</td>
<td>conventional gas jet</td>
<td>2 valves, opposing jet</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Takada et al. (1989)</td>
<td>modified swirl gas jet</td>
<td>2 valves, opposing jet</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Wasitniak et al. (1989)</td>
<td>modified swirl gas jet</td>
<td>2 valves, opposing jet</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Supple and Peth (1987)</td>
<td>conventional gas jet</td>
<td>2 valves, opposing jet</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Heng and Cheu (1987)</td>
<td>modified swirl gas jet</td>
<td>2 valves, opposing jet</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Ansari et al. (1989)</td>
<td>non-turbulent gas jet</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Rasul et al. (1989)</td>
<td>non-turbulent gas jet</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Caiombe and Turbino (1987)</td>
<td>carbureted 2-stroke</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Gostinbier et al. (1989)</td>
<td>G-24 gas jet</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>Breeze et al. (1986)</td>
<td>carbureted 2-stroke</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Breeze et al. (1989)</td>
<td>carbureted 2-stroke</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>McFarlin et al. (1987)</td>
<td>carbureted 2-stroke</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>Heaton et al. (1987)</td>
<td>carbureted 2-stroke</td>
<td>4 valves, single fuel injection</td>
<td>Low rpm</td>
<td>measured</td>
<td>measured</td>
<td>measured at 150°C TDC, engine 3900 rpm. Turbine efficiency 65% at 9000 rpm.</td>
<td></td>
</tr>
</tbody>
</table>

Table 2.0a. Experimental LDA Investigations of In-Cylinder Flows from 1988: Part 1
Table 2.0b. Experimental LDA Investigations of In-Cylinder Flows from 1988: Part 2
<table>
<thead>
<tr>
<th>Technique</th>
<th>2-probe 1-point, 2-point</th>
<th>2-probe volume, 2-point</th>
<th>Elongated volume</th>
<th>Scanning LDA</th>
<th>1-probe 2-point</th>
<th>Scanning LDA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>Megatech MK3</td>
<td>Ruggerini RP170</td>
<td>Waukesha CFR</td>
<td>Ricardo Hydra</td>
<td>2V Swirl gasoline</td>
<td>Ricardo Hydra</td>
</tr>
<tr>
<td>Bore (mm)</td>
<td>41</td>
<td>100</td>
<td>82.6</td>
<td>85.7</td>
<td>80</td>
<td>85.7</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>51</td>
<td>95</td>
<td>114.3</td>
<td>66</td>
<td>80</td>
<td>86</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>4</td>
<td>21</td>
<td>5.7</td>
<td>9</td>
<td>5</td>
<td>9</td>
</tr>
<tr>
<td>Motored Engine Speed (rpm)</td>
<td>500</td>
<td>600</td>
<td>600</td>
<td>1200</td>
<td>1000</td>
<td>1200</td>
</tr>
<tr>
<td>Combustion Chamber Shape</td>
<td>pancake</td>
<td>toroidal</td>
<td>pancake</td>
<td>pancake</td>
<td>pancake</td>
<td>pancake</td>
</tr>
<tr>
<td>Crank Angle BTDC (deg)</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Measurement Region</td>
<td>Mid-plane, 13 mm from head</td>
<td>SPG</td>
<td>15 mm from mid</td>
<td>4 mm from head</td>
<td>7 mm from mid</td>
<td></td>
</tr>
<tr>
<td>Longitudinal Integral Length Scale (mm)</td>
<td>6</td>
<td>6.9</td>
<td>1.3 (fc=50 Hz)</td>
<td>2.0 (fc=25 Hz)</td>
<td>3.9</td>
<td></td>
</tr>
<tr>
<td>Lateral Integral Length Scale (mm)</td>
<td>3.9</td>
<td>2.2</td>
<td>3.1</td>
<td>3.4</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2.1. Studies of Experimental Determination of Direct Integral Length Scales in Motored IC Engines using LDA: Definition of Length Scale Directions
<table>
<thead>
<tr>
<th>Mathematical Term</th>
<th>Abbreviation</th>
<th>Formula</th>
<th>Example Choice of Application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Generalised Mean Diameter</td>
<td>$D[p,q]$</td>
<td>$\left{ \frac{\sum_{i=1}^{n} d_i^p}{\sum_{i=1}^{n} d_i^q} \right}^{\frac{1}{p-q}}$</td>
<td></td>
</tr>
<tr>
<td>Arithmetic Number Length Mean Diameter</td>
<td>$D[1,0]$</td>
<td>$\frac{1}{n} \sum_{i=1}^{n} d_i$</td>
<td>Pollutants, Evaporation</td>
</tr>
<tr>
<td>Number Surface Area Mean Diameter</td>
<td>$D[2,0]$</td>
<td>$\left{ \frac{1}{n} \sum_{i=1}^{n} d_i^2 \right}^{\frac{1}{2}}$</td>
<td>Chemical Catalysis, Absorption</td>
</tr>
<tr>
<td>Number Volume Mean Diameter</td>
<td>$D[3,0]$</td>
<td>$\left{ \frac{1}{n} \sum_{i=1}^{n} d_i^3 \right}^{\frac{1}{3}}$</td>
<td>Chemical Reaction, Momentum Transfer, Hydrology</td>
</tr>
<tr>
<td>Surface Diameter Moment Mean</td>
<td>$D[2,1]$</td>
<td>$\frac{\sum_{i=1}^{n} d_i^2}{\sum_{i=1}^{n} d_i}$</td>
<td>Chemical Catalysis, Absorption</td>
</tr>
<tr>
<td>Volume Diameter Moment Mean</td>
<td>$D[3,1]$</td>
<td>$\left{ \frac{\sum_{i=1}^{n} d_i^3}{\sum_{i=1}^{n} d_i} \right}^{\frac{1}{2}}$</td>
<td>Evaporation</td>
</tr>
<tr>
<td>Surface Area Moment Mean – Sauter Mean Diameter</td>
<td>$D[3,2]$</td>
<td>$\frac{\sum_{i=1}^{n} d_i^3}{\sum_{i=1}^{n} d_i^2}$</td>
<td>Mass Transfer</td>
</tr>
<tr>
<td>Volume or Mass Moment Mean – De Broukere Mean Diameter</td>
<td>$D[4,3]$</td>
<td>$\frac{\sum_{i=1}^{n} d_i^4}{\sum_{i=1}^{n} d_i^3}$</td>
<td>Weight Equivalent Mean for Constant Density, Combustion Equilibrium</td>
</tr>
</tbody>
</table>

Table 2.2. Common Terms Used to Describe Mean Droplet Size Distributions
3.0. LASER and Phase Doppler Anemometry Measurements and the Optical Engine Test-Cell

3.1. Introduction

Non-intrusive LASER diagnostic and measurement techniques are extensively used for the measurement of the properties of multi-phase gas, vapour and liquid fluid flows. The widespread development of these techniques in the field of in-cylinder engine research has helped to drive the technological advances observed in SI and Diesel engine combustion systems. Transparent engines, rapid compression and continuous cycling machines and single-shot injection, high-pressure, combustion 'bomb' type vessels allow these varied techniques to be applied at conditions approaching those of a practical engine. Moreover, these techniques can often be applied without a significant compromise to the geometry of the engine combustion chamber involved. It is now possible to realise the development of complex combustion systems through an increased understanding of the complex interacting parameters, whether considered uniquely in isolation (motored, dynamic or static studies) or as a collective contribution to overall combustion performance (fired studies).

The type of optical method employed depends on the parameter to be measured. A summary of the most commonly reported techniques and their most frequent application in internal combustion engine research are presented in Table 3.0. These methods exploit four properties of light; namely refraction, diffraction, reflection and excitation through irradiance at a suitable wavelength. These can be grouped as techniques that observe Raman scattering, Rayleigh scattering, Mie scattering and Fluorescence theories. More broadly in the literature, these are categorised as elastic and inelastic light scattering techniques, imaging techniques or light extinction and absorption methods. They can be applied at a single point, along a 'line of sight' or over a 'full-field' plane or volume. They are principally applied to the qualitative and quantitative measurement of temperature, velocity, velocity fluctuation, fuel droplet diameter and chemical reaction species concentration in the combustion chamber. The most common of those found in the literature include LASER and Phase Doppler Anemometry (LDA and PDA); Flow Visualisation and Particle Tracking and Image Velocimetry (PTV and PIV); high-speed, LASER illuminated, still, Schlieren and shadow photography, digital video imaging and cinématography, and LASER induced fluorescence and incandescence techniques (LIF and LII).

This Chapter presents a complete description of the theories, experimental techniques and apparatus utilised in the investigations carried out within this thesis. The experimental studies combine the single point techniques of LDA and PDA, as applied to in-cylinder gas motion and gasoline fuel sprays. High-speed backlit shadowgraphy and LASER light sheet illuminated photography of the combustion chamber were also performed to aid visualisation of the fuel injection event.
A limited amount of qualitative and quasi-quantitative planar LIF, under motored engine conditions and high-speed flame photography during fired operation has been performed and this supports the main findings using LDA and PDA. Details of these investigations can be found in Gold et al., (2000) and deSerceu et al., (2002).

The principles of the LDA and PDA techniques are introduced along with the optical Ricardo 'Hydra' engine test-cell and high-pressure fuel spray rigs. Emphasis is placed upon the specific type of LASER-based technique and optical instrumentation used for these studies. Of particular importance is the intrinsic relationship between the type of measurement technique and the theory upon which it is based and its specific application to in-cylinder studies. In such cases, correct interpretation of the results is highly dependent upon a complete understanding of the experimental apparatus and processes involved. Both Sections 3.3. and 3.4. therefore provide descriptions of how best to apply these techniques to meet criteria that ensure good optical access, signal rate and signal to noise ratio, data quality and fidelity in the results. The limitations of the techniques and potential sources of measurement uncertainty are also discussed. The various optical engine builds are described along with the general test-cell operation and engine instrumentation. Specific engine builds, test equipment and procedures are presented in more detail in the relevant experimental chapters. Additional experimental techniques mentioned in the text are included for reference purposes and where data comparisons are drawn in future chapters.
3.2. Introduction to General Experimental Measurement Methods for Gas Flows and Fuel Sprays in Internal Combustion Engines

For many simple flow situations, the Pitot-static tube is an accurate instrument for reliable time-averaged velocity measurements. It is however not readily applicable to flows that exhibit fluctuations in the flow velocity through turbulent interaction, in regions of recirculating flow or where sensitivity to physical intrusion is observed. For these types of flow, several other methods of flow measurement are best employed. These can be used to measure instantaneous velocities at single points or over planes or volumes. Of the most reported, Hot Wire Anemometry (HWA), LASER and Phase Doppler Anemometry (LDA and PDA) and High-speed photography and Particle Image Velocimetry (PIV) are most readily applicable to IC engine investigations.

3.2.1. Hot Wire Anemometry

Semenov (1963) was amongst the first to perform turbulence measurements in IC engines. Semenov used Hot Wire Anemometry (HWA), which relies on a fine, electrically heated wire giving up heat to its surroundings. The change in temperature is associated with a change in electrical resistance or voltage where a constant resistance probe is employed. Compensation is required to correct for pressure and temperature effects. These calibration procedures become less reliable under fired conditions. The technique is physically intrusive and complex probes (e.g. Platts et al., 1996) are required to differentiate between flow components and to resolve flow directional ambiguity. Witze (1980) reports a direct comparison of IC engine measurements performed with both HWA and LDA in a modified Wisconsin L-head engine. The overall conclusion to the report, that included mean velocity, turbulence and gas temperature sensitivity tests, indicated that LDA was far superior for engine applications due to its non-contact and absolute measurement (linear response to fluid velocity) without calibration. It also offers robustness, high temporal and spatial resolution, insensitivity to the thermodynamic properties of the gases and has the ability to follow flow direction where no obvious mean flow pattern exists.

Witze (1996) offers a comprehensive review of in-cylinder experimental techniques (without the requirement of physical cylinder head modifications) and includes detailed references to the earlier HWA works of Barton et al., 1971, Bradshaw (1971), Windsor and Patterson (1973), Dent and Salama (1975) and Lancaster (1976).

3.2.2. Flow Visualisation, Particle Tracking and Image Velocimetry

Particle Tracking and Image Velocimetry (PTV and PIV) are two methods that permit the measurement of flow velocities within a two-dimensional plane of a given flowfield. Three-dimensional flowfield re-construction can be obtained from a large set of parallel planes taken from consecutive engine cycles (e.g. Faure et al., 1996) or from one cycle using a rapid, scanning light sheet (e.g. Brücker, 1997).

A LASER light sheet is used to illuminate particles added to the fluid that are neutrally buoyant. Light scattered from these particles can be captured over multiple exposures or using a pulsed LASER sheet source. As the time between exposures or pulses is known, the particle velocity can be estimated by means of autocorrelation or cross-correlation methods.
To achieve this, the flowfield plane is divided up into many interrogation zones such that, optimally, only one seeding particle is present within each grid area. The correlation is then performed between adjacent, overlapped interrogation zones in the same image (autocorrelation) or between zones in two consecutively recorded images (cross-correlation). A velocity vector is then estimated where a correlation peak is observed. Modern PIV systems use high-resolution CCD cameras and a double-cavity pulsed LASER. It is the repetition rate and required resolution of the camera that controls the final image geometric dimensions, data file size and the maximum velocity that can be measured. Generally, it is only with specialist equipment that it is possible to capture flow velocities representative of in-cylinder airflows and fuel spray droplets. Reuss et al., (1989) were amongst the first to report in-cylinder air velocity vector maps over a significant area of the combustion chamber and clearance volume of a motored gasoline engine. However, due to the available PIV measurement resolution, the engine speed was limited to 600 rpm. Many researchers have sought to employ the principle of dynamic similarity to match Reynolds numbers between working fluids. This is an effective means by which the mean flow velocities could be scaled to match the instruments measurement range. Commonly, air is replaced by water as the working fluid and the piston speed and valve lift profiles are adjusted such that Reynolds numbers are matched within the intake valve jet region. In all cases, an implicit assumption is made that true turbulence scales when flows are dynamically similar but that flow instabilities do not. Heikal et al., (1995) describe a dynamic flow water visualisation rig for the PIV study of in-cylinder flows during the intake stroke. A Perspex cylinder liner and piston are enclosed in a water-filled box to minimise optical distortion through the curved surfaces of the liner walls.

More recently, the use of refractive index matching fluids has permitted the characterisation of flows through optically distorting surfaces. Flow field vector plots can be achieved within the intake ports and complex piston bowl geometries, with the use of carefully selected polymer resin cylinder heads, pistons and valves. These fluids have allowed access to captive volumes in unmodified intake ports and piston bowls previously unattainable with cast iron or aluminium materials. The LASER sheet and camera can be orientated to produce vector maps in any cross-sectional plane. In this case, the use of water limits the study to the intake stroke only. Computational simulation is then used to estimate compression stroke and TDC conditions based upon the nature of the flow at the end of the intake stroke (Feng (2001)).

Typically PIV vector plots are averaged over many cycles to derive mean values and a fluctuation about that mean. The fluctuation represents the spatial variation of the flow velocity and not the temporal fluctuation intensity as measured by the LDA or PDA techniques. In an attempt to quantify these differences, Ullum et al., (1997), utilising state-of the-art PIV equipment, studied the perturbed and unperturbed flow behind a wall-mounted fence obstruction.

They showed that for large PIV sample sizes (1,000 to 3,000 PIV data samples for 95% statistical confidence), the mean velocity profiles obtained by both methods were of comparable magnitude when turbulence was relatively low. However, the PIV profiles of turbulent kinetic energy showed a lower peak value than the corresponding, single-point LDA measurements. Freek et al., (1997) observed differences of between 5% and 28% in turbulence intensity values between LDA and PIV measurements in a dynamic flow rig. Both of the above studies attributed the error to the difference in area between the PIV interrogation zone and the LDA probe volume size.
This introduces bias effects due to the finite bandwidth available for measurement. In PIV, the side of the interrogation zone must correspond to the diameter of the smallest turbulence structures to be measured. This is governed by the diameter of the pixels in the CCD array that make up the length of the side. If the spatial resolution is not adapted to the flow structures then velocity gradient broadening is observed. In this case, the velocity of the particle does not remain constant as it passes through the interrogation region. Out-of-plane velocity vectors can further complicate the effect. In the experimental study of Grosjean et al., (1997), PIV and LDA measurements are combined such that each is used to resolve a different scale of fluctuation in an unsteady swirling flow generated by a rotating cylinder. The study concludes that such a complementary technique is specific to swirling flows that exhibit orderly translational motion and not at all applicable to other types of flow fields.

A comprehensive review of the development of PIV in fluid flow research is presented by Grant (1997).

**3.2.3. High-speed Photography and Cinématography**

High-speed still photography, shadowgraphy, Schlieren photography and cinematography have been used for many decades in engine research studies where good optical access to the combustion chamber was available. These techniques have become increasingly important in resolving fundamental issues and have principally found favour in Diesel combustion systems (e.g. Rao et al., (1992a,b), Winterbone et al., (1994), Ishikawa and Niimura (1996) and Dec, (1997)). Gasoline research has focused upon flame propagation and turbulence interaction, flamefront structures and flame extinction (e.g. Whitelaw and Xu (1996)). These studies have been comprehensively reviewed by Hill and Zhang (1994) and more recently have been employed in the study of direct, in-cylinder and port fuel injection.

These methods offer full field visualisation of the chamber and it is now increasingly common to include studies where light is captured from emission by fluorescence and incandescence. Much use is made of pistons that incorporate quartz windows in the base of the piston bowl (Bowditch pistons). However, with the advent of more intricate piston geometry's, it has become increasingly complex to insert a window that provides a complete combustion chamber view without significant optical distortion. In such cases, researchers have sought to place windows within the cylinder head itself or between the cylinder liner and the gas face (e.g. Reuss et al., (1989), Salters et al., (1996) and Whitelaw and Xu (1996)). In certain studies, the complete cylinder liner has been replaced with a quartz (fused silica) or sapphire equivalent. These studies are limited to motored or short, skip-fired, operation. The accumulation of combustion products on the windows causes rapid deterioration in the image quality. As a result, precautionary correction methods are usually required (e.g. Winterbone et al., (1994)).

Fired studies rely on the light emitted from the combustion event to illuminate the objective lens. Through careful choice of film and exposure settings and with suitable lens magnification, it is possible to capture the finer details of propagating flame fronts or combusting Diesel fuel sprays. The rate of change of density in the evaporating fuel spray or within the hot combusting gases can be captured using the Schlieren technique, although this requires two parallel, flat windows positioned on either side of the combustion chamber.
To record events prior to combustion, it is necessary to provide a pulsed point or sheet source of light. Commonly, an Nd-YAG, Ruby or Copper-Vapour LASER is utilised because of the high-energy output, short pulse duration (which can effectively 'freeze' high-speed motion) and in some cases, for the high repetition rates available. LASER light sheets down to approximately 0.2 mm can be achieved with suitable sheet forming optics. It is the thickness of the light sheet that defines the depth of the image. For 'thick' sheets, the final image consists of an integration of all events within the sheet boundaries as viewed from the direction of the camera. The method is applicable to media that are considered 'optically thin', such that LASER light attenuation is of the order of 10% or less after passing through the medium.

In optically dense media, such as high-pressure fuel sprays, interpretation of fine structures becomes less reliable. In these cases light is scattered from multiple sources simultaneously (obscuration) and shadows are projected due to the irregular energy absorption throughout the cross-section. Some improvements in image quality can be gained through the use of co-planar light sheets (improved sheet energy homogeneity), post-processing (based upon image intensity profiles) or by the use of fluids that do not absorb light at critical wavelengths.

Seeding methods can be utilised to aid the light scattering from fuel sprays. Kamimoto et al., (1996) introduced silicone oil (1% by mass) dissolved within the base fuel. During evaporation of the fuel, the fine silicone particles became suspended in the fuel vapour phase. A copper-vapour LASER was used to illuminate a cross-section of the flow. Photographic images of Mie scattered light were captured perpendicular to the plane of the light sheet. More advanced methods involve the use of a fluorescent tracer or dopant, of matched properties, added to the charge gas or liquid fuel. A suitable LASER source tuned to the correct excitation wavelength induces fluorescence in the dopant. The fluorescence signal is not only dependent upon fuel (dopant) concentration, but also upon the temperature, pressure and oxygen concentration (quenching).

Filtering the collected light before the camera ensures that only the fluorescence signal (liquid and vapour) is present in the final image and not that attributed to the Mie scattered signal. Differentiation of the liquid and vapour fuel phases can be achieved by an extension to the LIF technique that utilises an Exciplex fluorescing chemical. In LASER induced exciplex fluorescence (LIEF), the fluorescence occurs at different wavelengths and with different intensity for each phase. Two intensified cameras are required with suitable filters centred on the two fluorescence wavelengths.

With precise selection of the LASER wavelength, (crystal frequency multiplication, Optical Parametric Oscillator (OPO) or pumped dye LASER), it is possible to capture fluorescence of the hydroxide (OH⁻) radical, polyaromatic hydrocarbons (PAH) and oxides of nitrogen (NOx) sites within the combustion chamber. If LASER energy typically greater than 100 mJ is possible over a short pulse duration (less than 10 nS), incandescence of irradiated soot can be observed which can be related to soot concentration and soot particle size.

Charge coupled device (CCD) cameras can be utilised to produce high resolution, digital images. Gated image intensifiers are required to capture light at low levels of intensity (e.g. during the study of lean combustion flames). However, due to the finite time required to refresh the CCD arrays (and the low repetition rate of an Nd-YAG LASER), these cameras have a limited operating frequency, even at reduced frame sizes or using frame straddling techniques.
This precludes continuous acquisition of transient events occurring in the same cycle and most studies are conducted over consecutive cycles, where the image acquisition trigger is phase shifted with respect to the engine TDC marker. The most advanced of the CCD cameras are capable of very high repetition rates but are limited by the number of frames that can be stored. As such high-speed ciné film cameras, drum cameras and rotating prism film cameras are typically utilised for fast, single cycle transient photography (Rao et al., (1992a,b)).

3.3. LASER Doppler Anemometry

The LASER Doppler Anemometer is an optical instrument that can be applied to the measurement of the velocity of gas and liquid flows. It utilises LASER light beams to measure a given velocity component at a single point in the flow as a sequence of near instantaneous samples. It was first reported for such measurements in the studies of Yeh and Cummins (1964). Since its establishment in the 1960's as a flow measurement technique, an extensive literature has developed that addresses the fundamental issues of particle seeding and light scattering, optical configurations, velocity bias, signal analysis, results processing, and instrument limitation and measurement errors. A comprehensive and expansive study is presented in the literature by Watrasiewicz and Rudd (1976), Drain (1980), and Durst et al; (1981), from which more detailed material can be gathered.

The LASER Doppler anemometer has been used for measurement in many diverse situations ranging from subsonic to supersonic high-speed flows; in free streams, in boundary layers, in liquid films (e.g. Wittig et al., (1996)), in flames and across shock waves (e.g. Delery (1993)). The LDA technique is non-intrusive to the flow and has a high spatial and temporal resolution. It requires no calibration (the output signal is a linear function that is proportional to the velocity) and with suitable frequency shifting techniques, directional ambiguity in the flow pattern can be removed. It is particularly suited to the measurement of turbulent flows.

This section aims to briefly introduce the LDA and PDA techniques (PDA as an extension of LDA), the theories of light scattering (with regards reflection and refraction) and light detection from solid and transparent particles. For the purpose of the measurements presented within this thesis, this description is limited to the specific application of LDA and PDA to IC engine studies, vis-à-vis gas flows and fuel sprays. The theory, experimental methodology, optical configurations, data processing and sources of potential error as well as practical instrument limitations are described. Both the heterodyne (optical beating) model and interference fringe model are used to explain the LDA technique, as these are the most generally described in the literature. For further reference, Durst et al., (1981) includes other models based upon light path length differences and the standing wave pattern approaches.

3.3.1. The LDA Principle and the Doppler Shift of Light

LDA is based upon the determination of the Doppler shift of LASER light scattered (reflected and/or refracted) from moving particles in a fluid flow either present naturally or seeded artificially. The Doppler shift effect is defined as the observed change or apparent change in the frequency of wave motion owing to the relative motion of the source and/or receiver.
This can be extended to the case where the frequency shift is produced by the movement of an intermediate object through which wave motion is transferred from a source to a receiver. The seeding particles (scattering centres) can be considered as moving receivers and transmitters of the LASER light.

The methods of LDA flow measurement are as diverse as the types of flow that are studied and hence this brief summary is concerned with the method most frequently applied to in-cylinder flow measurements. This is commonly termed the Dual beam (two LASER beams of equal intensity) or Differential Doppler technique. The physical principle of the differential Doppler technique is illustrated in Figure 3.0. A single LASER beam passes through a suitable beam splitter to produce two coherent beams of equal intensity that are crossed at the focal point of a common achromatic lens to form a small volume of high light intensity. The ellipsoidal intersection region of the two beams is the measurement or probe volume of the LDA. The dimensions of the probe volume of a commercial LDA system, as viewed through the vertical plane X-Z in Figure 3.1, with Gaussian beam diameter, $d_p$, a beam expansion factor, $E$ and a transmitting lens of focal length, $F$ are given by:

\[
\begin{align*}
\text{Length} & : \delta_z = \frac{4F\lambda}{\pi Ed_z \sin\left(\frac{\theta}{2}\right)} \\
\text{Height} & : \delta_s = \frac{4F\lambda}{\pi Ed_z \cos\left(\frac{\theta}{2}\right)} \\
\text{Width} & : \delta_\ell = \frac{4F\lambda}{\pi Ed_z s}
\end{align*}
\]

All LDA and PDA systems can be classified according to the number of velocity components that they measure simultaneously. A multiple component system uses the three highest spectral power output lines in the visible region of a suitable LASER source to measure three velocity components simultaneously. Helium Neon (HeNe) and Argon-ion (Ar+) Continuous Wave (CW) gas LASERS are most commonly used because of their long coherence lengths, stability to mode drift and relatively low cost. An Argon-ion LASER with broadband mirror assembly will produce green light at 514.5 nm, blue at 488 nm and violet at 476.5nm. Each colour is then used to measure a specific velocity component. For briefness, this description will however be limited to the treatment of a single component system operating at 514.5 nm and in the fundamental transverse LASER mode (TEM00).

The fundamental processes observed in measurements of instantaneous velocity using a LDA are most often described by the use of two models. The following sections briefly summarise the Heterodyne and Interference Fringe models.

(a) The Heterodyne Model

A single particle carried by the flow, moving through a single LASER beam scatters light from its surface in all directions with an unequal intensity distribution. The particle can be considered as a moving wave source. A collection lens focuses the scattered light onto a single photodetector located at a fixed position from the measurement volume.
The frequency of the scattered light is Doppler shifted relative to that of the incident beam. The particle produces a Doppler shift frequency that is equal to the dot product of the particle velocity and the difference in the scattered and incident wave vectors. The shifted frequency is however too high to be suitable for measurements. Therefore, in a Dual beam LDA arrangement, a second LASER beam is superimposed upon the first at a different angle of incidence. The light scattered from this beam has a different Doppler shift. When light scattered by the particle from both beams reaches the photodetector an interference (or optical 'beat') signal is produced (heterodyning). The Doppler beat frequency is equal to the difference in Doppler shifted frequencies and is low enough to be resolved with suitable electronics.

As the direction of the scattered wave vectors causing both Doppler shifts focused onto a single photo detector is the same, the resultant Doppler frequency created by these two Doppler shifts is equal to the dot product of the velocity and the difference in the incident wave vectors. It is therefore no longer a function of the collection angle. LDA measurements can be made in any direction perpendicular to the optical axis by rotating the laser beams. However, the position of the receiver will have a considerable effect upon the acquired signal strength.

The derivation of the Doppler frequency for a Dual Beam anemometer is shown in Figure 3.2. Each light beam of wavelength, λ is represented by a unit vector $\mathbf{n}_{1,2}$ in the direction of incidence. The angle of beam intersection is given by $\theta$. A particle traverses the probe volume with velocity vector, $\mathbf{v}$. The projection of vector $\mathbf{v}$ on the vector in the direction $(\mathbf{n}_1 - \mathbf{n}_2)$ is given by the dot product (in the direction of $(\mathbf{n}_1 - \mathbf{n}_2)$ with signed magnitude $|\mathbf{v}| \cos \alpha$). The Doppler frequency, $f_d$, is then given by
\[
f_d = \frac{m}{\lambda} \left[ \mathbf{v} \cdot (\mathbf{n}_1 - \mathbf{n}_2) \right]
\]
where $m$, is the refractive index of the medium. From the vector diagram
\[
|\mathbf{n}_1 - \mathbf{n}_2| = 2 \cos(90 - \theta/2) = 2 \sin(\theta/2)
\]

The measured component of $\mathbf{v}$, perpendicular to the fringe pattern (in the plane of the beams and perpendicular to the optical axis) is given by
\[
U = |\mathbf{v}| \cos \alpha
\]
where $\alpha$ is the angle between $\mathbf{v}$ and $U$.
With air ($m \equiv 1.0$), rearranging and substitution in Equation 3.3
\[
f_d = \frac{2U \sin(\theta/2)}{\lambda}
\]

(b) The Fringe Model
Rudd (1969) proposed an 'interference fringe' model as a means by which the LDA technique and signals could be more readily visualised.
Two monochromatic (of singular frequency and wavelength), coherent (adjacent and successive waves are in phase), linearly polarised LASER beams of equal beam paths are brought to a common focus that forms the measurement volume. An ellipsoidal, 3-dimensional measurement volume is produced that comprises of equally spaced, parallel, constructive and destructive interference fringes (light and dark stripes) formed by the coherent, incident, plane wave fronts. A bright fringe is generated where two peaks come together in phase. When a peak and trough are aligned the effect is cancelled and a dark fringe is observed.

The fringe pattern is formed perpendicular to the plane of the incident beams and runs parallel to the angle bisecting the intersection. An idealised fringe pattern is given in Figures 3.0. and 3.1. To ensure a high contrast, plane fringe pattern, the non-linear nature of the LASER beams must be considered. A LASER beam has a Gaussian radial intensity profile that must be considered when analysing its propagation. The accepted convention of the definition of the radial distribution of light intensity is given in Figure 3.3. The Gaussian beam radius, $r_0$ is defined at the $1/e^2$ intensity contour where the light amplitude is $1/e$ times the centre amplitude and the intensity is $1/e^2$ times the centre intensity. The Gaussian distribution of the light intensity at the intersection volume is equivalent in all three directions and defines the practical dimensions of the probe volume. The effective dimensions of the probe volume are then those given in Figures 3.1 and 3.3.

In the direction of propagation of the beam, the radius, $r_D$ varies between a maximum and minimum value. The position of the minimum beam radius, $r_0$ is termed the beam waist and the wavefronts are considered to be planar. At any other point in the beam either side of the waist, the wavefronts exhibit a variation in radius of curvature and a change in sign. The variation in beam radius along the length of the beam follows a hyperbolic profile. The divergence of the LASER beam in Figure 3.3. is defined as the angle, $\beta$ between the axis of the beam and the asymptotes of the hyperbola such that:

$$\beta = \frac{2\lambda}{m r_0}$$  \hspace{1cm} (3.7)

To ensure optimal quality (intensity, parallelism) of the fringes, the beam waist radius of the focused beam, $r_0$, and the beam waist radius of the unfocused beam, $r_0$ (leaving the laser) must observe the relation of Kogelnik and Li, (1966), and Dickson, (1970), such that

$$r_0 = \frac{f\lambda}{m r_s}$$  \hspace{1cm} (3.8)

The fringe characteristics are determined by the wavelength of the laser light, $\lambda$, and the intersection angle, $\theta$. From Figure 3.0., the fringe spacing, $s$, can be derived using trigonometry, such that,

$$s = \frac{\lambda}{2 \sin(\frac{\theta}{2})}$$  \hspace{1cm} (3.9)
The number of fringes, $N_f$, in the probe volume generated by two beams of Gaussian diameter, $d_g$, with beam expansion factor, $E$ and a transmitting lens of focal length, $F$ is given by

$$N_f = \frac{8F \tan(\frac{\theta}{2})}{\pi d_g E}$$  \hspace{1cm} 3.10

A particle carried by the flow enters the probe volume and passes through the fringe pattern, with a velocity component normal to the fringes, given by $U$. As it does so, the particle receives light that fluctuates with the passing of a dark or bright band.

The scattered light reflected from the particle thus fluctuates at a frequency that is proportional to the rate at which the particle traverses the fringes. For known fringe spacing, the frequency is then directly proportional to the velocity of the particle normal to the fringe direction. For fringe spacing, $s$, the frequency, $f_v$ is given once again by,

$$f_v = \frac{2U s}{\lambda}$$  \hspace{1cm} 3.11

A collection lens is used to focus the scattered light onto a photo multiplier tube (PMT), which converts the fluctuating light intensity signal into a voltage output. In all cases, the fidelity of the results depends upon correct instrument alignment by such a method as that proposed by Miles and Witze (1996) to assess measurement precision by measuring the fringe field uniformity. The distortion of the fringe pattern (variation in fringe spacing) by particles disturbing one beam near to the probe volume was studied by Ruck et al., (1993). They concluded that for realistic air flow applications with appropriate seeding, period length variations in the LDA signal would contribute to less than 0.5 % error in turbulence measurements.

3.3.2. General Principles of Light Scattering in LDA
LDA and PDA are concerned with the reflection, diffraction, absorption and refraction of plane light waves that illuminate a solid or transparent tracer particle within a fluid flow. Durst, et al., (1981) have shown that the fringe model can only partially describe these phenomena and that it is only possible to calculate the correct modulation of an LDA signal by electromagnetic scattering theory (Mie) or experimental methods. It is therefore important to understand the relationship between the incident light properties (wavelength, intensity, polarisation and phase), seeding particles, and flow direction. From Watrasiewicz and Rudd (1976), the magnitude and directional distribution of light intensity scattered from a particle depends upon:

- The total incident intensity illuminating the particle. It is assumed that the particle is spherical and that the reflective scattering is an average taken over the entire particle surface.
- The scattering angle. The intensity of scattered light is strongly dependent upon the particles direction relative to the incident beams.
- The normalised particle diameter, $d_p$. Generally the sizes of the particles are often comparable to the wavelength of the incident light.
- The ratio, $n$ of the refractive indices of the scatterer and the surrounding medium. An approximate refractive index criteria is that $(n-1)\alpha>0$.
- The polarisation orientation of the incident light.
In 1908, Mie derived equations to describe light scattering from particles from the solution of Maxwell's equations of electromagnetic fields. These theories have become collectively grouped under the Generalised Lorenz-Mie Scattering Theory (GLMT) for the description of light scattering from homogeneous, dielectric, spherical particles. The derivation of these theories is beyond the scope of these studies and as such the explanation given henceforth summarises only those results important to the LDA and PDA techniques. A complete history of the developments in GLMT to date is given in the literature by Gouesbet and Gréhan (2000).

Mie formulated a description of light scattering (reflection, refraction, diffraction and absorption) and stated that the intensity of the scattered light, which reached an observer, was a function of the intensity of the incident light and the Mie scattering function. For an unpolarised incident light wave, a particle will scatter light that consists of light polarised in the plane of observation (parallel polarisation) and that polarised perpendicular to it. The plane of observation is the plane formed by the direction of propagation of the incident wave and the direction of observation. The intensity of light in both scattered polarisation planes varies with the parameters listed above. A simplistic schematic of the principle of Mie scattering theory is illustrated for the case of an incident LASER beam (monochromatic and linearly polarised) in Figure 3.4. A particle at point P scatters light as it passes through the beam along the x-axis. An observer in space, for example at point A, receives light at an angle, \( \vartheta \), in the z-PA plane, and at an angle, \( \Phi \), from the xz-plane of polarisation.

The intensity of scattered light at point A is a function of the incident light intensity, \( I \), and the Scattering function, \( S \). The scattering function is dependent upon \( \vartheta \), \( \Phi \), the relative refractive index of the scatterer to medium, \( n \), and the Mie or size parameter, \( q \), where \( q \) is given by

\[
q = \frac{\pi d_p}{\lambda} \tag{3.12}
\]

Following Mie's scattering theory, the scattered light intensity received at a point, is given by the expression,

\[
I_{(x,y,z)} = I_s(\vartheta, \Phi, q, n) \tag{3.13}
\]

Figure 3.5. shows the polar diagrams of the calculated Mie scattered intensity of light from a particle relative to the direction of the incident beam direction as indicated by the bold arrow. The radius indicates light intensity plotted on a logarithmic scale. Mie scattering theory predicts that the scattered light intensity has a maximum in the forward scatter direction. The complex irregularities in the intensity envelope on the angular distribution diagram are due to diffraction effects. The effect of increasing particle size is illustrated in Figure 3.5a to 3.5c. As particle size, \( d_p \) relative to the incident light wavelength, \( \lambda \) is increased, the intensity distribution becomes more complex with many more maxima and minima. In addition, the LDA signal modulation is reduced and determination of the Doppler frequencies becomes increasingly difficult. Figure 3.5abc shows that most of the scattered light is in the direction of propagation of the incident waves. In addition, the intensity of scattered light due to different scattering modes varies with the scattering (receiving) angle.

Figure 3.6. illustrates the different modes of light scattering from a transparent particle illuminated with an unpolarised light wave.
The polar diagram of intensity shows light scattering in the parallel and perpendicular planes of polarisation by reflection and two orders of refraction derived from geometrical optics theory and the Lorenz-Mie theory.

Where possible LDA systems are designed to best utilise the forward scattered light intensity onto a large receiving lens and photodetector physically opposed to the LASER source and transmission optics (forward scatter systems). The incident light is focused to a small point to ensure high power density. In flow situations where physical restrictions do not allow such an optical path, use is made of light scattered laterally (side-scattering systems) or backwards (backscatter systems). Backscatter systems are commonly utilised in IC engine research where it is only possible to provide one optical access point into the combustion chamber. The scattered light is collected along the same path as the beam transmission, or with a small angular offset, (off-axis backscatter systems). The scattered light illuminates the transmitting lens and passes to a second lens that focuses it down to a point (or pin-holed aperture) in front of the photodetector. As can be seen from Figure 3.5abc, the intensity in the side and backscatter directions is significantly reduced compared with that in the forward direction and that the collection angle for maximising light intensity varies with particle size. The backscatter signal can be improved by the use of a larger area lens positioned closer to the measurement volume. It should be noted that two particles of the same size but of different refractive index would produce two different intensity distributions.

### 3.3.3. Particle Seeding of Gas Flows

Particle seeding is the most important factor in LDA measurements. To measure fluid velocity, small seeding particles are introduced into the flow. LDA does not measure the velocity of a fluid, but the behaviour of tracer particles within the flow naturally or artificially introduced. If the particles are sufficiently small, the slip velocity between particles and the fluid is negligible and hence a measurement of true fluid velocity can be obtained. The particles must follow the flow frequency sufficiently closely but be sufficiently large as to scatter sufficient light to give an acceptable SNR from the photodetector. An additional constraint on particle size is termed the 'modulation depth' of the LDA signal. Large particles will tend to produce a large low pedestal frequency component but with a small amplitude Doppler signal that can be difficult to resolve.

The choice of seeding particle is often a compromise based upon the operating environment (e.g. combustion studies) and the nature of the flow to be studied (the expected velocity fluctuations); the type of LDA technique employed; the optimal angle of scattering (scattered light intensity, particle size and signal strength) and the optical set-up (spacing of the fringes). Typically, seeding particles are used of the order of 1-2 μm in diameter in airflow but this may vary with the type of fluid flow investigated and the fringe spacing selected. It is good practice where possible to choose a fringe spacing that is at least four times the mean particle diameter.

Table 3.1. shows some of the most commonly utilised seeding particle materials, range of diameters and their application in IC engine type research. A good signal level and quality are best achieved with only a single particle present in the measuring volume at any one instant.
Several particles crossing the measurement volume may produce multiple scattering effects inside the probe volume that can negate true signals or lead to indeterminable signals (different phases between particles) that are difficult to interpret using current signal processing techniques (Doppler ambiguity). The correct concentration of seeding is particular to the type of experiment performed and preparatory tests are required to find a suitable seeding level. Generally, the introduction of seeding particles to a fluid flow should represent only a very small fraction of the total volume such that interaction between particles is minimised. As the particle concentration increases, the apparent fluid viscosity is changed and a change in pressure drop is observed (Boothroyd (1966)). Crowe (2000) published a summary of turbulence modulation experimental data found in the literature. The percentage change in turbulence intensity was plotted against the ratio of particle diameter to turbulence length scale for all studies. The dilute concentrations of small particles observed in LDA measurements and their effect upon the mean flow and turbulence, through augmentation or suppression, could be considered, at most, to attenuate the turbulence intensity by approximately 5% at TDC conditions.

In-cylinder LDA measurements performed in a motored or fired engine depend upon the intake stroke to fill the chamber with gas and seeding particles. Once the intake valves are closed, the mass of seeded mixture remains constant. During the compression stroke, the density of the air increases rapidly and hence, the volume fraction occupied by the incompressible seeding particles can increase by a factor of ten from intake to TDC. In sparsely seeded regions of the cycle, the spatial distribution of the particles becomes less uniform and particle averaging bias effects are introduced (Section 3.3.6.1c). In practice, a compromise in seeding density is required to ensure that all four strokes of the engine cycle can produce validated measurements with acceptable data rates; that turbulence is not artificially altered and that velocity bias, within a given ensemble-averaged crank angle window, is minimised. The number of data points per averaging window is not constant through each of the strokes and therefore sufficient sample numbers are required to ensure statistical confidence. This implies that a compromise must be made between the seeding density and the number of engine cycles over which averaging will take place.

### 3.3.4. Directional Ambiguity

An LDA probe volume that comprises of an equally spaced fringe pattern will produce a Doppler signal that is independent of the direction of the traversing particle. Additionally, a stationary particle will not produce a signal. Therefore, to eliminate directional ambiguity, one beam is frequency shifted relative to the other. This has the effect of adding a component of constant velocity to the fringes that depends upon the direction of the frequency shift. A stationary particle then produces a signal equal to the imposed frequency shift. A moving particle will produce a signal of lower frequency (moving with the fringes) or higher frequency (moving against the fringes).

Beam splitting and frequency shifting in LDA and PDA systems is most commonly achieved by the use of a rotating diffraction grating (RDG) or acousto-optical modulator (Bragg cell). Both devices rely upon diffraction of the LASER beam at a given frequency. The properties and governing equations of the RDG utilised in the Ricardo Consulting Engineers' LDA system (Figure 3.7a,b) are given in Figure 3.8 for reference.
The diffraction grating is etched upon a rotating glass disk. A LASER beam that passes through the grating is diffracted over many orders with varying intensity. The grating line pairs move through the beam with a constant rotational speed and thus impart a frequency shift to the diffracted LASER beam.

In the Dantec LDA system, a DISA Bragg cell was utilised to provide a 40 MHz frequency shift after beam splitting. Within the Bragg cell, the beam passes through a transparent medium that is excited by a propagating acoustical wave. The sound waves induce periodic density variations that diffract the LASER beam in many directions with differing intensities. The tilt angle of the Bragg cell and the acoustic signal intensity is then adjusted such that first order diffraction is selected for the output. The intensity is maximised and the frequency of the output beam is shifted by an amount equal to the sound wave oscillations. Generally, Bragg cells are preferred to diffraction gratings for LDA frequency shifting as they permit greater frequency shifts, are easier to control and have better stability and transmit more light.

3.3.5 Doppler Signal Processing

The electronic signal processors required to determine the frequency and phase shifts due to the Doppler effect are central to the role of an LDA or PDA system. The processor must optimise the signal rate and the quality of signal; both of which are complex functions of the flow field to be measured, the seeding particle size and density and the optical set-up. The transmission lens focal length and the beam separation determine the fringe spacing and the frequency bandwidth of the signal processor determines the measurable velocity range. LDA systems can be operated in a continuous mode (high seeding level requirement) or burst mode (alternatively termed individual realisation mode).

The output signal from the photodetector is made up of randomly arriving, frequency modulated signal bursts mixed with periods of inactivity. Figure 3.9. shows the composition of a typical Doppler signal. The shape of the burst signal is dependent upon the Gaussian distribution of light intensity across the diameter of a LASER beam operating in a single longitudinal mode and the fundamental transverse mode (TEM00). The definition is reproduced in Figure 3.3. Within the probe volume, the intensity of the light distribution across the fringe pattern varies as a result, such that fringes at the edges of the volume are weakly illuminated when compared to those at the centre. The resultant scattered light signal thus varies in amplitude from an arbitrary value greater than zero. As such, the Doppler signal is a composite of two parts; a low frequency Doppler Pedestal signal and a high frequency Doppler burst signal.

Generally, modern Doppler processors use signal level threshold criteria and low and high-pass filtering in ‘burst detection circuits’ to validate a signal and remove the Doppler Pedestal. The remaining frequency signal is then utilised to determine the velocity.

There are four methods that have been applied to estimate the frequency of the Doppler signal. Their application is generally dependent upon the type of flow to be measured or to situations where multiple particles are present within the probe volume or high levels of noise (optical and electronic) are to be encountered. The experimental LDA and PDA work detailed within this thesis was carried out using a combination of Doppler processors. These are listed in Table 3.2. for completeness.
These types of processor can be summarised simplistically, as follows:

1. **The Frequency Tracker:** Frequency tracking processors utilise a frequency tracking filter to follow variations in a given signal as long as the phase-locked-loop remains closed. A phase comparator monitors the Doppler signal and the output of a voltage-controlled oscillator. Frequency variations in the Doppler signal register as a change in phase and the output frequency of the oscillator is adjusted to match. As such, the frequency of the oscillator is a measure of the frequency of the Doppler signal. However, in poorly seeded flow situations, discontinuous or sporadic Doppler bursts mean that the tracker can lose its oscillator 'lock' and its ability to follow the signal frequency modulation.

2. **The Period Signal Counter:** Time domain processors generally use an analysis routine that 'counts' a number of 'zero crossings' in the Doppler signal generated by the dark fringes in the probe volume fringe model, i.e. to measure the transit time of a specified number of cycles within one Doppler burst. If a single frequency is present at one instant, then the time difference between 'zeros' is the inverse of the frequency of the signal. As a check to ensure accurate measurement and include only single particle crossings, most counters compare the time for a seeding particle to pass five and eight fringes. If this value is within a given accuracy factor, then the Doppler event is accepted.

3. **Covariance Processors:** In these types of processors, a part of the Doppler signal is delayed by a known time constant, $t$. This corresponds to the addition of a phase shift of $t$ times the frequency of the signal. The shifted signal is then compared with the original remaining signal in a phase detector of the correlation type. The output of the cross-correlator is proportional to the cosine of the phase difference. The correlation estimates this difference in phase between the two signals. This phase difference is then divided by the added delay to give the signal frequency. The phase detector can determine particle size with Doppler signals from two different detectors as the input.

4. **Fourier Transform or Burst Spectrum Analysers:** A Fourier transform is used to convert the Doppler time signal into the frequency domain. This is achieved by the direct implementation of a Fast Fourier Transform (FFT) in hardware. The transform produces a spectral output of form equivalent to the Doppler frequency contribution. The frequency is generally estimated from a parabolic curve fit to the measured spectrum.

### 3.3.6. Measurement Uncertainties in LDA Measurements

There are two possible sources of error in LDA measurements performed within the cylinder of an IC engine. These can be categorised as systematic errors, or those that pertain to the instrument set-up and operation; i.e. those that are introduced by the experiment and its environment. Secondly, statistical errors are introduced through the estimation of flow parameters due to a finite number of samples of the velocity signal. The estimated statistical measurement uncertainties in particular situations are presented with the results in subsequent chapters.
3.3.6.1. Systematic Errors

(a) Optical System Set-up

This relates to the physical errors introduced during the set-up of the optics that can affect the beam angles and position of the measurement volume. An error in the beam angles will affect the fringe characteristics and therefore the measured mean and RMS velocities. An error in the measurement volume location will produce uncertainties in the velocity estimates at a given point in the flow. Both types of error can be minimised by following good experimental practice. The uncertainty in the velocity measurement due to the optical set-up will be assessed in the relevant chapters using the method of Boutrif and Thelliez (1993). In their particular LDA set-up, the error in the LASER wavelength was estimated at 0.2% and the error in beam separation, at 4%. Under these conditions, Boutrif and Thelliez (1993) stated that velocity measurement errors attributable to optical set-up were always less than 5%.

(b) Signal Noise

Signal noise in LDA signals is the primary reason for rejection of data. The methods of burst detection using threshold criteria employed by Doppler signal processors are sensitive to the signal to noise ratio. Noise is present electronically (photodetection shot noise, preamplifier circuits etc.) and as optical noise (higher order LASER modes and light scattering). In particular, optical noise is highest in systems with mirrors and windows (fouled and scratched), where high-density seeding or fuel sprays are present (multiple particles and light scattering from outside the probe volume - 'phase noise') and where the LASER beam is reflected off many surfaces. To reduce the optical noise and maximise the signal to noise ratio, it is necessary to use anti-reflective (AR) coatings and to match the LASER power with seeding density and optical set-up. A broadband 'Herbar' coating made by Melles Griot can reduce reflection losses from 4% to 0.3 % over a wide range of wavelengths (unpolarised light). Monochromatic objective lenses and photomultiplier lens are coated with a single layer of Magnesium fluoride of ¼ wavelength thickness that reduces reflection losses to less than 2%.

The saturation of the LDA signal is principally a result of optical veiling glare which plays an increased role close to surfaces in reducing the signal to noise ratio (SNR). Subsequent measurements are effected by the recovery time of the photomultiplier tube (PMT) employed. The effects can be particular to the kind of LDA system utilised and the type of optical access and/or engine build.

In practice, it is experimentally difficult to remove all sources of flare; this effect will always happen at measurement points close to surfaces, or certain parts of the cycle where the piston passes through the measurement volume. This gives rise to very high light levels at the PMT that can potentially damage the cathode. The effects can however be minimised by employing sensible veiling glare rejection precautions.

There will always be a 'trade-off' between the data acquisition rate and an acceptable SNR and therefore lengthy runs will be inevitable to ensure confidence in the collected data. A combination of the following methods were employed for these studies:

1. A rotating mechanical shutter and shutter driver and timer unit was synchronised with the engine cycle TDC 'non-firing' marker pulse for a given engine speed.
The shutter is driven from a phase locked motor which is controlled from a crankshaft driven oscillator with a variable phase delay. This could be tuned to attenuate the LASER source for a given duration and protected the PMT from overloading when solid surfaces passed through the measurement volume (e.g. piston and valves).

2. For back scatter LDA systems it is important to maximise the angle between the plane of the transmitted beams and the collection optics (off-axis backscatter). However, with restricted optical access, the required probe volume size can be a limiting factor.

3. Light scattering within the combustion chamber was minimised by coating the engine surfaces with a matt, black, anti-reflective coating or paint (cobalt oxide). Where possible, both the piston and cylinder head were coated.

4. LASER power was kept to a working minimum of between 100 and 200 mW at the probe volume.

5. Good quality optical glass with suitable AR coatings was utilised to minimise veiling glare in the direction of the collection optics. This reduces flare from beam walk off (multiple reflections between the two surfaces of a lens or window).

6. Flare rejection was achieved by the use of achromatic lenses well corrected for spherical aberration.

7. Although unavailable for this study, it is recommended that pre-amplified PMTs are used that have a high frequency response, a quicker recovery time from overload protection and wider dynamic range.

(c) Particle Averaging Bias (Velocity Bias)

In LDA and PDA measurements, the probe volume is most commonly fixed at a point of interest in the flow region. A sequence of velocity or diameter measurements is generated due to the passage of consecutive particles. The spatial distribution of the particles is considered uniform. In laminar flows the particle arrival times are generally Poisson distributed. When turbulence is introduced the particle arrival times become distorted. At high flow velocities, the probability per unit time of a particle entering the measurement volume is greater than in regions of low velocity (e.g. flow reversal or stagnation). In such flows, an average of the velocity values (and the estimated velocity variance) is biased towards the higher flow velocities.

The effect is most pronounced in sparsely seeded, turbulent gas flows that exhibit regions of rapid flow reversal. In such cases, measured values can be weighted by the inter-particle arrival time, particle residence time or distributed within equal, consecutive time bins from which the first sample is selected. The residence time weighted mean velocity, \( \bar{u}_R \), is given by

\[
\bar{u}_R = \frac{\sum u_i \Delta t_i}{\sum \Delta t_i}
\]

where \( u_i \) is the individual velocity measurement and \( \Delta t_i \) is its residence time in the measuring volume.

(d) Rotating Machinery and Periodicity

As detailed in the above sections, the application of both the LDA and PDA techniques to in-cylinder measurements is additionally complicated by the periodic and unsteady nature of the fluid flows involved.
Much of the experimental expertise was derived from reviews of the literature where investigators had studied developing or fully-developed turbulent flows in wind tunnels and channels (e.g. Romano et al., 1993), Carrotte et al., 1993). In-cylinder flows are inherently non-stationary, turbulent flows that exhibit high mean velocity fluctuations across the engine cycle and the potential for variations from cycle-to-cycle. The engine speed is not constant during the cycle and a crank angle encoder is necessary to provide a time reference. Single point LDA offers high temporal resolution but the sample times are correlated with the flow velocity. The arrival rate of particles is not independent of the flow velocity (velocity bias) and is additionally governed by the temporal location within a given stroke or engine cycle.

To eliminate high sampling velocity bias in turbulence intensity estimates and to accommodate sharp velocity reversals, a high frequency shift is required. Hilton (1991) developed algorithms to correct for sampling velocity bias effects on statistics computed from single position, engine LDA data. Glover et al., 1988a designed a scanning LDA system using a camshaft driven rotating mirror that enabled the probe volume to be traversed across measurement region. The temporal resolution of the system was compromised but the sample times were independent of the flow velocity.

(e) Velocity Gradient Broadening

An LDA signal is generated by a particle as it traverses the measurement volume normal to the direction of the fringe pattern. In a two-dimensional cross-section, the particle can follow a parallel trajectory through any part of the fringe pattern defined by the probe volume diameter and length.

A fluid flow that has a velocity gradient normal to the fringe pattern will produce different velocity values for particles at different positions in the velocity gradient as they follow differing trajectories through the probe volume. Such an effect produces an apparent velocity fluctuation with time and thus an increase in the measured variance, $V_s'$ of the velocity signal. This is given by

$$V_s' = \left( \sigma_p \frac{\partial U}{\partial y} \right)^2$$

where $\frac{\partial U}{\partial y}$ is the mean velocity gradient at the point of measurement and $\sigma_p$ is the standard deviation of the distribution of the particles in the $y$-direction across the probe volume. In practical LDA systems, $\sigma_p$ is approximately limited to $\frac{1}{4}$ of the probe volume diameter by the collection optics. The extra variance must be then be subtracted from the measured value.

(f) Finite Transit Time Broadening

The increase in measured velocity fluctuation due to transit time errors occurs as a result of processing of a Doppler burst. The frequency content of the burst is best estimated where there are a large number of fringe crossings. If the number of fringes, $N$ in the probe volume is small, then the potential error in estimating the frequency content of the signal is greater. As such, any subsequent particles, with constant velocity, $U$ will produce inconsistent results.

The increase in the measured variance, $V_i'$ of the velocity is given by

$$V_i' = \left( 2 \sqrt{2} \frac{U}{N} \right)^2$$

where $U/N$ is the mean velocity of the particles.
Again, the true variance is calculated by subtracting the increase in measured variance due to finite transit time broadening.

**(g) Fringe bias**

The probability of a particle generating a measurable signal depends upon the direction of the particle relative to the LASER beams. A particle moving parallel to the fringes would not be measured. The fringe bias effect is governed by the signal processor (the number of cycles required to resolve the frequency) and the optical set-up (the number of measurable fringes on a path through the centre of the volume, perpendicular to the fringes). The ratio of these numbers will dictate the 'acceptance angle' for the velocity vector. The fringe bias effect can be effectively minimised by the use of suitable frequency shifting at frequencies of several times the Doppler frequency.
3.4. Phase Doppler Anemometry

3.4.1. The PDA Optical Principle
Phase Doppler Anemometry is a single particle, light scattering technique, extending from the LDA technique, which can provide a simultaneous measurement of a solid, liquid or gaseous particle diameter as well as its instantaneous velocity. The PDA has the ability to provide local spatial and temporal droplet characteristics in a spray at large distances from the nozzle and within environments inaccessible to the more widely used particle sizing instruments, based upon LASER light diffraction (Fraunhofer diffraction pattern or Low Angle Light Scattering, LALLS) methods. A comprehensive review of the current status of spray diagnostic methods, including PDA and LALLS, can be found in the literature of Bachalo (2000).

This brief description of PDA will be limited to a conventionally termed PDA system as schematically represented in Figure 3.10. Details of planar or Dual PDA systems can be found in the literature (e.g. Dantec, 2000). The transmission system and probe volume construction is as described in Section 3.3.1. for a differential mode LDA with beam intersection angle, θ. In the PDA set-up, the scattered light is almost always collected in a side or forward scattered direction by the receiving optics. The scattering angle, φ, made between the receiving optics and the transmission axis in the horizontal plane is chosen such that only one mode of light scattering (reflection or refraction) is predominant in the direction of the receiving optics. Particle velocities are calculated from the frequency of the Doppler burst signal, by the same method as that described for the LDA technique. Droplet sizing is achieved by a second photodetector, which is positioned at a fixed distance from the first, governed by the elevation angle, ψ, in a conventional PDA and located within the same receiving optics. The plane of polarisation (perpendicular or parallel to the scattering plane) of the incident LASER beams determines the light collected.

Light scattered onto each detector experiences a phase shift due to the difference in optical path lengths. The difference in phase between these two signals can be related to the particle diameter using a ray tracing geometric optics approach. It is assumed that the particle is spherical (free of surface distortions and with a diameter generally greater than 0.5 μm) and homogeneous (any inclusions must be significantly smaller than the LASER wavelength). The particle and continuous medium refractive indices must be known. A linear relationship between the phase shift and the droplet diameter is then observed when the receiving angle is chosen for one dominant mode of light scattering. If several modes of scattering are present with comparable light intensity at a given receiving angle, then non-linearity in the relationship is observed.

3.4.2. Reflection and First Order Refraction and the Phase-Diameter Relationship in PDA Measurements
The principle scattering modes of reflection and refraction can be described in terms of light rays and through the use of geometrical optics. A more rigorous approach is to use electromagnetic theory in which field amplitudes scattered from each beam, are summed as they interfere at each of the detectors. An incident LASER beam is initially both reflected and refracted at the surface of a transparent particle.
The ratio of light intensity reflected to that refracted (Fresnel coefficients) is dependent upon the angle of incidence, plane of polarisation and the ratio of the particle to the continuous medium refractive indices. The angle of the refracted ray is given by Snell's law:

$$\cos \alpha = n \cos \alpha'$$  \hspace{1cm} (3.17)

where $\alpha$ and $\alpha'$ are the angles between the surface tangent and the incident and refracted ray. The angle, $\alpha$, of the incident ray will determine any subsequent internal reflections and refraction at the particle-medium interfaces. These are generally denoted by an integer, $p$. In Figure 3.11, $p=0$ is the first surface reflection, $p=1$ is the first refracted ray and $p=2$ is the ray leaving the droplet after one internal reflection. The light intensity decreases with each $p^{th}$ order and PDA systems generally utilise only reflection and first and second orders of refraction. The angle between the incident ray and the $p^{th}$ order ray exiting the sphere is given by

$$\nu = 2(p\alpha' - \alpha)$$  \hspace{1cm} (3.18)

As discussed in Section 3.2.2, the scattered light intensity varies for differing scattering angles ($\nu$-dependency); with the incident intensity; with the size parameter, $q$; with refractive index, $n$ and with incident polarisation as illustrated in Figures 3.5. and 3.6.

Bachalo et al., (1984) have reported that the forward scatter amplitude functions that describe light rays reflected, refracted and diffracted by a transparent sphere using geometrical optics theory show good agreement with Mie scattering calculations where droplet diameters were greater than the light wavelength, such that ($q>10$). This is schematically illustrated in Figure 3.6. This equates to a droplet diameter, $d_p$ of greater than 1.6 $\mu$m for a conventional PDA system. In summarising previous studies, Bachalo et al., (1984) note that for $q>15$, and for scattering angles greater than 10°, diffraction is significantly lowered and can be avoided by suitable placement of a receiving optic. Under these conditions and assuming that the scattered light is taken as an average over the angular fluctuations due to rays $p=0$ and $p=1$ interfering at the receiver, the scattered intensity can be considered proportional to the square of the diameter.

However, in practical, polydisperse spray environments the amount of light attenuation by other droplets cannot be determined. In these cases the analyses are extended to describe the interference fringe pattern formed by the reflected and refracted scattered light rays and the difference in optical path lengths (differences in phase) observed at the receiver. A spherical droplet passing through two coherent, linearly polarised beams of equal intensity, which intersect at a small angle (<10°), scatters light from each beam as if the other beam was not present.

When observed from a common point in space, both scattered beams have an approximately equal scattering angle and amplitude function. An interference fringe pattern is formed with spatial frequency determined by the relative phase difference of the interfering light waves scattered from both incident beams. The relative phase difference can be shown to be linearly related to the drop size using geometrical laws. It should be noted, however, that multicomponent scattering interference (additional interference between light scattered by a combination of refracted and reflected rays) will occur at certain scattering angles. Therefore, to ensure a linear phase-diameter relationship, the PDA receiving optics must be placed at a suitable collection angle. Figure 3.12. illustrates an important result for collection angles.
It shows the percentage of light reflected from a water droplet surface in air, for the s-state (perpendicular to the plane of incidence) and the p-state (parallel to the plane of incidence) polarisations, at different angles of incidence. The results are plotted for reflectivity at the interface between both low to high and high to low refractive index transitions. For low to high refractive index, the combined reflected intensity begins to rise significantly above angles of incidence greater than approximately 40°. For high to low, the same percentage reflectivity is present at angles as low as 10°. For the low to high case, at an angle of approximately 53°, the p-state reflectivity becomes zero whilst the s-component is approximately 12 %. This is termed the Brewster's angle for water in air. At Brewster's angle, p-state light is wholly refracted through the medium and is linearly polarised. For the high to low index interface, Brewster's angle is reduced to approximately 37° and the percentage reflectivity of the s-component is of the order of 20%. For PDA systems, the incident LASER light is linearly polarised, and Brewster's angle is determined from the ratio of refractive indices, to ensure that p-state polarisation reflectivity is close to zero and only first order refracted and not reflected light is collected at the receiver.

In most PDA applications it is therefore preferable where possible to select the first order refraction (p=1) at Brewster's angle to maximise light intensity in the receiving direction. Two or more photodetectors are utilised to measure the spatial frequency of the scattered fringe pattern. Each photodetector has a fixed location relative to the others in the collection optics. A droplet passing through the probe volume produces a Doppler burst signal at each detector but with a different phase. The shift in phase is resolved by measuring the time between all zero crossings in each of the signals and then dividing that by the period of the total burst signal. For a given optical set-up, the measured differences in phase at the two detectors increases linearly with increasing droplet size as illustrated in Figure 3.13a., cases 1 to 3. As the particle size increases, the difference in phase shifts, \( \Phi \) can exceed 2\( \pi \) cycles and the resolved particle diameter becomes undistinguishable with one determined from a much smaller phase shift.

Commercial PDA systems utilise a third photo-detector to increase the droplet diameter measurement sensitivity range by resolving the 2\( \pi \) ambiguity as shown in Figure 3.13b. In such systems, a phase shift measurement is made between detectors 1 and 2, and between detectors 1 and 3. Detector 3 is located close to detector 1 such that the largest droplet to be measured by the instrument does not exceed a phase shift difference of 2\( \pi \). The sensitivity and diameter resolution of the instrument between detectors 1 and 3 can be low (compared to that between 1 and 2) as it is used for particle size validation and not measurement.

In certain PDA configurations, the centroid locations of the three photo-detectors is governed by an aperture mask plate fitted behind the receiving optics collection lens which allows light to fall selectively upon each detector from the same drop. The measurable droplet size and velocity ranges are determined by this geometry and the frequency bandwidth of the signal processor. The sensitivity of the instrument for differing droplet size ranges can be improved by changing the distance between apertures in the mask, thus changing the gradient of the phase-diameter relationships. After passing through the front lens and aperture mask, the scattered light is focused down through a composite lens onto multi-mode fibre optics onto the photomultiplier tubes. Focus and alignment of the collection optics is achieved by the projection of a slit shaped aperture or spatial filter, mounted in front of the fibre bundles, onto the probe volume.
The aperture is selected to ensure that its projection is positioned upon the intersection point of the two incident beams; i.e. it must match the beam waist at the probe volume. The width of the slit and the diameter of the incident LASER beams define the size of the effective measuring volume. As such, the measuring volume of the PDA is much smaller than the intersection region of the two LASER beams. This removes the effects of light scattered from sources outside the probe volume and ensures the light at the photodetectors comes from the same droplet.

For the commercial Dantec system, the particle diameter, \( d_p \) for reflection, is given by the relationship

\[
d_p = \Phi \left( \frac{\lambda}{2\pi} \right) \sqrt{\frac{1 - \cos \theta \cos \phi \cos \phi}{\sin \theta \sin \phi}}
\]

where, \( \Phi \) is the measured difference in phase shift at any two photodetectors and \( \lambda \) is the wavelength of incident light. For first order refraction, the particle diameter is given by

\[
d_p = -\Phi \left( \frac{\lambda}{2\pi} \right) \sqrt{\frac{1 + \cos \theta \cos \phi \cos \phi}{\sqrt{1 + n_2^2 - n_2^2}}} \sin \theta \sin \phi
\]

where \( n \) is the ratio of refractive indices of the particle and medium. These equations do not contain any calibration constants. It should be noted that the phase difference for refraction has an opposite sign to that for reflection.

### 3.4.3. Measurement Uncertainties in PDA

PDA measurements are subject to the measurement errors common to LDA, as discussed in Section 3.3.6. In addition, the increased complexity of the technique along with the assumptions stated in the previous section, introduce more sources of experimental uncertainty and the potential for low signal to noise ratios. These errors are introduced through the optical set-up, non-linearity in the phase-diameter relationship or by the physical properties of the sprays and droplets within the environment to be measured.

#### 3.4.3.1. Systematic Errors

(a) Optical Set-up, Light Collection and High-Voltage PMT Balance.

In PDA the optical geometry and hence, the measurement accuracy depends upon three angles. These are related to the beam intersection, the elevation of the receiving apertures and setting of the scattering angle. These uncertainties are considered to have a minimal effect upon the measurement precision when compared to non-linearity's within the phase-diameter relationship. Pitcher and Wigley (1992) state that at a given refractive index (1.33) and for a fixed linear relationship, the variance error for a calculated droplet size will be of the order of 5% and 10% for changes in refractive indices of between 1.27 to 1.45 and 1.22 to 1.45 respectively.

(b) Trajectory Ambiguity Effect (TAE)

Section 3.3.1. described the radial Gaussian distribution of light intensity across a LASER beam and its effect upon the LDA or PDA control volume. A particle entering the probe volume scatters light with an intensity that varies with its position through the fringe pattern. Transparent particles, travelling along differing paths (with differing light intensity) through the fringe pattern at the same time, will scatter different amounts of reflected or refracted light.
The phase difference is dependent upon the location of the particle in the probe volume. In particular, the effect becomes more pronounced for large droplets (diameters greater than 50\% of the measurement volume diameter, Dantec, 2000). Particles smaller than the focused beam diameter are assumed to be nearly uniformly illuminated and therefore free of the TAE.

The physical problem of TAE is geometrically illustrated in Figure 3.14, using the notation of Gouesbet and Gréhan (2000). In this case, the collection optic is set-up at a near-forward scattering angle, \( \theta \), of approximately 30\(^\circ\) for a water droplet. The refracted or reflected energy transported to the receiver is the sum of its Fresnel coefficient, \( R \) and \( r \) respectively and the incident intensity in the Gaussian beam. For refraction, the energy transport is greater than for reflection at a given intensity (\( R > r \)). The magnitude of the illuminating intensity varies over the range, \( e \) (at the beam extremity) to \( E \) (near the beam centre). For the upper case, the light at the receiver is dominated by refraction as \( RE > re \). In the lower case, a condition is reached where \( rE \) tends to \( Re \) and no single mode dominates. A PDA system that is set-up to collect first order refracted light scattering will produce erroneous results (or invalidated data) when this condition is predominant. It is good experimental practice to repeat measurements with different probe volume dimensions by varying the Gaussian beam diameter and intensity and beam separation or focal length. A full analysis of the effects of the Gaussian intensity distribution of LASER beams in PDA measurements upon sizing errors is given by Hardalupas and Liu (1997).

(c) The Slit Effect

The slit effect occurs where transparent particles traversing the probe volume intersect with the edges of the slit shaped projection (effective measurement volume) of the spatial filter in the receiving optics. The effect is illustrated simplistically in Figure 3.14. At one edge of the slit, refraction becomes the dominant collected light scattering mode as any reflection occurs outside of the slit region. At the opposing edge, reflection becomes the dominant mode and refracted rays exit the droplet outside of the slit region.

(d) Signal Validation

Signal acceptance criteria in PDA must be selected such that light scattered from the smallest drops encountered in a spray will result in a validated measurement. A droplet diameter population will otherwise be biased towards those larger, validated droplets. To ensure small droplets are detected, high LASER power is required as the scattered light intensity increases as the square of the droplet diameter. In addition to signal amplitude validation (thresholding), there exists an uncertainty factor applied to the measured phase difference between detectors. This introduces a tolerance band to the phase diameter plot where the width is most commonly given as a percent of the total size range. This is casually termed the sphericity factor, or a measure of a particles' geometric fit to a perfect sphere.

(e) Light Collection Vignetting

Vignetting of the receiving optic focal lens occurs where measurements of the spray can only be obtained with relatively small optical access or the optical boundaries of the windows are curved. This is commonly encountered in the studies of sprays to be measured in-situ; sprays injected into high-pressure vessels; combusting sprays or bomb calorimeters or toxic sprays in hazardous environments. The effect is represented schematically in Figure 3.15. The scattered light from the probe volume forms a cone with the front lens of the receiving optics.
If a window is placed between the lens and the spray, the cone can become 'clipped' and the region of illumination upon the front lens is reduced. Significant reduction of the illuminated region will result in the poor scattered light intensity at one or more of the photodetectors. The data collection and validation rate are then dramatically reduced. In such environments, a longer focal length receiving lens can be used as a compromise.

(f) High Concentration (Dense) Sprays
The analysis of droplet measurements in liquid sprays with PDA become increasingly unpredictable in regions where the spray is no longer dilute and can be considered dense. In these types of flows, the transport of the dispersed phase elements (e.g. droplets, bubbles) depends upon the spatial distribution of the elements. Ruff and Faeth (1995) define a dense, dispersed flow as one that has dispersed-phase volume fractions greater than ~10%. In such flows, the proximity and hence, collisions, breakup and coalescence or agglomeration (mass, momentum and energy transfer) between elements is frequent. For a PDA system, this results in greater signal noise produced by the presence of multiple droplets in the probe volume, oblate or prolate spherical drops or scattering centres in the rays adjacent to the beam intersection. The experimental data rates are reduced and biasing of the diameter distribution occurs. In most cases, those droplets that produce a valid measurement represent only a small subset of the total droplet population.

Dense spray regions are most generally observed over a short distance from the injector tip. In the primary breakup region the presence of a liquid core or ligaments results in a very low probability of dispersed discrete spherical droplets being acquired. In addition, it is likely that conditions have exceeded the maximum droplet concentration in the probe volume that will ensure single droplet occupancy. In many studies, near nozzle measurement locations yield few validated samples compared to points further downstream in the fuel jet. Pitcher et al., (1990) carried out PDA measurements in a dense Diesel and ethanol fuel spray inside a two-stroke Jenbach JW 50 engine. Analysis of the results showed that the dense spray region extended up to 100 nozzle diameters downstream of the single-hole Bosch fuel injector of 0.27 mm diameter. Pitcher et al., (1990) used the absence of data to determine the length of the liquid core. By observing the duration of the data void between the leading and trailing edge, they observed that the core length was directly proportional to the injected fuel load.

Methods to improve measurements in dense sprays have been proposed by Koo and Martin (1991), Pitcher and Wigley (1992) and more recently, by Araneo and Tropea (2000). These have centred upon selection of the best experimental set-up by systematically altering the operating parameters relating to LASER power, photomultiplier high voltage and SNR validation thresholds. In addition, Pitcher and Wigley (1992) and Araneo and Tropea (2000) suggest probe volume dimensions that minimise the likelihood of multiple droplet occupancy and ensure maximum light intensity by decreasing the volume diameter. However, if the ratio of beam diameter to particle diameter is too small, Gaussian trajectory effects (TAE) become important (Hardalupas and Liu (1997)).

Fandrey et al., (2000) propose a combination of signal amplitude and half integer (3.5) phase ratio discrimination between detector pairs (1-2 and 1-3) as a means by which accurate particle sizing in dense sprays can be achieved.
Reflection based signals are rejected due to their relatively weak signal intensity. In this case, it is the intensity signal thresholding limits that define the PDA effective probe volume and not the size of the traversing particle. From the construction of the phase-diameter relationship it can be inferred that refraction based signals are positive, and increase from zero with increasing droplet diameter whereas phase shifts from reflection based signals are negative, starting at zero and decrease with increasing droplet diameter. For each case the phase shifts between detector pair 1-3 increase or decrease \( r \) times faster than that between detector pair 1-2, where \( r \) is the ratio of detector separation in each pair. In a first order refraction system, the refraction-based signals are recorded from 360° on the phase plot for very small particles as the signal processor has been set to look for positive phase shifts. If the phase ratio between detectors is an integer value then there must always exist a refraction based signal pair that corresponds to a reflection based pair. Fandrey et al., (2000) therefore suggest the use of a half integer value to distance refraction signals from reflection-based signals. Experimental measurements of volume flux obtained by applying the discrimination techniques and those from direct measurement (based upon a known volume measure of fluid over a fixed time interval) showed a good comparison for a moderately dense air-blast spray.

Ruff and Faeth (1995) present a comprehensive study of non-intrusive measurement techniques applicable to dense spray regions that can be used to validate PDA measurements.

\((g)\) Changes in Relative Refractive Index

In PDA, accurate measurements of droplet sizes depend upon the linearity of the phase-diameter relationship that is related to the refractive indices of the scatterer and scattering medium. The refractive indexes of both are complex functions of the incident light wavelength and local temperature and pressure. Significant in-cylinder pressures and temperatures will affect the liquid and gas densities and introduce refractive index gradients and change Brewster's angle. At high pressures the incident LASER beams can be defocused or steered from their true intersection point.

Variations in the refractive index of the liquid droplet will result in the greatest change in signal phase and intensity received at the collection optic. Estimates of the fuel temperature during injection under engine operation are usually based upon an average between injector tip temperature and the fuel critical temperature or boiling point. This assumption is not valid where multi-component fuels are considered. As such, Pitcher et al., (1990a,b) quantified the sensitivity of PDA measurements over a maximum range of refractive indices by using both light and heavy fuels at critical and room temperatures respectively. The effect on the phase-diameter relationship was then calculated from available data on fuel density variation with temperature and the Eykman relationship:

\[
\frac{(n_l^2 - 1)}{(n_l + 0.4)} = const \times \rho
\]

where \( n_l \) is the refractive index of the liquid and \( \rho \) is the density. Pitcher et al., (1990a) concluded that for scattering angles of approximately 70° and for \( n_l > 1.27 \), the phase/refractive index relationship remained practically linear for all droplet diameters investigated. However, the 30° scattering angle showed much greater sensitivity to refractive index changes, with a poor phase response due to variations, for droplets of 10 micron diameter or less.
3.5. The Application of the LDA, PDA and Photographic Techniques to an Internal Combustion Engine- Experimental Apparatus

3.5.1. Preamble to the Application of LDA and PDA in Optical Engines

Witze (1996) reports that Hutchinson et al., (1978) were amongst the first to perform time-resolved LDA measurements in a reciprocating, plastic, model engine at 200 rpm. These early in-cylinder studies required extensive modifications to the engine and usually required the insertion of windows either in the piston crown or cylinder head.

The research emphasis was placed upon the application of the technique to engine airflows but shortly afterwards, more complex indirect, non-intrusive flow measurement studies emerged using LDA under more realistic engine conditions (e.g. Liou and Satavicca (1983), Fansler and French (1988, 1992), Glover et al., (1988a, b), Arcoumanis et al., (1990), Corcione and Valentino (1990, 1991), Haddad and Denbratt (1991), Moriyoshi et al., (1993b), Vannobel (1996), Hong and Chen (1996, 1997), Stangimaer et al., (1998) and Himes and Farrell (1999)) or in steady state flow rigs (e.g. Carabateas et al., (1996), Nogi et al., (1998)). These motored studies were followed with fired engine investigations (e.g. Rask (1979), Wigley et al., (1981), Boulrif and Thelliez (1993), Miles et al., (1994) and Whitelaw and Xu (1996)). The validity of motored measurements used to determine the characteristics of fired combustion systems has long led to a division in the literature. The general consensus at present, however considers that results obtained in a motored engine are representative of the flow characteristics up to the point that combustion occurs in a fired engine (Chan and Turner, (2000)).

PDA measurements of Diesel and gasoline fuel sprays have traditionally been carried out within ambient or low-pressure quiescent chambers (e.g. Kume et al., (1996), Pontoppidan et al., (1997), Han et al., (1997), Comer et al., (1998), Preussner et al., (1998), Araneo and Tropea (2000)) or steady state flow port models (e.g. Dementhon and Vannobel (1991), Vannobel et al., (1993)). The high pressure and temperature environments required to approach the thermodynamic conditions representative of a Diesel engine are more difficult to achieve. Typically, these are attained through either a rapid cycling type machine (Pitcher and Wigley (1991), Carabell et al., (1993), Warrick et al., (1996), Dec, (1997)) or a high-pressure, 'bomb' type vessel (e.g. Karimi, (1989), Naber et al., (1996), Renner et al., (1994)). Pressure purge-type 'bombs' are generally supplied from an external high-pressure bottle or receiver charged from a compressor. Heating of the charge is either achieved externally or in-situ with electrical heaters. For hot, combusting studies, these types of vessels are in almost all cases limited to a single shot of fuel, before purging of the chamber contents and the subsequent re-charge is required. In a few cases (e.g. Reitz et al., (1988)), it is possible to introduce a continuous flow of fuel into the chamber with varying amounts of air motion.

Gasoline fuel spray studies have until recently been concerned with closed or open valve, intake stroke, manifold injection and without a real requirement for elevated in-cylinder pressures. As such, experiments were performed in low-pressure chambers or within modified, optically accessed intake ports and in some cases, through glass cylinder liners or pistons during the intake stroke.
With direct injection gasoline combustion systems, fuel injection into the cylinder can occur late in the compression stroke where gas temperatures and pressures influence the mixture preparation processes. Much of the experimental expertise developed within the Diesel spray studies at high pressures is directly applicable to gasoline fuel spray research.

3.5.2. The Optical Hydra Engine Test-cell
The following sections describe the general details of the experimental apparatus used for this study. Variations used for specific experiments are provided in the text of the relevant experimental chapters.

3.5.2.1 The Hydra Engine Test Bed
The installation of a unique experimental test cell within the Department of Mechanical and Manufacturing Engineering of the University of Brighton commenced in December 1995. The aims of the programme were to enable the characteristics of in-cylinder airflows in IC engines to be studied using an optically accessed, single cylinder, Mk1 Ricardo ‘Hydra’ engine motored with a DC dynamometer. The engine, motor and engine ancillaries were commissioned in early 1997. Subsequently, the test cell capabilities have been continually extended. Figure 3.16. shows a plan of the test cell indicating the primary systems outlined below. The test cell is sub-divided into the following sections classed as primary systems:

- Engine, intake and exhaust systems, optical components and associated ancillaries.
- Dynamometer, controller and tachogenerator.
- Oil supply temperature and pressure control and filtration.
- Water supply pump and temperature and pressure control.
- Air supply, temperature and pressure control.
- Seeding particle generator, pressure, quality and quantity control.
- LDA system and elements control.
- LASER system, control and cooling.
- Safety systems.
- Fuel delivery supply, pressure and temperature control.
- Injection system control.
- Ignition system control.
- Synchronisation and engine timing resolver systems.

(a) The Ricardo Hydra Optical Research Engine
The Ricardo 'Hydra' is one of a range of single cylinder, optical research engines (Figure 3.17.). The design of the engine allows various types of optical access to the combustion chamber under motored operation or for periods of limited skip firing. This is achieved by the use of a dry lubricated, extended piston arrangement that ensures that the combustion chamber and optical surfaces are not contaminated by oil carry over.
The extended piston is fastened to an oil lubricated lower piston, in a double piston arrangement, where each piston runs within its own separate cylinder liner with the same stroke. The upper piston runs in a fabricated steel cylinder block extension (extended barrel) that is fitted to a standard Ricardo single cylinder 'Hydra' crankcase that houses a single piece crankshaft, connecting rod and the modified lower piston. The extended barrel houses the upper piston wet cylinder liner that is chemically surface heat treated for LDA applications. The piston rings on the lower piston are replaced by two sets of shaft oil seals that prevent upward contamination of oil mist. In addition, radially distributed holes in the lower piston skirt allow oil to lubricate the seals. Lubrication to the crankshaft, main bearing shells and piston skirt is via splash feed from an external oil pump that draws oil from the sump.

The extended piston within the barrel extension is the effective swept volume of the engine and the lower crankshaft must be replaced in order to vary the stroke. The bore may be varied by exchange of the upper, hardened, Tufftrided cylinder liner. The upper piston is attached to the lower piston directly, and shims and spacers with ring dowels are used to ensure that both pistons are co-linear, rotationally orientated and that the correct cylinder head clearance is set for the required compression ratio. Compression within the combustion chamber is maintained by the use of three sets of Rulon® PTFE piston rings as illustrated in Figure 3.18. The rings are fabricated from a compounded form of TFE fluorocarbon mixed with other inert chemicals. The top two sets of rings provide the compression seal and are made from Rulon® AR and Rulon® J. Both of these are highly resistant to abrasive wear and have a very low coefficient of friction.

Nitrile rubber 'O' rings are used to provide an outward side thrust against the cylinder liner. The lower, larger ring (rider ring) is made from Rulon® impregnated with bronze. This sacrificial ring ensures that the upper piston runs true in the liner and does not rock during piston turn around. A special applicator tool is necessary for mounting of the rings, and a second tool is required for insertion of the extended assembly within the given bore. In-cylinder gauge pressure is recorded by a Kistler type 6121/6123 piezoelectric pressure transducer (sensitivity of 16.2 pC per bar) mounted flush with the combustion chamber surface and a Kistler type 5001 charge amplifier.

The upper barrel also incorporates the optical 'tipping' mirror required for access to the combustion chamber through a quartz window situated in the piston crown. The mirror is mounted across the barrel diameter and between the legs of the extended piston. It is mounted on tapered roller bearings and can be positioned accurately by means of a lockable rotational micrometer as shown in Figure 3.19. The extended barrel is also fitted with an extraction cowl necessary for the removal of oil mist and seeding particles that have escaped passed both sets of rings, which can lead to window fouling.

The Mk1 Ricardo 'Hydra' Research engine is mounted upon a rigid steel channel section sub-frame fixed to a reinforced concrete plinth as shown in Figure 3.20. Adjustable mounts allow the engine to be aligned with a Lawrence Scott LA38 DC dynamometer set upon isolation pads. These pads and seismic mass of the plinth minimise transmission of vibration from both the engine and dynamometer and from the surrounding environment. Motoring speed of the dynamometer is controlled through a KTK DC controller and a digital tachogenerator where the signal is generated by a 100 tooth wheel.
The engine crankshaft flywheel is driven by the dynamometer through a GWB 128-10 flexible coupling and spline drive shaft with a Mönninghoff electromagnetic toothed clutch decoupler.

The three-phase power supply cables for the main test-cell systems are trunked under the floor, whereas the data-only cables are suspended in conduit from the ceiling. The water supply and return is also underfloor with anti-siphon valves. Engine data acquisition and feedback control connections are made via a primary breakout box on the engine plinth and a secondary box mounted next to the control racks as shown in Figure 3.21.

(b) Oil and Water Systems
These comprise the external systems required to supply both oil and water at the correct temperature and pressure to the engine. The components of each system are listed in Table 3.3. Both the water and oil circuit temperatures are controlled by the use of NiCr/NiAl type K thermocouples and platinum/rhodium thermocouple’s (PRT). The thermocouples are positioned in both circuits at locations going into and exiting the engine; at the heater outlet and at two positions considered equidistant between them. The oil and water are both heated by immersion heaters located remotely from the engine. Two PID controllers switch the heaters and set the water temperature to 80 °C and the oil temperature to 60 °C. Fine-tuning of the oil temperature is achieved by the use of an additional, PID controlled, three-port mixing valve and a shell-in-tube, water-oil, heat exchanger. Water temperature fine-tuning is regulated manually with a shell-in-tube, water-water, heat exchanger.

An impeller type pump maintains the water pressure at approximately 2 bar. The system is vented to atmosphere via an open tank positioned at the highest point in the water circuit. A sight class and float switch ensure the correct water level in the tank. A purge valve on the cylinder head and stepped pipe arrangement in the extended barrel ensure that air does not build up within the system. A gear type pump supplies oil at 3 bar to the lubrication gallery in the Hydra crankcase. Oil is drawn from the sump, pressurised, filtered and then heated before being returned to the engine crankcase and cam boxes. An additional oil pump is used to aid the oil scavenge from the cam boxes back to the sump.

(c) Engine Air and Seeding Systems
The air system is required to ensure correct intake and exhaust of air to and from the engine, in conjunction with a seeding rig and damping plenum volume required for the LDA technique. Particle seeding of the inlet charge is achieved with a light grade Magnesium (II) Oxide powder with a mean diameter of 1-2 μm and density of 3580 kgm⁻³. A schematic of the seeding rig is given in Figure 3.22. The seeding material is fluidised in a vertical column as a dry powder and then passed through a cyclone filter to remove larger particle. It is then mixed with the air upstream of the engine damping plenum volume. The intake system incorporates a 20 litre damping volume to remove any air intake oscillations and provide a reservoir of seeding particles.

The seeding rig control unit enables the control of the seeding rate of the particles into the plenum. This is achieved by altering the duration and frequency of the motion of the pneumatic shuttle dispenser piston. In addition, a second circuit was introduced to enable a portion of the exhausted air and seeding mixture to be fed back into the inlet damping volume.
A simple exhaust gas recirculation loop with a gate valve allowed the seeding particle density in the plenum to be augmented as required for higher engine speeds. This mechanism was also used to assess the sensitivity of the LDA system. Engine-out powder samples were analysed, using the Scanning Electron Microscope (SEM) in the Electron Microscopy Unit at the University of Brighton, to assess sample purity (particle quality, sphericity) and to verify the mean particle size range of approximately 1-2 μm.

The air and particle mixture is drawn from the plenum reservoir into the engine inlet manifold during normal engine operation. The manifold depression is set manually by a butterfly throttle valve located upstream of the intake plenum and manifold runners. The absolute manifold pressure and exhaust backpressure are measured by two Kistler type 4045A2 piezo-resistive pressure transducers and charge amplifiers.

An exhaust filter system removes the seeding particles from the exhaust flow whilst minimising backpressure, and hence reverse flow, into the engine. The filtration is achieved by the use of a cyclone unit fitted with a fibre air filter, chosen to eliminate particles between 1 and 2 μm in diameter. An extractor fan placed above the filter bank controlled the pressure drop across the filter and along the exhaust system. The pressure drop could be altered by a butterfly valve vented to atmosphere. The fibre filters were replaced with active carbon filters for fuel injection studies.

(d) High-Pressure Fuel Injection System

The Ricardo high-pressure fuel delivery system provides fuel to the injector at the required temperature and pressure and incorporates many safety features. It is comprised of a low and high-pressure pneumatic air system, a fuel de-gassing unit, two water-cooling circuits and a low and high-pressure fuel system.

The low-pressure, 1 bar regulated air system initially purges the fuel system of previous residues found in the reservoirs, pipes and galleries. A low-pressure lift fuel pump then transports fuel to a header reservoir that then feeds several measurement burettes instrumented with a type K thermocouple. The fuel is then pumped by a 12 volt car fuel pump through a fuel filter and a multi-pass water heat exchanger at a pressure of 1.25 bar. The low-pressure fuel is then compressed to 110 bar by a piston displacement pump that primes the high-pressure galleries. When fuel is required, a pneumatic dump valve operated by the high-pressure (6 bar) air system transfers high-pressure fuel to the fuel injector line at 100 bar through a single-pass water heat exchanger. Gauges mounted on the user panel of the rig indicate the high-pressure gallery and fuel line pressures as shown in Figure 3.23. The controls shown in the photo allow the fuel pressure to be manually controlled.

(e) Engine Timing Resolver and Ignition and Injection Systems

An engine timing resolver is utilised to provide accurate crank angle (time base) and stroke information for injection and ignition signals and to provide trigger locations for camera and LASER diagnostic equipment. The crank angle position is relayed via a Digitech DE1028 optical shaft encoder that generates 1000 pulse per revolution and a once per revolution marker pulse.

A Lumenition optical pick-up set at TDC non-firing (0 CA) is utilised on the camshaft to provide discrimination between two (360 CA) and four (720 CA) stroke operation using a simple logic circuit.
Three visible LED lights on the front of the unit indicated that all three signals; encoder pulse, encoder marker pulse and camshaft pick-up pulse, were correctly aligned with the engine non-firing, TDC position. A wiring schematic of a typical engine set-up used in these studies is shown in Figure 3.24.

3.5.3. Summary of LDA and PDA Systems
In the course of the research work undertaken, two LDA systems have been used. The first system will be referred to as the RCE Counter LDA and the second, as the Dantec MultiPDA. Both are single component Differential Doppler systems operating in a backscatter mode. Consequently, only one window is required into the combustion chamber.

The RCE Counter LDA is a classical LDA configuration, utilising a rotating diffraction grating to achieve the beam splitting and the necessary frequency shifting. The RCE classical LDA Differential Doppler Anemometer is shown in Figure 3.7a, b. and utilises a dove prism arrangement for beam rotation. LASER beam transmission is achieved by highly reflecting (99%) broadband dielectric mirrors. Although the RCE Counter LDA was re-commissioned for some preliminary results, it will not be further explained here as the majority of results were obtained with the newer Dantec system. A comparison of both LDA systems and the operating parameters available for the studies carried out within this thesis is presented in Table 3.4. The Dantec LDA system was acquired after the start of the work, through a successful Engineering and Physical Sciences Research Council (EPSRC) grant application. This system is capable of carrying out short and long-range velocity and turbulence intensity measurements as well as PDA measurements of sprays and uses a Bragg Cell to achieve the requisite frequency shift after beam separation.

3.5.4. Summary of Experimental High-Speed Photographic Systems
The characteristics of direct injection, high-pressure gasoline fuel sprays were studied with the use of four high-speed camera systems. The Engineering and Physical Sciences Research Council (EPSRC) loan pool provided two of these systems. Each camera was used with a point source or light sheet illumination provided by a Spectra-Physics Stabilite 2017 Argon-ion CW LASER (green, 514.5 nm) and a Melles Griot, HeNe CW LASER (red, 632.8 nm). In addition, flash photography and shadowgraphy were performed with diffuse backlighting provided by two, high-power, halogen flash lamps. LASER light sheet generation and beam positioning were achieved with mirrors and cylindrical lenses, selected to produce a collimated sheet of approximately 1 mm in thickness. The sheet height and rotation were adjusted depending upon the type and orientation of optical access preferred. The method employed for these studies is given in more detail in the relevant experimental chapters. The four systems are summarised:

(a) Kodak EktaPro HS Motion Analyzer, Model 4540
The Kodak EktaPro camera utilises a high-speed video recording system to capture up to 4,500 full frames per second (fps) of 256 x 256 pixels each and with 256 levels of grey. Higher frame rates are achieved by a reduction in frame size. These rates increase incrementally from 9000 fps (exposure rate of 111 μs) to 40,500 fps (exposure rate of 25 μs) with a final image size of 64 x 64 pixels. Images are stored digitally at up to 5,120 images for full frame pictures or 81,920 images at the minimum picture size.
Event image capture is provided by the use of both pre and post-triggering capabilities. Additionally, captured images can be transferred to videotape or to a PC through an IEEE-488 bus. Image size selection and frame rates can also be selected through the PC digital data interface (DDI).

(b) Sony Mini DV Handycam Vision DCR-TRV900E and miroVIDEO™ DV300 PC card
The Sony digital video cassette camera is a compact, tripod or handheld camera with both optical and digital zoom capabilities. The selectable shutter speed of up to 1/10,000 allows instantaneous events to be captured, however sequences are recorded at the PAL standard of 25 fps. Unfortunately, the camera cannot be triggered externally. As such, the primary role of this camera was as a quick and easily configurable investigative tool used to highlight regions of the fuel spray that might best be further studied by the more dedicated high-speed camera systems. The digital video is transferred to the dedicated miroVIDEO™ DV300 PC card via IEEE1394 (Firewire).

(c) LAVision Flowmaster CCD Camera
The LAVision Flowmaster is a 12 bit charge coupled device (CCD) camera capable of high-resolution digital image capture with shutter speeds from 100 ns to 1000 s. The full-frame image resolution is 1280 x 1024 pixels. In DoubleShutter mode, two images can be captured with 200 ns interframing time. The CCD array operates at −12 °C. The camera is linked to the PC via fibre optic and is controlled through the LAVision DAVis software.

(d) DRS Technologies IMACON 468 Ultra High-Speed Digital Imaging System
The Imacon 468 is an ultra high-speed digital camera that utilises a pyramid beam splitter to provide up to eight high-resolution images from a single optical input. Each image is projected onto an output window that is coupled to an independent intensified CCD of 576 x 385 pixels. Each ICCD has a variable exposure duration, individual gain and independent interframe time. Exposure times on each channel range from 10 ns to 1 ms in 10 ns steps. The interframe times range from 10 ns to 10 ms in 10 ns steps. This gives a framing rate of between 100 and 100,000,000 fps. As with the Kodak system, the Imacon can be triggered prior to, during or after an event. The system operating parameters and data analysis are controlled from a dedicated PC. The software includes functions for image modification, colourisation, filtering, binary conversion and tools for the calculation of velocity, distance, angle and area.
3.6. Conclusions of Chapter 3

This chapter reviewed the theory of the experimental measurement techniques of LDA and PDA. In addition it highlighted the practice of such techniques as applied to the specific environment of in-cylinder airflows and fuel sprays. The intrinsic relationship between the type of measurement, the theory upon which it is based and its application in these special cases was stressed. The correct interpretation of the results is dependent upon a complete understanding of the experimental apparatus (and its set-up) and the physics of the processes involved. In LDA and PDA, the absence of data is often of equal importance to the collection of validated data for a complete description of an airflow or fuel spray.

LDA and PDA are two techniques that offer the high temporal and spatial resolution required to resolve the smallest scale structures in the air and fuel flowfields. PIV can be utilised to provide a mean, bulk flow description over the complete field for comparison. A combination of both techniques can be used to resolve different scales of fluctuation in an unsteady flowfield or to corroborate one set of measures against another. High-speed photography offers full-field visualisation of flames, fuel sprays or seeded airflows. These first indications can be used to provide a specific target for the higher resolution techniques.

Where possible the LDA or PDA is best utilised in a forward scatter mode to optimise the light scattering intensity. In IC engine research however, optical access is often restricted and a back-scatter arrangement can be used as a compromise. The back-scatter signal is of a lower intensity, the SNR is reduced and the data rejection rate is increased. This results in longer experimental engine runs to achieve significant data sets. The specific measurement requirements must therefore be studied in detail and optimised for the particular experiment. In particular, the particle seeding of airflows, directional ambiguity resolution and Doppler signal processing must be established prior to experimental data acquisition. Particle seeding of the airflow is of particular importance. A seeding particle must be chosen that is able to follow the flow velocity and frequency sufficiently closely whilst being large enough to scatter sufficient light. Particle concentrations are equally important in IC engine studies. The 'best' signal is achieved when only one particle is present within the probe volume. This is often difficult to attain in engine studies, where the induced volume of air and particles is present throughout the intake, compression and power strokes. The same limitations can be observed in dense fuel spray measurements using PDA. However, the majority of fuel spray measures performed upon automotive fuel sprays are limited to ambient pressure bombs or single shot, high-pressure vessels. The application of PDA within an unmodified reciprocating engine and the inherent measurement difficulties (and measurement interpretation) that this presents is rarely presented or discussed in the literature.

The chapter describes both the correct practice and limitations of the LDA and PDA techniques as applied to in-cylinder airflows and fuel sprays. It is concluded that the SNR is the most significant cause of experimental uncertainty or data rejection under these circumstances. LDA and PDA must be treated in different respects as regards data analysis. Nonetheless, the principal measurement uncertainties are those due to the reciprocating motion of the engine and the cycle encoder resolution, velocity bias, velocity gradient and to a lesser extent, finite transit time resolution. A full analysis of experimental results must take into account these effects upon the repeatability of the data result sets.
Figure 3.0. The Principles of the Differential LASER Doppler Anemometry Technique

Figure 3.1. Probe Volume Construction and Dimensions
Figure 3.2. Vector Diagram for The Heterodyne Model of a Dual Beam LASER Doppler Anemometer
Fundamental Transverse Mode, TEM$_{00}$

Gaussian Radial Intensity Distribution

$$I = I_0 \exp\left(-\frac{8r^2}{4r_0^2}\right)$$

**Beam Radius, r (m)**

**Gaussian Beam Diameter, 2r$_0$ (m)**

**Effective Probe Volume Width**

**Figure 3.3. Definition of Gaussian LASER Beam Diameter and Effective Probe Volume Dimensions**
Linearly polarised incident light of intensity, $L_1$ and wavelength, $\lambda$

Continuous medium of known refractive index

Particle, $P$ of diameter, $d_p$ and known refractive index

Figure 3.4. Principles of Mie Scattering from a Homogeneous Spherical Particle
Figure 3.5. Polar Plot of Relative Intensity of Light Scattering with respect to the Incident Beam Direction for a Typical LASER Wavelength with Varying Particle Sizes (Dantec, 2000)

\[ d_p \equiv 0.2\lambda \]

\[ d_p \equiv 1.0\lambda \]

\[ d_p \equiv 10\lambda \]
Figure 3.6. Logarithmic Polar Plot of Relative Intensity of Light Scattering with respect to the Incident Beam Direction for a Typical LASER Wavelength: Effect of Incident Light Polarisation upon Light Scattering Modes (Dantec, 2000)
Figure 3.7b. Ricardo Classical LASER Doppler Anemometer System
Glass disk with three tracks of radial lines. The line pair spacing varies in a radial direction.

- Number of line pairs (Track 1): $N = 16384$
- Line Pair Width: $6.08 \, \mu m$
- First Order Diffraction Angle for $\lambda = 514.5 \, nm$: $4.85^\circ$
- Maximum Frequency Shift using First Order Beams: $10 \, MHz$
- Motor Speed: $350 \, revs^{-1}$
- Direction of Rotation: Reversible

**Governing Equations**

Diffracted beams occur at angles that satisfy:

$$\sin \theta = \left( \frac{p \lambda}{d} \right)$$

where $p$, defines the diffraction order, $\theta$, defines the diffraction angle measured from the normal and $d$, defines the line pair spacing on the grating.

Rotating the grating at constant speed, $f$, revs$^{-1}$, with $N$ line pairs, the two first order beams give a net frequency shift

$$f_s = 2Nf$$

*Figure 3.8. Details of the Rotating Diffraction Grating*
Figure 3.9. Typical LDA Doppler Signal Burst
Continuous medium of known refractive index

Liquid Spray of known refractive index

Scattering plane

Light receiving optics

Location of Photo detectors

Figure 3.10. Schematic of a Conventional PDA System

Figure 3.11. Scattering Mode Based upon Scatter

Variation of refractive index with angle of incidence

Figure 3.12. Scattering Angle due to Meridional Plane
Figure 3.11. Scattering Modes Based upon Simple Geometric Optic Theory

Variation of Reflectivity with Angle of Incidence

Figure 3.12. Brewster's Angle for a Water Droplet in Air
Figure 3.13a. Effect of Particle Size upon the Phase-Diameter Relationship in a Commercial PDA System (from Dantec, 2000)

Figure 3.13b. The Use of Three Photodetectors to Resolve the $2\pi$ Ambiguity Problem in the Phase-Diameter Relationship (from Dantec, 2000)
Figure 3.14. The Trajectory Ambiguity and Slit Effects (from Gouesbet and Gréhan (2000))

Figure 3.15. Light Collection Vignetting (Collection optic from Dantec, 2000)
Figure 3.16. Plan of the Optical Engine Test Cell
Figure 3.17. The Ricardo Single Cylinder Optical Research Engine (schematics courtesy of Ricardo Consulting Engineers)
Figure 3.18. Extended Piston, Dry Lubrication Piston Ring Arrangement
Figure 3.19. Quartz Piston and Optical 'Tipping' Mirror Build (schematics courtesy of Ricardo Consulting Engineers)
Figure 3.20. Research Engine, Plinth and Ancillaries

Figure 3.21. Layout of Engine Data Acquisition and Control Systems
Figure 3.22. LDA Particle Seeding Rig
Figure 3.23. The Ricardo High-Pressure Fuel Delivery System
Figure 3.24. Typical Installation:

Engine Timing Resolver and Ignition and Injection Systems
<table>
<thead>
<tr>
<th>Property</th>
<th>Air Flow and Fuel Droplet Velocity</th>
<th>Fuel Spray Characterisation</th>
<th>Species Concentration and Distribution</th>
<th>Temperature and Combustion Studies</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coherent Anti-stokes Raman Scattering (CARS)</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
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<tr>
<td>Degenerate Four Wave Mixing (DFWM)</td>
<td>x</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Doppler Global Velocimetry (DGV)</td>
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<td></td>
<td></td>
<td></td>
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<tr>
<td>High-Speed Photography/Video/Digital/Cine</td>
<td>x</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Heterodyne Holographic Interferometry</td>
<td></td>
<td></td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>LASER Doppler Anemometry (LDA)</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LASER Induced Fluorescence (LIF)</td>
<td>x</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LASER Induced Incandescence (LII)</td>
<td></td>
<td></td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>Particle-counter-sizer-velocimeter (PCSV)</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Phase Doppler Anemometry (PDA)</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
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<tr>
<td>Point Diffraction Interferometry (PDI)</td>
<td></td>
<td></td>
<td>x</td>
<td></td>
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<tr>
<td>Particle Image Velocimetry (PIV)</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
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<tr>
<td>Particle Tracking Velocimetry (PTV)</td>
<td>x</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Planar LASER Induced Fluorescence (PLIF)</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
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<tr>
<td>Rayleigh Scattering</td>
<td>x</td>
<td>x</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Raman Doppler Velocimetry (RDV)</td>
<td>x</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Raman Spectroscopy (RS)</td>
<td></td>
<td></td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>Schlieren Method (non-quantitative)</td>
<td></td>
<td></td>
<td>x</td>
<td></td>
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</table>

*Table 3.0. Experimental Optical Methods Applied in IC Engine Research*
<table>
<thead>
<tr>
<th>Particle Material</th>
<th>Fluid Environment</th>
<th>Approximate Diameter Range (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicone Oil</td>
<td>Atmospheric air</td>
<td>0.8-2.6</td>
</tr>
<tr>
<td>Titanium Dioxide</td>
<td>Atmospheric air/combustible mixtures/oxygen plasma (2800K)</td>
<td>0.4-3.2</td>
</tr>
<tr>
<td>Magnesium (II) Oxide</td>
<td>Atmospheric air/methane-air flame (1800K)</td>
<td>0.8-2.6</td>
</tr>
<tr>
<td>Zirconium Dioxide</td>
<td>Combustion studies</td>
<td>0.6</td>
</tr>
<tr>
<td>Ammonium chloride</td>
<td>Atmospheric air</td>
<td>0.1-1.0</td>
</tr>
<tr>
<td>Water</td>
<td>Atmospheric air</td>
<td>1.0-2.0</td>
</tr>
<tr>
<td>Evaporated engine oil</td>
<td>Hot atmospheric air</td>
<td>2.0-3.0</td>
</tr>
<tr>
<td>Aluminium (II) oxide</td>
<td>Hot atmospheric air</td>
<td>&lt;8.0</td>
</tr>
</tbody>
</table>

Table 3.1. Common Particle Materials and Size Ranges
Suitable to IC Engine LDA Applications

<table>
<thead>
<tr>
<th>Doppler Processor</th>
<th>Method of Doppler Frequency Estimation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Cambridge Consultants Ltd. Mk 1</strong></td>
<td>Frequency tracker</td>
</tr>
<tr>
<td>Doppler Signal Processor and Mean Frequency Counter</td>
<td></td>
</tr>
<tr>
<td><strong>TSI Model 1980b Doppler Processor</strong></td>
<td>Period signal counter</td>
</tr>
<tr>
<td>Model 1984b Input Conditioner / 1985b Timer</td>
<td>Counting 16 fringes with 1-20% tolerance on 10/16 comparison.</td>
</tr>
<tr>
<td>Model 1988 Analogue Output</td>
<td></td>
</tr>
<tr>
<td><strong>DISA Model 9055L Doppler Processor</strong></td>
<td>Period signal counter</td>
</tr>
<tr>
<td>Model 55L96 LDA Counter</td>
<td>Counting fringes with comparison tolerance of 1.5-12%</td>
</tr>
<tr>
<td>Model 55L91 Data rate module / 55L97 HV</td>
<td></td>
</tr>
<tr>
<td><strong>Dantec/Invent 58N80 MultiPDA</strong></td>
<td>Covariance signal analysis</td>
</tr>
<tr>
<td>Correlation Type Processor</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.2. Doppler Signal Processing
<table>
<thead>
<tr>
<th>System</th>
<th>Engine Water</th>
<th>Engine Oil</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid</td>
<td>Water/glycol-50/50 mixture</td>
<td>Standard Motor Oil</td>
</tr>
<tr>
<td>Heater</td>
<td>1 kW Santon immersion heater.</td>
<td>2 kW Santon line heater.</td>
</tr>
<tr>
<td>Temperature</td>
<td>6 times type K thermocouple</td>
<td>6 times type K thermocouple</td>
</tr>
<tr>
<td>Measurement</td>
<td>1 times PRT</td>
<td></td>
</tr>
<tr>
<td>Heater Controller</td>
<td>Tactical 310 PID heater controller.</td>
<td>Horiba Heated Line PID controller.</td>
</tr>
<tr>
<td>Temperature Setpoint</td>
<td>80 °C</td>
<td>60 °C</td>
</tr>
<tr>
<td>Pump and Motor Type</td>
<td>Beresford B30 impeller water pump and 3ph induction motor.</td>
<td>Albany L254 gear pump and GEC Machines Alpak VP4 induction 3ph motor.</td>
</tr>
<tr>
<td>Operating Pressure</td>
<td>2 bar</td>
<td>3 bar</td>
</tr>
<tr>
<td>Scavenge Pump</td>
<td>--</td>
<td>KMEW motor trochoid gear oil pump.</td>
</tr>
<tr>
<td>Filter</td>
<td>--</td>
<td>Standard automotive oil filter.</td>
</tr>
<tr>
<td>Mixing Valve</td>
<td>--</td>
<td>Satchwell ALX 1256 3-Port Mixing Valve and PRT.</td>
</tr>
<tr>
<td>Mixing Valve Controller</td>
<td>--</td>
<td>West 2071 PID Microtune Controller.</td>
</tr>
<tr>
<td>Reservoir</td>
<td>Pressurised water header tank.</td>
<td>Engine sump</td>
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*Table 3.3. Water and Oil Systems*
<table>
<thead>
<tr>
<th>Details of System</th>
<th>RCE Classical LDA</th>
<th>Dantec FibreFlow LDA/PDA</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>For A=514.5 nm</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Focal length of objective (mm)</td>
<td>400</td>
<td>50</td>
</tr>
<tr>
<td>Measuring volume (μm)</td>
<td>188</td>
<td>121</td>
</tr>
<tr>
<td>Measuring volume (mm)</td>
<td>1.72</td>
<td>1.52</td>
</tr>
<tr>
<td>Beam intersection (°)</td>
<td>6.25</td>
<td>4.59</td>
</tr>
<tr>
<td>Fringe spacing (μm)</td>
<td>2.36</td>
<td>3.22</td>
</tr>
<tr>
<td>No. of fringes in measuring volume</td>
<td>80</td>
<td>38</td>
</tr>
<tr>
<td>No. of fringes viewed by PMT</td>
<td>72</td>
<td></td>
</tr>
<tr>
<td>Obliquity of transmitting to receiving (°)</td>
<td>6.61</td>
<td>6.61</td>
</tr>
<tr>
<td>Max. frequency shift (MHz)</td>
<td>10</td>
<td>40</td>
</tr>
</tbody>
</table>

**LASER**
- Lexel Model 95
- 4W Argon Ion
- Spectra-Physics Stabilite 2017 5W Argon Ion and 2670 Remote Controller

**Frequency shifting**
- TPD rotating diffraction grating
- DISA Bragg cell

**Photomultiplier**
- RCA-4526
- Disa 55X08

**Signal Processor**
- 1. Cambridge Consultants Ltd. Mk 1 Tracker and Mean Frequency Counter
- 3. DISA 55L96 Counter
- Dantec/Invent Adaptive MultiPDA 58N80 Correlation Type Processor

**Transmission System**
- Free beam
- Dantec 60x24 Fibre-Optic Manipulators Anaconda Sealite Single Strand Fibre-Optic cable and light sheath. Dantec Ø 14 mm FibreFlow Probe Head

**Collection Optic**
- Backscatter lens
- Dantec PDA 57X10 Classical

**Beam expansion**
- Dantec Ø 60 mm X4 Beam Expander for 160 and 400 mm

**Interface**
- PC with National Instruments Data translation PCI Interface Board 58G30

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*Table 3.4. Comparison of LDA Systems and Optical Parameters*
4.0. Airflow Measurements in a Pent-roof, Manifold Injection, Gasoline Combustion Chamber

4.1. Introduction

This Chapter and the subsequent Chapter present the results of experimental studies of the characteristics of the in-cylinder air motion in two motored, pent-roof, gasoline engines. In this Chapter, a modern manifold injected gasoline engine is presented. This provides an experimental baseline for comparison with other systems. The following Chapter presents results from a top-entry, G-DI combustion system. In each case, mean flow and turbulence characteristics are compared and their differing roles within each system discussed.

Experimental measurements were carried out using the 'Through Glass Piston' technique, with an extended upper piston and angled mirror. The measurements were carried out on a single cylinder of a Volvo B230 manifold injection Spark Ignition cylinder head. Measurements of the instantaneous velocity were made in the mid-cylinder tumble plane; in the pent-roof and cylinder bore. The cylinder head and intake ports were considered to give a relatively low tumble ratio on a steady state flow rig.

The experimental apparatus and methods used in this study are described in detail along with the LDA instrumentation and optical set-up. Some of the practical problems (early measurement indicators) associated with these types of measurement and their potential solutions are outlined. One-dimensional velocity measurements were carried out in the pent-roof of the combustion chamber and over the swept volume in the mid-cylinder tumble plane. The method of raw data preparation and analysis is reproduced from Chapter 3 for clarity. A preliminary data set is reproduced to demonstrate the salient features of a 'typical cycle', with particular regard to instantaneous and averaged velocity characteristics, across the four-strokes of the motored cycle. The valve opening, peak lift and closing events are clearly marked at their respective crank angles.

Chapters 2 and 3 previously noted that extraction of reliable information about the mean and RMS velocity characteristics from LDA data required careful consideration. The signal measured is comprised of turbulence generated by the physical fluid motion and that due to system measurement noise and velocity bias. Each velocity measurement recorded during an experiment is composed of mean and turbulent components that vary throughout the four strokes of the cycle and from cycle to cycle. As such, the data in the results section of this Chapter describes the temporal characteristics of the ensemble-averaged mean and RMS velocities presented across the intake and compression strokes. In addition, the important spatial structure of the dynamic mean flow field is described. A method of re-construction and refinement of the LDA mean velocity spatial field is qualitatively compared with velocity vector plots gained from PIV data obtained in the same engine geometry.
In the final section of the Chapter, the systematic errors due to the measurement method and optical set-up are discussed. Also, an assessment of the statistical uncertainty present in the ensemble-averaged mean and RMS estimates is given along with an appraisal of the criteria for data smoothing for visualisation purposes. The conclusions drawn from this preliminary study were taken forward and implemented within the larger scale study presented in Chapters 5 and 6.


4.2.1. Experimental Test Procedure

A comprehensive description of the engine, test-cell and experimental apparatus is given in Chapter 3 for reference. This section therefore describes only those features particular to these specific sets of experiments.

4.2.1.1. Engine Configuration

The experimental study was conducted with a specially modified single cylinder of a four cylinder, 2.3 litre, 16 valve, dual overhead camshaft (DOHC) cylinder head as shown in Figure 4.0. Table 4.0. details the specification of the research engine. Optical access to the combustion chamber was achieved by the insertion of a 67 mm diameter quartz window into the piston crown as described in Chapter 3. The diameter was restricted in size by the valve cut-outs required in the piston crown. The valve-timing diagram for this engine is given in Figure 4.1.

A rectangular, tilted, 'Tipping mirror' was housed within the extended piston barrel, fixed centrally upon two half-shafts set in needle bearings. One half-shaft was coupled to a 360° rotational vernier mounted externally to the extended barrel. This enabled the mirror to be inclined to the horizontal plane. In addition, the mirror was tilted by 8.4° about the plane of its surface. This ensured obliquity between the transmitted LASER beam pair and reflected light at the receiver lens.

A timing logic circuit was constructed to gate alternate TDC pulses from the crankshaft optical encoder with an optical pickup situated on the camshaft set to 0° CA. The resultant signal gave 2000 pulses per 4-stroke cycle, or 2 turns of the crankshaft. The Doppler processor used this signal as an external trigger/clock reset input. A time history (from the start of the cycle) of all measurements was then available for measurements over the full 720° CA at a given engine speed.

Monitoring of in-cylinder peak pressure and inlet manifold depression (IMAP), as indicators of correct engine operation, was achieved by the use of piezoelectric and piezoresistive pressure transducers coupled to charge amplifiers and a digital storage oscilloscope. To measure in-cylinder pressure, the cylinder head was modified to accept a Kistler Type 6121 transducer and Type 5001 charge amplifier. The transducer was mounted flush with the wall of the combustion chamber. For IMAP measurement, a Kistler 4045A2 transducer and amplifier was installed in the inlet manifold plenum chamber directly after the throttle valve. The experimental set-up is illustrated schematically in Figure 4.2. and photographically in Figure 4.3.
4.2.1.2. Measurement Locations, Automated, Rectilinear Traverse and LASER Beam Targeting

The experimental measurement locations within the Volvo engine build were divided into three sets as shown in Figure 4.4. The orthogonal axis set x, y and z are defined. The first set of measurements (SP-series) were performed along a line that was the axis of the spark plug location, offset by 2.5 mm from the cylinder axis (y-axis datum), and at four different heights above the gas face (z-axis datum). The second set of measurements (SW-series) were performed over the tumble, mid-cylinder plane, above the gas face, within the pent-roof combustion chamber. Finally the third data set was acquired for a vertical measurement plane coincident with the mid-cylinder, tumble plane, to a depth of 70 mm below the gas face. The parameter file naming convention for the T-series data set was as follows:

**Tumble Plane: Filename- T(P/N)x(P/N)y**

where P and N denote a positive or negative direction respectively in either the x or y-axis directions as defined in Figure 4.4. The letters x and y are replaced with the distance in (mm) from the origin, T00P00. The z-axis value forms the directory structure due to the restriction of the number of characters permitted by the MS-DOS file naming convention.

Locating the measurement positions was achieved by an automated Time and Precision and North East Electronics, rectilinear traverse that was installed upon an Ealing Electro-Optics Optical breadboard and anti-vibration air table as shown in Figure 4.5. This was used to position the beam expander focal lens and LASER beam pair on the angled mirror. Measurements at each location were made in two orthogonal directions perpendicular to the optical axis by rotating the extended barrel and mirror through 90°. The convention was defined as follows:

| U-Direction: | Perpendicular to the crankshaft axis and towards the intake. [Tumble] |
| V-Direction: | Parallel to the crankshaft axis and positively towards the cam pulleys. [Cross-tumble] |

For each point to be examined it was necessary to find the true LASER beam intersection probe volume location within the combustion chamber. An etched Perspex target grid was positioned upon the extended barrel in place of the cylinder head as shown schematically in Figure 4.6a, b, c. Shims and spacers were used to alter its height relative to the gas face.

The engine was turned to mid-stroke and the LASER beam intersection probe volume then targeted at points on the grid. The alignment of the optical axis depended upon the position of the measurement but was found to be approximately parallel to the cylinder axis for points near the centre of the chamber. By rotation of the mirror and jogging the x and y co-ordinates of the traverse, the probe volume was positioned close to each of the grid points.
The method of PMT anode current 'peaking' was utilised to ensure the probe volume was correctly targeted. The LDA processor was set-up to simulate the arrival of Doppler bursts by selection of a burst detector mode that uses a 26 kHz, internally generated, trigger signal. As the probe volume approaches the target grid at low LASER power, the PMT anode current rises until a plateau is reached. The anode current will reach a maximum value at the point where contact is made with the surface and the backscatter signal is strongest. Additionally, as the probe volume is brought close to the intersection of two x-y etched lines on the Perspex grid, LASER light visibly propagates along the converging lines upon the surface of the target plate in the x-y plane. The positioning in the third dimension is achieved by maximizing the anode current. The final position of the traverse (transmission optic) is then digitised and input into a software routine and the angle of the mirror is recorded against this data. The targeting process was repeated for both the 'Sweep' (SW) and 'Point' (SP) positions defined previously. The individual 'Sweep' and 'Point' positions are given in Table 4.1. with the Computer Numerical Control (CNC) code used to automate the traverse. The resultant automated traverse routines showed good repeatability with an estimated error of approximately +/- 0.3 mm in beam position. This method was necessary due to the unknown nature of the particular engine to be investigated. Further measurements used mathematical relationships, based upon glass material and geometrical properties of the engine and optics, to relate the probe volume location with that of the transmission optics for a given mirror orientation (Savic and Begg, (1998)).

4.2.1.3. LDA System Set-up and Measurement Method

In a preliminary series of instrument range and sensitivity tests, mean and RMS velocity were measured at several locations within the combustion chamber. With each configuration, a certain amount of 'fine tuning' was required of the LDA parameters. The seeding rig air and particle delivery rate was optimised at approximately 80 litre/min. The LDA data validation rates were seen to increase with increased seeding density but optical window fouling resulted in reduced experimental runs and fewer validated Doppler events. For the purposes of the first series of long focal length tests, the final LDA set-up was as detailed in Table 4.2.

The Doppler processor velocity channel hardware filter bandwidth was initially selected in the central 1.20 MHz range, giving a velocity measurement range of between -3.86 and +3.86 ms⁻¹. The available frequency bandwidths and corresponding velocity ranges (based upon optical set-up and frequency shift) are given in Table 4.3. The hardware burst detector bandwidth was then selected to give the largest minimum particle residence time in the probe volume smaller than the minimum calculated residence time. The minimum calculated residence time is determined from the probe volume dimensions and the maximum permissible velocity for the selected bandwidth. The minimum calculated burst period inversely corresponds to the minimum calculated residence time. These parameters set the temporal and spatial resolution of the LDA system.

In the Dantec system, the default value is equal to the smallest value greater than 0.71 times the filter bandwidth divided by the minimum fringe count. For the 1.20 MHz range with 38 fringes, the transit time clock resolution (effectively the cycle counting resolution) is 22.3 μs/bit.
The minimum calculated transit time is 31.67 µs and therefore the minimum allowable burst detector bandwidth is 11.21 kHz. From the above rule, the nearest available burst detector bandwidth is 30 kHz. The system hardware amplification gain was set to 'high' (10 dB greater than the 'low' setting).

For all the measurement points the photomultiplier tube voltage was varied from between 800 to 1200 volts under steady state flow conditions. At higher voltages and close to surfaces or during phases of dense seeding, the photomultiplier anode current protection circuit was operated. Processing these results showed spurious velocity measurements attributed to PMT signal saturation. A compromise, reduced, voltage of approximately 1 kV was used for all measurement points at the expense of data rate. For those points close to the pent-roof and glass window surfaces a lower value was used.

The effect of velocity channel signal validation was also investigated. The signal validation threshold was varied from -6 dB to +3dB. Any signal with a SNR below the selected level was rejected based upon a set of signal level criteria. Again, the effect of signal validation varies with measurement location and the proximity to fixed or moving surfaces. Points close to reflecting surfaces produced higher levels of optical noise and the SNR deteriorated significantly resulting in validated data rates too low to record with any significance. An intermediate value of -3dB was selected for the majority of points in the chamber that gave a satisfactory data rate. Where the data validation rates were low, the running time of the engine was extended to enable acquisition of the required number of samples.

The LDA data from the Dantec MultipDA 58N80 processor was recorded using Dantec FLOware version 3.3 software on a desktop PC and exported as raw data (including all Doppler measures) and as ensemble-averaged plots of mean velocity and turbulence intensity over the complete engine cycle.

4.2.1.4 Experimental Method and Engine Preparation

The upper glass piston was inserted into the extended cylinder barrel by the method described in Chapter 3. The piston crown and combustion chamber were coated with black cobalt oxide. The cylinder head with valves was then assembled on top of the extended cylinder barrel. Correct orientation was observed by location of the cylinder head and gasket by two ring dowels. The four cylinder head bolts were tightened to 60 Nm following the specified order. The camshafts, hydraulic followers and bearing shells were then assembled in the correct order. The remaining parts of the oil and water circuits were then completed and tested. Throughout the tests, the oil was maintained at 3 bar pressure and 60 °C temperature. The water temperature and pressure were 80 °C and 2 bar respectively. Once the engine had attained stable oil and water, pressures and temperatures, the timing belt was fitted and valve timing was carried out using TDC Firing as the reference crank angle. This ensured that the hydraulic followers were under pressure and that the engine had thermally expanded to its correct dimensions and clearances. This is important in a single cylinder, extended piston arrangement, where the overall engine height and hence timing belt length are greatly increased.
A dial gauge and threaded dog-legged bar were positioned on the edge of each camshaft bucket in turn. The method of peak valve lift timing was used by turning the engine flywheel over by hand in the direction of normal engine rotation with the toothed dynamometer coupler disengaged. This ensured that the timing belt was always tight on the drive-side of the crankshaft. Fine-tuning of the valve timing was achieved using the slots provided in the camshaft pulleys. The belt slack was then taken up by the use of an eccentric, idler pulley located on the belt side opposed to the drive-side. The engine was then turned over by hand for several rotations and the timing marks verified.

The Lumenition camshaft pick-up and Digitech DE1028 crankshaft optical encoder were then fitted and set to the TDC Non-Firing condition. Finally the rocker cover, intake and exhaust manifolds, spark plug, thermocouples and pressure transducers were put into place. Prior to the tests, the throttle valve setting was positioned at WOT and the seeding rig delivery pipe was connected to the damping plenum with no exhaust gas recirculation.

The engine was then motored at 500 and 1000 rpm ± 5 rpm. The throttle was then set to give 800 mbar IMAP during the intake stroke as recorded by a digital storage oscilloscope. The engine camshaft and crankshaft timing signals and in-cylinder pressure traces were also monitored to ensure correct engine operation. A small drop in in-cylinder pressure in all engine cycles suggested incorrectly bedded upper piston rings at this stage. With particle seeding of the intake system, the pressure drop over consecutive cycles was monitored. A significant drop was then taken as an indication of piston ring failure.

### 4.2.1.5. Data Acquisition

In each case, the following results were required from the raw data sets:

- Temporal evolution of the mean and RMS velocities over the engine cycle.
- The spatial mean velocity field during the intake and compression strokes.

The method of processing raw data was as described by Hadded and Denbratt (1991), to ease direct comparison of results. Data was recorded as sequential velocity and crank angle pairs, \( v_i \) and \( \text{CA}_i \), where \( i = 1 \) to \( n \) samples and \( n \) was the total number of validated data points recorded over many consecutive engine cycles. The mean values of the velocity and RMS turbulence intensity were calculated by ensemble-averaging the data in 500 discrete intervals, each of width \( \frac{720}{500} = 1.44° \) CA. The series for 500 crank angle intervals or time bins is then given by

\[
E_{j}, E_{j+1}
\]

where \( j = 1, 500 \) and

\[
E_{j} = (j - 1) \left( \frac{720}{500} \right)
\]

The ensemble-averaged velocity is given by

\[
\bar{v}_{j} = \frac{1}{n_j} \sum_{i} v_i
\]
RMS Turbulence Intensity is given by

\[ v_j = \sqrt{\left( \frac{1}{n_j} \sum (v_i - v_j)^2 \right)} \]

where \( n_j \) is the total number of data points in each crank angle interval. The summation is conditional upon:

\[ EI_j < CA_1 < EI_{(j+1)} \]

The statistical accuracy of these velocity estimates can be described by the standard error, \( E \), which is defined as the standard deviation of individual mean and RMS estimates about the true mean. The standard error of the mean velocity estimate is given by:

\[ E(\bar{v}_j) = \left( \frac{v_j}{\sqrt{n_j}} \right) \]

The accuracy of the RMS estimate is affected by the shape of the velocity distribution. For the case of a Gaussian velocity distribution, it is given by:

\[ E(v_j) = \left( \frac{v_j}{\sqrt{2n_j}} \right) \]

The 3 times standard error, \( E \) bands for data are presented in Section 4.2.3.2. The probability of the true value being within these error bands is approximately 99.5%.

4.3. Experimental Results

4.3.1. Preliminary Observations and the Perspex Model Cylinder Head

The preliminary investigative results referring to Sweep 1 (SW1, gas face, mid-cylinder, (0,0,0)) and Sweep 8 (SW8, (0,0,5)) locations for instantaneous velocity measurements in the tumble, U-direction are presented in Figures 4.7. and 4.8. Each set is chosen to highlight a particular phenomenon observed in the experiments. The graphs show plots of 50,000 instantaneous velocity measurements against crank angle and particle arrival time with crank angle. The SW1 location exhibits a distinct 'bunching' of measurements around TDC with few validated measurements falling in between. At this location, the piston surface is within several (mm) of the measurement location.

The particle arrival times over the course of the measurements are continuous. This would suggest that measurements occurred across the complete cycle. However, once plotted against the engine time base, the uneven distribution of measured velocities with crank angle is confirmed. This suggested that some other mechanism was responsible for triggering false measurements. Furthermore, these effects were inconsistent with measurement location, appearing random and varying from point to point and test to test, irrespective of engine speed. This is illustrated in Figure 4.8. for Sweep location 8 (SW8) which is only 5 mm above the SW1 location into the pent-roof chamber. In this case, the results showed 'stripes' of constant velocity against crank angle.
The particle arrival time was this time, consistent with a spread of consecutive measurements across many crank angles. These early measurement indicators suggested errors due to the timing circuit or seeding density but subsequent tests using a simulated encoder signal against measured velocities in a test section, revealed no apparent errors. The configuration of the LDA system was modified extensively and in each case was calibrated against a known signal that produced 100% signal validation consistently.

With the aim of eliminating reflection effects due to the cylinder head, a simple Perspex dummy cylinder head was constructed to enable the beam intersection to be viewed. A short focal length lens and a pinhole aperture were used to verify the true intersection and fringe formation at various measurement points and throughout the complete engine cycle. Several areas of the ‘tipping’ mirror were also selected to assess the effects of the degradation of the silvered surface and the quartz window was removed and inspected for imperfections. The validated data rate was enhanced by the introduction of the recirculatory system, supplying the damping volume intake with a portion of seeded exhaust gas. Despite these modifications, the dummy head again produced evidence of ‘bunching’ of results.

Continuation of tests using a new target map in the dummy head indicated a tendency for the PMT to overload at certain crank angles. This only became apparent when the engine was turned over by hand. Approaching TDC, the backscatter signal was due to interference by reflection at the first air/quartz window interface and not due to the scattered Doppler burst. This was particularly prevalent at points along the gas face where the primary internal reflection moved across the focus of the backscatter collection lens. The intensity of the reflected light was high in comparison with that of the seeded scatterer and became more so as the angle of incidence formed with the normal to the air/quartz interface reduced. Understanding this effect, it was then possible to select angles and mirror positions that minimised the problem. The subsequent results were free of the PMT overload effects for all but a few of the total number of measurement locations.

It should be noted that it was not possible to eliminate the PMT overload effects from all the points investigated. As such, these preliminary results can only be used to infer general trends in the mean flow and turbulence intensity across specific regions of the engine cycle and that particular care must be taken in the interpretation of results measured close to TDC.

4.3.2. Features of a Typical Motored Engine Cycle Velocity Data Plot

The general characteristics of velocity measurements performed in a motored engine are illustrated in Figure 4.9. This shows a plot of instantaneous, ensemble averaged mean and RMS velocities for a measurement point selected within the combustion chamber during motored operation. This plot illustrates the salient features of a ‘typical’ set of raw LDA data over the four strokes of the cycle, at 500 rpm and acquired over approximately 1200 engine cycles. It should be noted that in this example, the data density is considered low and not suitable for rigorous statistical analysis. Nonetheless the data serves to highlight the salient features adequately. The intake and exhaust valve timings and peak lifts are included for reference. The data has been processed as discussed in Section 4.2.1.5.
The plot symbol style has been selected to highlight those regions in the engine cycle that have a high data density. These produce a bold shadow in the instantaneous velocity results. The engine cycle can be divided up into the four strokes: intake, compression, expansion and exhaust. Zero degree crank angle (0 CA) is defined as TDC non-firing. During the intake phase and up to the point of maximum intake valve lift and piston speed, the instantaneous velocity is at a maximum as the air is rapidly drawn into the cylinder to fill the available volume. As the piston approaches BDC, the magnitude of the velocity begins to decrease. At the approximate point of intake valve closure, the mean velocity can be seen to tend to zero. High velocities that generate the highest levels of RMS turbulence velocity dominate the intake phase. During the latter stages of compression, the turbulence decays to an approximate level that is maintained up to TDC. Also during compression, the bulk flow patterns resulting from the intake processes are modified by the changing geometry and broken down into smaller eddies. After TDC both the mean velocities and turbulence intensity decay to a minimum at approximately EVO. During the expansion stroke the piston is pulled downwards by the DC motor and the in-cylinder gas decompresses. The in-cylinder pressure at the point of EVO is below that of the atmospheric pressure in the exhaust port and so a reverse flow of gas into the cylinder occurs that precedes the conventional exhaust flow out. A peak in the measured mean velocity is observed in the opposing direction that generates velocity fluctuations as indicated by the RMS velocity curve.

This is an effect observed in motored engines and especially in optical engines where piston ring blow-by during compression can be significant when piston ring wear is high. Towards BDC of the expansion stroke, the pressures are equalised and no substantial bulk flow is apparent. As the piston begins the exhaust stroke, air is forced from the cylinder and turbulently reacts with the air that has just entered. At exhaust valve maximum lift conditions, the bulk flow has stabilised once more, RMS turbulence decreases and the chamber empties of gas.

The fourth plot in Figure 4.9. shows the number of individual velocity measurements that occurred in each of the 500 crank angle time bins used in the ensemble-averaging process. During the intake phase, where the velocity is highest, the number of particles crossing the measuring probe volume is highest. The piston then passes through BDC intake where the volume is at a maximum and then to IVC where no more seeded air will enter the cylinder. During this phase the particle counts are significantly reduced, reaching a local minima at IVC. As the compression stroke proceeds, the particle counts increase steadily to a second maxima near TDC firing. The pattern is then repeated during the expansion stroke but in the exhaust stroke, the counts remain constant as seeded particles are ejected from the cylinder at a rate that is approximately proportional to the decrease in in-cylinder volume. As such, the data set here used to illustrate the engine cycle, does not contain enough data points to be able to carry out a rigorous statistical analysis. It should be noted also that these flow characteristics, although exhibiting features typical to in-cylinder sink flows measured in the pent-roof and in the positive tumble direction, cannot be used to generally describe flow patterns at all in-cylinder locations.
4.3.3. Temporal Characteristics of Instantaneous Velocity, Ensemble-Averaged Mean and RMS Velocity across the Engine Cycle

The temporal evolution of the ensemble-averaged mean and RMS velocities for the SW, SP and T-series measurement locations are given in Appendices A, B and C respectively. The measurement locations and corresponding figure numbers are summarised in Table 4.4. The above list of figures is incomplete as regards measurement locations defined by the grid outlined in the above sections. This was due to poor data rate or overloading of the PMT that required certain result sets to be rejected. For example, Figures A4. and B3. are included to illustrate such effects and were not used in the subsequent data analyses.

In the first series of figures, the instantaneous and ensemble-averaged mean and RMS velocities are plotted against crank angle over the complete engine cycle. In addition, a smoothing function has been applied to the ensemble-averaged data to reduce 'noise' and discontinuities in the averaged plots. The choice of and effect of applying a smoothing function is discussed in the following Section 4.4.3. The final set of T-series figures shows plots of smoothed ensemble-averaged data. In all cases, the method of data analysis and smoothing are identical for all measurement locations.

4.3.3.1. Intake Stroke, SW and SP Measurement Locations in the Pent-roof

The Figures A1. to A9. and B1. to B4. relate to the SW and SP point locations in the pent roof chamber respectively. It is useful to refer to Figure 4.4. for the exact locations of the measurements. The sets can be grouped by location. In the first region, SW1, 2, 3, 8, 9, 11 and SP1, 2, 4 are positioned on the intake side or centrally in the pent-roof chamber and in proximity to the intake valve. In the second region, SW5 and SW6 are in the exhaust side of the chamber. The third region includes all T-series measurements in the swept volume from 10 mm below the gas face to a depth of 70 mm.

As the valve begins to open, air is rapidly drawn into the cylinder in a highly turbulent manner producing an initially negative velocity component in SW1, 2 and 3 and SP1 and 2. SW8 and SW11 however have an initially positive (towards the intake side) mean velocity component. Due to their central locations, the air that has been drawn in through the closing exhaust valve initially affects these locations. This is however not evident in SP2 which is slightly above SW11 and towards the exhaust side and remains negative throughout the intake stroke. After the exhaust valve closes and the intake begins to lift significantly, both SW8 and SW11 points reverse direction and exhibit mean velocities in the direction of the jet inflow. SP4 at 4.2 mm above the gas face shows a significant drop in mean velocity at this crank angle and then fluctuates from zero to between -1 to -3 ms⁻¹ for much of the intake stroke.

A similar characteristic is observed at SP1 from 90 CA. The intake generated jet flow is abruptly deflected at SP1 by the apex of the pent-roof and is forced vertically downwards and horizontally inwards and outwards. The outward jet is then bounded by the cylinder wall and the inward jet then meets an opposing jet from the other valve. This location has by far the highest RMS turbulence velocity amongst all the measurement points in the cylinder.
The results show that SW1 experiences a horizontal velocity component of magnitude greater than twice that of location SW2 and more than four times that of location SW3 during the early intake phase.

The velocity magnitude at SW1 was of the order of 10 times the mean piston speed at 500 rpm. Conversely, SW3 reports a significantly higher RMS velocity than SW2 that is greater again than SW1. This suggests that location SW3, directly underneath the valve orifice area is situated in a region of highly fluctuating and recirculating flow with substantial out of field velocity components. SW2 located beneath the flat of the valve also experiences the same phenomena but to a lesser degree. The mean velocity at SW1 is relatively large and in the direction towards the exhaust valves. The velocity measured at this point in the jet flow is at a maximum at peak valve lift where the RMS value drops to a local minima. However, SW2 shows a positive mean velocity at the same instant suggesting that a recirculatory flow has been established behind the valve curtain. This is also evident slightly earlier in SW3 but the mean flow reverses as the valve begins to lift significantly. The low velocity magnitudes recorded at location SW3 (intake valve wall side) and SP4 (intake valve centre side) suggests that the major velocity component in this region is vertically downwards from the inclined port orifice. This is also observed to a lesser extent at SP2, which has more of a horizontal velocity component due to its location relative to the valve seat.

The data derived from locations SW8 and SW9 are difficult to analyse in this context due to the valve passing through SW9 and partially obscuring SW8. In addition, the three-dimensional nature of the flows (four valves) and the measures along the mid-cylinder plane (between valve sets) means that out of plane components can have a significant effect on measurements made of one single velocity component.

On the other side of the pent-roof chamber, at location SW6, the magnitude of the mean velocity component in the horizontal direction is relatively small, but the direction is towards the intake side up until intake valve closure. This suggests that the intake jet flow that passes above the valve and through SW11 (indicated by the negative velocity magnitude during this phase) impacts with the far cylinder wall (with the angle of the pent-roof) and is deflected downwards and slightly inwards thereafter. This effect is evident in SW5 also during the early stages of valve lift but as the jet becomes established through SP2 and SW1, 2 and 8, the velocity magnitude is reduced through collision and mixing of the two bulk vortical structures. This is illustrated by the relatively high RMS velocity values recorded up to 90 CA at SW5.

4.3.3.2. Compression Stroke SW and SP Measurement Locations in the Pent-roof

At the three locations SW1, 2 and 3, the mean flow and RMS turbulence characteristics are similar from intake valve closure to mid-compression stroke. As the piston approaches TDC, SW2 and SW3 show minimal bulk mean motion. SW1 however exhibits mean motion in the direction of the exhaust side up to and beyond TDC with a linear reduction in velocity magnitude. In SW1, SW2 and SW11, the RMS velocity at TDC is of a comparable magnitude to the ensemble-averaged mean velocity. The SW3 location shows a marked increase in RMS value due to a rapid change in mean velocity direction at 10 CA BTDC.
This is not thought to be a physical effect of the flow motion but that due to the proximity of the piston to the measurement location, the mirror angle and the refraction and reflection from the pent-roof wall that obscures the back-scatter signal.

For most of the measurement locations, the bulk mean velocities generated in the intake stroke decay during compression up to approximately 10CA BTDC. From this point to TDC, there is a small local increase in mean and RMS velocity. This can be observed in SW2, 3, 4, 5, 6 and 11, as well as SP1 and 2. At all but the SP1 point, the change in mean horizontal velocity component magnitude between three-quarter compression stroke and 5 CA BTDC is an increase of approximately 50%. The final velocity magnitudes were in the range of approximately 0.5-0.7 ms\(^{-1}\), or approximately 0.25 times the mean piston speed, excepting the centrally located SW 1 and 5 positions, whose final velocities were of the order 2 to 3 ms\(^{-1}\). The central location SW1, which exhibits a comparatively large velocity magnitude throughout the compression stroke, records a reduction of approximately 50% over the aforementioned range.

Figures B1., B2., B4. and B5. show the ensemble-averaged mean and RMS velocities for the SP spark plug measurement locations. Of particular interest is point SP1, located at 0.8 mm depth from the apex of the pent-roof and co-incident with the spark plug electrode. At 40 CA before TDC the RMS velocity begins to ramp up significantly until 10-15 CA later, it has reached a value that is 4 times that at 40 CA BTDC. In this region, spark ignition would be initiated. Over the same period, the mean velocity at the same point had reduced by 2 ms\(^{-1}\). At TDC however, the horizontal velocity component is still significantly large at -6.5 ms\(^{-1}\). The RMS velocity at the same instant is approximately 2.5 times this value. At SP2, 2 mm below SP1, the mean velocity has dropped to -1 ms\(^{-1}\) and the RMS velocity approximately 2.5 times this value again. The location at SP4 shows a rapid acceleration of the horizontal velocity component as the piston approaches TDC. A change in velocity magnitude of this order is not generally observed at TDC and this portion of the ensemble-averaged data should be ignored.

4.3.3.3. Intake and Compression Strokes for the T-series Measurement Locations in the Cylinder Volume

The smoothed ensemble-averaged mean and RMS velocity data for the T-series measurements at z=-10, -20, -30, -40, -50 and -70 mm depths from the cylinder head gas face are presented in Figures C1. to C17. using the file naming convention previously outlined. A description of the general trends exhibited in the results is given below. It should be noted that in each set of data, there is a period where the piston passes through the measurement location. As the z-depth increases the amount of measurable engine cycle is reduced. This is indicated in the plots by discontinuities in the mean and RMS curves as zero velocity is recorded. As such TDC non-firing and TDC fired cannot be measured at any of these locations. The data from the T-series results are more adequately characterised in the following section that shows the reconstruction of the full velocity field for the horizontal measured component relative to the instantaneous piston position.

In the first instance, the experimental results were analysed for each respective z-depth at a fixed y displacement from the mid-cylinder axis. The TP00P00 data points lie along this axis. The mean velocity at each depth observes a similar characteristic after the initial piston obscuration.
At z=-10, 20 and 30 mm, a positive velocity component is observed up to approximately 90 CA in the intake stroke and just prior to peak valve lift. Following this point, there is a sharp velocity reversal and a peak in RMS velocity value. The magnitude of the velocity reversal is comparable in either direction. The crank angle of the flow reversal occurrence increases with increasing z-depth. At z=-10 mm the reversal occurs at 90 CA; at z=-20 mm, 100 CA and at z=-30 mm, 115 CA. This suggests that during the early stages of valve lift, the predominant jet flow is over the top of the intake valve and towards the exhaust valve.

The flow then follows the combustion chamber roof cylinder wall boundaries and is driven downwards by the motion of the piston and along its flat surface. This motion can be described as forward tumble. As maximum valve lift is approached, the jet flow from below the valve, trapped against the cylinder wall begins to attain the same order of magnitude. The flow exits from below the lifting valve in a fan shaped wedge formed between the valve lip and bounding intake cylinder wall. It then proceeds to descend the intake sidewall to meet the forward tumble flow. The colliding flows are manifested as an increase in RMS velocity late in the intake stroke and towards intake valve closure. At z=-10 and z=-20 mm, the RMS peak is that due to the intake jet and mixing in the valve regions. At z=-30 mm, the same sharp RMS peak is evident but now it additionally spans at least twice the number of crank angles, past intake valve closure and into the compression stroke where 2 ms\(^{-1}\) was measured and zero mean velocity. At z=-50 and z=-70 mm the mean velocity is that due solely to the forward tumble flow but the RMS spread is still evident into the compression stroke for z=-50 mm.

The characteristic flow patterns described at TPOOPOO locations can be corroborated by examination of the positive and negative halves of the tumble plane. For the TPOOPOO positions, the point of flow reversal was approximately 80 to 90 CA ATDC. At this point, the intake side valve jet was said to predominate. In the positive y=15 mm region the flow accelerates towards the exhaust side from several ms\(^{-1}\) to approximately 10 ms\(^{-1}\) during the same crank angle period. No positive velocity components are measured at z=-30 mm. It is not until a depth of z=-50 mm that the positive forward tumble motion is recorded at y=15 and 30 mm.

In the negative half of the tumble plane, the flow characteristics at z=-10, 20 and 30 mm and y=-15 mm are very similar to those observed along the mid-cylinder axis. Again, the crank angle at which the flow reversal occurs varies with z depth. At z=-10 mm it occurs at approximately 80 CA; at z=-20 mm, 100 CA and at z=-30 mm, 130 CA. At z=-50 and -70 mm the flow is predominately that due to the forward tumble motion. At all y=-15 locations, the ensemble-averaged mean velocity peak intake values are greater than those recorded along the mid-cylinder axis. At z=-30 mm, the peak value is three times that recorded at TPOOPOO but the RMS velocity is approximately one half of the value. This confirms that the mid-cylinder axis position is affected by its location between the two intake valves and the three-dimensional competing flows. The y=-15 mm region is contained in a more ordered jet region of the intake bulk flows.

At y=-30 mm, the measured velocity component at z=-10 mm depth is relatively high and directed towards the exhaust side during the intake valve opening phase. During intake valve closure, the mean velocity rapidly decays to a zero mean value. At z=-20 and -30 mm, the velocity has reversed and is recorded in the positive direction. This would indicate a local region of flow recirculation, outside the 'view' of the glass window constraints.
The relatively high RMS velocities at z=10 and z=20 mm decrease by over three times at depth z=30 mm but the peak velocities remain comparable.

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

The ensemble-averaged mean velocity data is presented for all the in-cylinder measurement locations during the intake and compression strokes at fixed time (CA) intervals. The presentation of the data in this manner provides a visual representation of the mean velocity flows in the horizontal (tumble) direction. The data is presented in both vector and contour plot form. As some of the measurement locations provided insufficient or 'bad' data, an interpolation routine was used to construct a rectangular grid that mapped the entire cylinder swept volume. The method of velocity field construction developed is illustrated in Figures 4.10a., b. and c. and can be summarised in five steps outlined below:

4.3.4.1. Step 1: Measured Mean Velocity Spatial Field Construction for a Fixed Crank Angle

The first step involved sorting the data at each spatial location for a fixed crank angle. Figure 4.10a. shows the LDA measurement point locations with the piston at BDC to show the complete measurement field. A typical vector plot at these locations for 120 CA is shown adjacently. It should be noted that some points have a zero mean velocity and are therefore not represented by a velocity vector. Also, the x-axis full-scale length is the bore of the cylinder. The viewable section through the glass piston is indicated by the dotted line.

4.3.4.2. Step 2: Definition of a Grid Field

Secondly, a rectangular grid was constructed over the plane of measurement to be used as a data field for the contour plots. Only three grid locations fitted into the pent-roof area. The number of nodes in the grid across the cylinder was equivalent to the number of LDA points. The number of nodes below the gas face was twice that of the number of LDA measurement locations. The grid filled in those points that had no measurement data attached to them. The grid size was 10 mm by 5 mm and extended to the 30 mm window boundaries from the cylinder axis. The grid was selected in this way to ensure that the boundary values of the contour plot were those due to the true experimental values where possible at those points and crank angles. The grid size used was relatively coarse to mirror the spread of the LDA point locations and within the limit of the study to visualisation of the bulk flow patterns. The data of this preliminary study was not considered adequately complete (or refined) to be applicable to the identification of smaller scale flow structures. The final rectangular grid, including the SP, SW and T-series points is shown in Figure 4.10.b. with the piston at BDC.

4.3.4.3. Step 3: Interpolation of LDA Measurements to Grid Field Nodes

An interpolation scheme was then applied in the horizontal and vertical directions between the velocity measures at the LDA point locations and the destination nodes on the superimposed grid.
A typical interpolated contour plot and vector plot of the horizontal velocity component is given in Figure 4.10c. Several types of interpolations were considered. Amongst those were a simple linear interpolation routine and an inverse distance routine. In each case, the boundaries (defined by the LDA measures) were kept constant. In the cases where a boundary value was not available at a particular crank angle (generally in the upper intake positive quadrant of the cylinder), the assumption of a zero magnitude component of horizontal velocity was used. The assumption was made through observation and comparison with PIV data performed on the same cylinder head as presented in Figures 4.11a to d. For the intake stroke, this compared favourably with PIV data performed in a dynamic flow visualisation rig using water and Reynolds analogy (raw data extracted from Pommier, (2000)).

Figures 4.11a. shows the PIV vector map with both velocity vector components and the velocity field with just the horizontal component at 50 CA. In the two-dimensional vector fields produced by this method, the velocity component in the uppermost wall side jet region of the intake flow was dominated by the vertical component and the magnitude of the horizontal component can be regarded as an insignificant contribution to the velocity magnitude. The single component PIV vector maps for the rest of the intake stroke are presented in Figures 4.11b to d. For the entire intake stroke, the horizontal component in the upper intake side quadrant of the cylinder is close to zero. It should be noted that the assumption of a near zero horizontal velocity component is not valid in the other region of 'bad' data at z= −50 and -70 mm and y=−30 mm from approximately 130 CA to BDC as shown in Figures 4.11c and d. During the compression stroke the global mean velocity is very low in comparison to the intake stroke and it is not until the piston approaches TDC that significant velocities are induced within the flow. At this point, it is the flow in the pent-roof that is studied and all LDA point values are known.

The linear interpolation calculated the value for the grid node based upon its location within an individual LDA measurement point cell. The value at the LDA point was linearly interpolated to the grid data point within the cell. The inverse distance interpolation performed the same routine but weighted the individual values in the grid by the distance between LDA point and grid node. In this case, the weighting was performed between the nearest eight possible (co-ordinate system octants) LDA data points to the destination grid node rather than a single closest point or all the data points. This was preferred due to the coarse nature of the grid relative to the total dimensions of the cylinder volume. The weighting value increased with decreasing distance between LDA and node points. The weighting function is defined as the inverse of the distance multiplied by an exponent. The choice of exponent was made by comparison of the magnitude of the interpolated velocity vectors at 5mm each side of several of the LDA data points not included in the contouring grid; namely those at the y=±15 mm points from mid-cylinder. The exponent was then selected so that the LDA value was the average of the two values either side of its location in the horizontal direction. A value of 3.5 was selected. A comparison of the two interpolation techniques showed that similar results were achieved by both methods. However, for areas of the grid that had no LDA data point for interpolation at a particular crank angle, the inverse distance technique produced a more intuitive approximate value based upon the other neighbouring measured LDA data. The resulting inverse-distance interpolation performed upon the vector field in Figure 4.10.a is shown in Figure 4.10.b.
A qualitative comparison between an averaged PIV vector map and the interpolated LDA data at BDC is presented in Figure 4.12, along with a LASER illuminated flow visualisation frame taken from the Dynamic Flow Visualisation Rig (DFVR) at the University of Brighton (Pommier, 2000). The visualisation image has been rotated 90° in the clockwise direction to the LDA frame.

4.3.4.4. Step 4: Construction of Flooded Contour Plot for Velocity Field Visualisation

The final interpolated velocity field reconstruction was then converted into a contour plot with the velocity magnitude selected as the contouring variable. This is shown in Figure 4.10c. The range of the contouring levels was selected to include the maximum and minimum values in the data set for that particular crank angle and are therefore not comparable from image to image. The number of contouring levels governs the amount of colour levels between velocity magnitude maximum and minimum values. As the range of measured values changed significantly between the early intake and compression strokes, a different level step (velocity increment) was selected for each crank angle and the number of contour levels varied over a small range. The higher the number of contour levels, the greater the apparent smoothing effect between colours. The choice of number of contour levels is therefore arbitrary and serves only to enhance visualisation of the data. The number of contour levels was chosen in the range of between 50 and 100. The generated contour plots for the intake and compression strokes are shown in Appendix D.

4.3.4.5. Step 5: Extraction of Velocity Scalar Profiles across Cylinder Bore

The interpolated mesh generated in the above step was used to extract velocity scalar profiles at constant crank angles along vertical lines for varying y-displacements across the cylinder bore. This step is shown in Figure 4.10c. For each vertical line, a maximum of 25 data points were extracted from the interpolated field of the velocity magnitude. Towards the edge of the cylinder, 20 points were extracted. The location of the lines and number of points was chosen to allow a simple visual comparison with PIV measurement data performed over the same plane during the intake stroke. Once the interpolated field had been created it is possible to extract velocity magnitude data along any line within the bounds of the 2-dimensional field. The complete data set is presented in Appendix E for the intake and compression strokes.

4.3.5. Comparison with PIV Mean Velocity Vector Plots

The spatial distribution of the ensemble-averaged mean interpolated velocity field was qualitatively compared with PIV vector maps to assess the validity of the method proposed and the assumptions stated in the previous section. The PIV data was derived from averaging ten vector maps measured during the intake stroke in a dynamic water simulation rig. A cross-correlation method was used between consecutive LASER illuminated digital images in a given plane, to resolve the direction and velocity magnitude of seeding particles artificially added to the flow. The engine cylinder head, bore, stroke, piston speed and valve lift characteristics were identical to that used in the motored LDA study.
Details of this set-up, dynamic similarity, experimental method and results analysis is presented comprehensively in the published literature and in the theses of Faure, (1997) and Pommier, (2000). The results are presented in Figures 4.13.a to c. for the spatially interpolated LDA velocity scalar profiles and the two-dimensional PIV vector map of mean velocity. In addition the horizontal component of the mean PIV velocity plot is included to aid direct comparison. For this case, the component values have been magnified to aid visualisation between the plots.

Observation of the plots indicates that there is a good agreement between the bulk flow motions measured using the PIV technique and the interpolated velocity field derived from a much coarser grid of LDA measurements. This appears to be the case for the early and mid-stages of the intake stroke. In the latter stage of the stroke, the PIV vector maps indicate a relatively low velocity across the piston from the intake to exhaust side. This is not mirrored in the LDA velocity values have showed a relatively small magnitude in the direction from exhaust to intake side. At these points, the measured velocity values are very low and as previously stated, the assumption of zero velocity at the boundary for the unknown values at these crank angles in the LDA measures was not valid.

As a result of these constraints, the interpolated results in this region have tended to zero or towards very small velocity values weighted by too few or distant LDA data points. The complex three-dimensional nature of the intake generated flow pattern is shown in Figure 4.14. for the mid-cylinder, cross-tumble plane during the intake stroke from 50 to 150 CA. In particular, these results explain the unsteadiness observed in the ensemble-averaged mean and RMS velocities of the previous sections in the mid-cylinder SP, SW and to a lesser extent, T-series measurements. This region is subjected to colliding and competing eddies, exiting from around the complete valve lip diameters. The PIV results show large areas of flow re-circulation outwards and upwards into the pent-roof region, adding to the velocity fluctuations. Therefore these significant out of plane flow motions will contribute to the measured RMS velocity in the tumble plane manifested as intense fluctuations in the measured instantaneous measured velocity.

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

4.4.1. Experimental Uncertainties

The assessment of the contribution of experimental errors in the LDA measurements performed can be divided into the two categories extensively covered in Chapter 3. The first relates to the systematic errors and the second, to the statistical errors introduced through estimation of the flow characteristics from a limited number of samples. The latter category is discussed below in Section 4.4.2. Uncertainties due to velocity bias and crank angle broadening are considered in Chapter 5, where the data quality is considered suitable for such analyses of the smaller scale flow structures. In almost all cases of engine LDA measurements, it is the uncertainty due to the limiting number of velocity samples in a crank angle window that is the significant factor in assessing experimental error. The main errors in the experimental set-up and method can be summarised:

(a) Beam Intersection Position. The initial experimental set-up relied upon the beam targeting method described in Section 4.2.1.2.
The method of finding an anode current 'peak plateau' at an intersection point on the target grid surface was sensitive to within two or three turns of the traverse lead screw. This was equivalent to ± 0.3 mm of the surface location. The traverse routines were returned to the zero datum after each set of measures as a check and showed excellent repeatability. The manufacturers estimate for the accuracy of the rectilinear traverse position is ± 0.1 mm. The rotational vernier used to set the angle of the mirror was accurate to ± 0.5°.

Using these estimates and a horizontal beam pair of 400 mm focal length, the greatest positional error in the z-depth direction was ± 0.2 mm and ± 1.75 mm in the x-direction. The error was minimised by careful selection of a small range of mirror angles centred about 45° to the horizontal. The error in location of the beam pair in the y-direction was that due to the positional error of the traverse along this axis. The error due to the small out-of-plane mirror surface angle was considered negligible. Therefore, the maximum error in beam positioning is that due to the combination of targeting, mirror angle and traversing. The maximum values in the x-direction were ± 2 mm, in the y-direction, ±0.4 mm and in the z-direction, ± 0.5 mm.

(b) Optical Set-up. The experimental study of Boutrif and Thelliez (1993) estimated the error in the LASER wavelength to be 0.2% and the error in beam separation to be 4%. Under these conditions, Boutrif and Thelliez (1993) stated that the velocity measurement errors attributable to fringe errors through optical set-up were always less than 5%. Riahi and Hill (1993) estimated the error in the calculated fringe spacing to be approximately 2%, which gave a maximum error of 4.2% in their dimensionless velocity data. They stated that the estimated error in the determination of the Doppler frequency by the counter type processor was 0.25 %. The correlation type processor used in these studies has a much higher resolution and uses an internal clock timer that increments at a frequency of 1 MHz and with a -40 MHz frequency shift and therefore the error in frequency determination can be assumed to be considerably smaller than that of the older type processors.

(c) Engine Speed. A high-resolution crank angle encoder was used for the trigger and timing input for the LDA processor. The encoder produced 1000 pulses per revolution. This ensured that the LDA measure was always related to a true crank angle position even when the engine speed fluctuated through the cycle. The crank angle encoder was not used as the engine speed control. Instead, a belt-driven tachogenerator on the output of the dynamometer was used in a feedback control loop with the dynamometer controller. The controller had a potentiometer input selectable by the operator. The error in the engine speed was assessed by the use of an optical timing gun positioned against the engine flywheel and secondly, by the use of a frequency counter connected to the flywheel pick-up. Both methods showed good agreement with the optical shaft encoder and were plotted against the 10-turn dynamometer speed potentiometer position for a range of ten engine speeds. A linear curve fit produced a scaling factor for the potentiometer position. The smallest selectable scale on the potentiometer was 1/100 of a full turn. This was equivalent to approximately ±5 rpm at engine speeds above 500 rpm.

4.4.2. Statistical Accuracy of the Mean and RMS Velocity Estimates

The equations that describe the standard error in the mean and RMS estimates are given in Section 4.2.1.5.
The standard error is used here because the ensemble-averaged value in a finite crank angle window is considered to be a statistical estimate made from small samples of a much larger population. That is, theoretically, a population derived from an infinite number of measures in each time bin over an infinite number of engine cycles. For a random process, the population distribution in each window is assumed to be normally distributed. A sampling error is therefore introduced due to the difference in results based upon the sample with those of the total population.

The standard deviation of the sample is the standard error (of a particular distribution) and can be used as a measure of the fluctuation in properties (mean and standard deviation) from one sample to the next. If the population is normally distributed, the sampling distribution of the mean is also normal and equal to the population mean. The standard deviation is equal to the standard deviation of the population divided by the square root of the sample size. Diamantopoulos and Schlegelmilch (1997) state that for sample sizes greater than 30, the standard deviation of the population in the above expression can be substituted with the standard deviation of the sample. They go on to explain that if the sampling distribution of the mean can be approximated by a theoretical normal distribution, the form of the distribution in the population is of limited concern.

Figures 4.15a, b, c and d. show the standard error (standard deviation of the sample) in the ensemble-averaged mean and RMS velocity estimates for a relatively small data set over the four strokes. For certain crank angles, the LDA data is discontinuous. The average validated data rate over all engine cycles was approximately 0.2 kHz, which includes periods in the cycle where the piston intersected the probe volume. The total data collected was approximately 22,000 validated data points over approximately 1000 cycles at 500 rpm.

The standard error, \( \sigma \) in each crank angle window is plotted along with the ensemble-averaged mean data. The error bars on the mean data are the \( \pm 3\sigma \) (three times standard error) indicators. The probability of the true value being within these error bands is approximately 99.5 \%. For clarity, only every second error bar is shown on the ensemble-averaged data. The results show clearly that during the phases of the cycle with high RMS velocity, the standard error bars can attain very large values that would suggest that the ensemble-averaged estimate is likely to be a poor representation of the true mean velocity.

This is particularly evident during the early intake (approximately 20-130 CA in Figure 4.15a.) and during the exhaust blow down at approximately 500 CA in Figure 4.15c. However, during the compression stroke, where the data rate is high and the turbulence intensity is relatively low, the \( 3\sigma \) standard error was within the range of 1-3 ms\(^{-1}\). The value at TDC is however five times that reported by Hilton, (1991), who recorded 0.2 ms\(^{-1}\) in a similar combustion system. Elsewhere in less seeded parts of the cycle, Hilton, (1991) reported \( 3\sigma \) values in the range of 1 ms\(^{-1}\). In those experiments, at least ten times the number of data points was collected. The standard error in the RMS velocity estimate is also large during the intake period and even into the compression stroke. At some crank angles, the \( 3\sigma \) bands are twice that of the measured value, suggesting regions where very few (clearly less than 30 samples per crank angle window) were collected. It should be noted, that some of the peaks are found a regions in the cycle where the piston intersected the probe volume.
In this preliminary study, there is insufficient data numbers across the engine cycle, distributed within each 1.44° crank angle interval to be able to draw satisfactory conclusions about the statistical accuracy of the velocity estimates other than in the most heavily seeded and less turbulent mid-to-late compression stroke. The estimate of the mean flow (as presented in the previous Sections of the Chapter) is statistically more accurate than that of the RMS velocity fluctuation.

From the analysis of the errors, it can be concluded that a minimum of at least 30 samples is required per crank angle although 100 samples is oft cited as a general figure in the literature. For 500 crank angle windows and assuming even distribution of velocity values, this would equate to a minimum of 50,000 validated Doppler events. The measurement point selected has approximately half the number of validated data points of the ‘best’ point and twice that of the smallest data set. A more rigorous analysis of the standard error would need to include all the measurement points. In addition, the velocity probability density function within an individual crank angle window would need to be calculated to assess whether any significant deviation from a Normal distribution was evident. In some studies data is erroneously presented over larger crank angle intervals of 10 or more crank angles. These are commonly centred around the point of spark ignition.

Over these intervals, the sample size would be of the order of 7 times that of the present study and well within the 100 samples per window range. However, the effect of crank angle broadening (velocity gradient over the interval), especially evident in high velocity regions of the cycle, requires particular analysis. The interval of 1.44 CA used in this study was considered to be the largest acceptable interval and the effect of a velocity gradient is studied in Chapter 5.

4.4.3. Effect of Smoothing Ensemble-Averaged Data

Data smoothing of fluctuating or ‘noisy’ data for visual purposes is often subjective and can mask important characteristics in the flowfield. The aim therefore is to find a smoothing routine that follows the ensemble-averaged plot sufficiently closely during regions of fluctuation (e.g. the intake stroke) whilst removing the spurious sharp peaks and troughs superimposed upon the mean characteristics. Several smoothing algorithms were tested on the ensemble-averaged mean velocity data in an attempt to remove the small-scale ‘noise’ and discontinuities caused by intermittent breaks in the LDA data acquisition or too few samples recorded within an individual crank angle bin. Simplistically, each smoothing operation shifts the velocity value at that point towards an average of the adjacent values on either side.

Figure 4.16. shows 6 different data smoothing approaches applied to a typical ensemble-averaged data set acquired during the compression stroke. This set was chosen because it contains both discontinuities and varying degrees of fluctuation in the mean profile. In each case, the effect of one of the three smoothing parameters is varied in isolation. The first parameter refers to the number of times the smoothing function is to be performed. This is illustrated in the first plot where a single pass is compared to 10 passes of the same smoothing routine. Both routines show similar trends during parts of the cycle where the ensemble-averaged mean fluctuates minimally. However, in more intensely fluctuating regions, the 10-pass plot over dampens the mean profile. As such, the single pass was considered adequate in this case.
The smoothing coefficient or relaxation factor was the second parameter investigated. The value was varied from between 0.1 and 0.95. The larger the number, the greater the amount of smoothing per pass. These are plotted in the second graph for a single pass. The smoothed function is indistinguishable at this scale from the ensemble-averaged plot for a 0.1 coefficient. Conversely, the 0.95 coefficient weights the average value between neighbouring points towards the first, left-hand point used in the averaging procedure. In regions of fluctuation, the curve appears to be phase shifted to that of the ensemble-averaged mean. In all cases, a coefficient of 0.5 was considered best placed between the two extremes.

The final parameter investigated is illustrated in the third plot of Figure 4.16. This relates to the choice of adjacent data points to use for the averaging procedure. In the first instance, the boundary points are fixed and the average is calculated as normal. In the second and third cases, the points are smoothed based upon the assumption that the first or second derivative normal to the boundary is constant. The results for both these types of boundary conditions are very similar at these scales. Both these methods follow sharp fluctuations in the curve particularly well and as such produce peaks and troughs of their own because of the required boundary conditions.

The final choice of smoothing routine, used for the presentation of the ensemble-averaged mean and RMS velocity estimates, was selected as the average between fixed boundaries points with a relaxation coefficient of 0.5. This was performed over a single pass.
4.5. Conclusions of Chapter 4

The characteristics of the airflow in a modern, side-entry, MPI, pent-roof combustion chamber with a flat-top piston have been established as a means to predicting a baseline for comparison with similar measurements performed in other combustion systems. The results described the temporal evolution of the mean and RMS velocities over the entire engine cycle and the reconstructed mean spatial velocity field during the intake and compression strokes. A method for the reconstruction of the spatial mean velocity field showed good agreement with mean vector maps measured using PIV in a Dynamic Flow Visualisation Rig (DFVR) with the same engine geometry.

The salient features of a typical, motored, engine cycles' instantaneous, mean and RMS velocities as well as validated LDA data counts per crank angle interval are presented. The results show that during the intake phase of the engine cycle, a bulk flow pattern is established both above and below the intake valve in the mid-cylinder tumble plane. Initially, the majority of the flow is over the top of the intake valves and towards the exhaust ports. The jet flows follow the combustion chamber roof and descend the cylinder wall and across the piston surface. At peak valve lift, the jet flow below the valves becomes more significant and starts to attain the same order of magnitude. This flow descends the intake side of the chamber and meets the established forward tumble pattern. This is observed as an increase in measured RMS velocity at the point of collision, lower down in the cylinder and at approximately IVC. The peak intake mean and RMS velocities were observed during the period between maximum intake valve lift and maximum instantaneous piston speed. Local regions of flow re-circulation are observed behind the valve edges. The centre of the pent-roof chamber below the spark plug location also shows high levels of turbulence generation due to out of plane flow interactions between the competing intake jets. These results were corroborated by PIV vector maps performed in the cross-tumble plane at the same in-cylinder locations. At the measurement points in the regions of the jet flows, the flow is more ordered and the RMS fluctuation reaches a local minimum at peak valve lift.

Towards the end of the intake stroke, a forward tumbling flow pattern fills the swept volume of the cylinder. This pattern is then substantially modified during the compression stroke. However, the RMS turbulent velocity component decreases during the early compression stroke and maintains a constant magnitude up to TDC. At the base of the pent-roof, the mean velocity component is of a comparable magnitude to the RMS velocity at TDC. Within the apex of the pent-roof chamber, the RMS fluctuation increases. The bulk motions decay until approximately 10 CA BTDC. After this point, the mean and RMS velocities both increase marginally. The spark plug gap measurement location at 0.8 mm from the apex of the chamber shows increasing turbulence intensity at the point of a typical spark ignition timing. At TDC, there is a 50% increase in the mean horizontal velocity component at the same location. Generally, the mean velocity component is approximately equal to 0.25 times the mean piston speed at TDC.

The study has also established the operating criteria, quantified the experimental uncertainties and set the limitations of the experimental LDA technique. Spurious results and data processing techniques were investigated. The standard error in the sampling of the mean velocity results was greatest during the intake stroke. This highlighted the sensitivity of the technique to particle seeding and data density. It was seen that in this preliminary study, there were too few validated data in some of the crank angle intervals to infer conclusions about the smaller scale flow structures. The results therefore present the characteristics of the mean velocity flowfields.
Figure 4.0. Single Cylinder of Multi-cylinder, Side-Entry, Manifold Injection Gasoline Volvo B230 Engine

Figure 4.1. Four-stroke Valve Timing Diagram for Side-Entry, Manifold Injection Gasoline Engine Build
Figure 4.2. Schematic of 'Tipping' Mirror LDA Set-up
In an Optical Hydra Engine
Figure 4.3. Photographs of 'Tipping' Mirror LDA Set-up in an Optical Hydra Engine with Volvo B230 Cylinder Head (above) and 'Scanning' Mirror LDA Set-up (below)
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Figure 4.5. Automated Traverse and Optical Air Breadboard for LDA Transmission Optic
Figure 4.6a. Isometric Schematic of LASER Transmission Optic, 'Tipping' Mirror and Etched Target Plate

Figure 4.6b. Side View Schematic of LASER Beams, 'Tipping' Mirror and Etched Target Plate
Figure 4.6c. Plan View Schematic of LASER Beams, 'Tipping' Mirror and Etched Target Plate
Figure 4.7. Preliminary In-Cylinder LDA Data Sets for Motored Operation in the Volvo B230 Cylinder Head – Sweep 1 Location
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Figure 4.9. Features of a Typical Motored Engine Cycle
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Figure 4.10b. Interpolation Grid Nodes and LDA Locations

Figure 4.10c. Flooded Contour Plot and Extracted Velocity Profiles based upon Interpolated Velocity Field
Figure 4.11a. Averaged PIV Velocity Vector Maps During the Intake Stroke at 50 CA for the Volvo Cylinder Head: Vertical and Horizontal Velocity Components (above) and Extracted Horizontal Velocity Component Only (below) at Same Scale (raw data extracted from Pommier, 2000)
Figure 4.11b. Averaged PIV Velocity Vector Maps During the Intake Stroke for the Volvo Cylinder Head: Extracted Horizontal Velocity Component at 70 CA (above) and 90 CA (below) at Same Scale (raw data extracted from Pommier, 2000)
Figure 4.11c. Averaged PIV Velocity Vector Maps During the Intake Stroke for the Volvo Cylinder Head: Extracted Horizontal Velocity Component at 110 CA (above) and 130 CA (below) at Same Scale (raw data extracted from Pommier, 2000)
Figure 4.11d. Averaged PIV Velocity Vector Maps During the Intake Stroke for the Volvo Cylinder Head: Extracted Horizontal Velocity Component at 150 CA (above) and 170 CA (below) at Same Scale (raw data extracted from Pommier, 2000)
Figure 4.12. Comparison of Averaged PIV Vector Map, Re-constructed LDA Velocity Field at 120 CA and Flow Visualisation Frame for the Intake Stroke (data extracted from Pommier, 2000)
Figure 4.13a. LDA Interpolated Mean Velocity Horizontal Scalars, PIV Mean Velocity Vector Map and PIV Vector Horizontal Component Scaled-up for the Intake Stroke at 60 and 100 CA
Figure 4.13b. LDA Interpolated Mean Velocity Horizontal Scalars, PIV Mean Velocity Vector Map and PIV Vector Horizontal Component Scaled-up for the Intake Stroke at 120 and 140 CA
Figure 4.13c. LDA Interpolated Mean Velocity Horizontal Scalars, PIV Mean Velocity Vector Map and PIV Vector Horizontal Component Scaled-up for the Intake Stroke at approximately 160 and 180 CA
Figure 4.14: PIV Mean Velocity Vector Maps for the Mid-Cylinder Cross-Tumble Plane during the Intake Stroke.
(Data Extracted from Pommier, (2000)).
Figure 4.15a. Standard Error in Ensemble-Averaged Mean and RMS Velocity Estimates for the Intake Stroke
Figure 4.15b. Standard Error in Ensemble-Averaged Mean and RMS Velocity Estimates for the Compression Stroke.
Figure 4.15c. Standard Error in Ensemble-Averaged Mean and RMS Velocity Estimates for the Power Stroke
Figure 4.15d. Standard Error in Ensemble-Averaged Mean and RMS Velocity Estimates for the Exhaust Stroke
Figure 4.16. Effect of Smoothing Operators upon Ensemble-Averaged Mean Velocity Data
<table>
<thead>
<tr>
<th>Base Engine</th>
<th>Ricardo Mk1 Optical Hydra Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder Head</td>
<td>Volvo B230, 2.3 litre, DOHC, Four-stroke, Inline, Manifold Fuel Injection</td>
</tr>
<tr>
<td>No of Cylinders</td>
<td>One cylinder of four multi-cylinder</td>
</tr>
<tr>
<td>No of Valves</td>
<td>Four per cylinder</td>
</tr>
<tr>
<td>Bore</td>
<td>96.0 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>80.0 mm</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>158.0 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>10:1 Nominal</td>
</tr>
<tr>
<td>Combustion Chamber Geometry</td>
<td>Pent-roof</td>
</tr>
<tr>
<td>Piston geometry</td>
<td>Flat Top with Valve Cut-outs</td>
</tr>
<tr>
<td>Spark Plug Location</td>
<td>Central</td>
</tr>
<tr>
<td>Intake System</td>
<td>Straight, side-entry, siamesed port</td>
</tr>
<tr>
<td>Exhaust System</td>
<td>Straight, side exit, siamesed port</td>
</tr>
<tr>
<td>Direction of Air Motion</td>
<td>Forward tumble</td>
</tr>
<tr>
<td>Valve Timing</td>
<td>IVO 21° BTDC</td>
</tr>
<tr>
<td></td>
<td>IVC 59° ABDC</td>
</tr>
<tr>
<td></td>
<td>EVO 59° BBDC</td>
</tr>
<tr>
<td></td>
<td>EVC 21° ATDC</td>
</tr>
<tr>
<td>Maximum Valve Lift</td>
<td>Inlet 9.8 mm</td>
</tr>
<tr>
<td>Intake Valve Inner Seat Diameter</td>
<td>32 mm</td>
</tr>
<tr>
<td>Optical access</td>
<td>‘Through-piston’ with ‘Tipping’ mirror and quartz piston crown window</td>
</tr>
</tbody>
</table>

*Table 4.0. Manifold Injection Engine Specifications*
**Table 4.1. Automated CNC Traverse Routines for ‘Through Piston’ Measurement Grid Locations**
### LDA Optical Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Fringe spacing, μm</td>
<td>6.4363</td>
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<tr>
<td>Number of fringes</td>
<td>38</td>
</tr>
<tr>
<td>Wavelength of light, nm</td>
<td>514.5</td>
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<tr>
<td>Gaussian Beam Diameter, (1/e²), mm</td>
<td>0.27</td>
</tr>
<tr>
<td>Beam Separation, mm</td>
<td>32</td>
</tr>
<tr>
<td>Beam Focal Length, mm</td>
<td>400</td>
</tr>
<tr>
<td>Optical Frequency shift, MHz</td>
<td>-40</td>
</tr>
<tr>
<td>Probe volume dimensions, mm</td>
<td>0.2428</td>
</tr>
<tr>
<td>Probe volume dimensions, mm</td>
<td>0.2426</td>
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<tr>
<td>Probe volume dimensions, mm</td>
<td>6.0704</td>
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<tr>
<td>LASER Power, W</td>
<td>0.2-0.6</td>
</tr>
</tbody>
</table>

*Table 4.2. LDA Optical Parameters for ‘Through Piston’ Pent-roof Gasoline Engine Measurements*

<table>
<thead>
<tr>
<th>Hardware Filter Bandwidth (MHz)</th>
<th>Velocity Range (ms⁻¹)</th>
<th>Lower limit</th>
<th>Upper limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.12</td>
<td>-0.39</td>
<td>+0.39</td>
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</tr>
<tr>
<td>0.40</td>
<td>-1.29</td>
<td>+1.29</td>
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<td>1.20</td>
<td>-3.86</td>
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<td>4.00</td>
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<td>+12.87</td>
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<tr>
<td>12.00</td>
<td>-38.60</td>
<td>+38.62</td>
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<td>45.00</td>
<td>-64.30</td>
<td>+225.2</td>
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*Table 4.3. Hardware Filter Bandwidth and Corresponding Velocity Ranges for LDA Optical Configuration with 40 MHz Frequency Shift*
<table>
<thead>
<tr>
<th>Figure No.</th>
<th>Location No.</th>
<th>Figure No.</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>SW1</td>
<td>B1</td>
<td>SP1</td>
</tr>
<tr>
<td>A2</td>
<td>SW2</td>
<td>B2</td>
<td>SP2</td>
</tr>
<tr>
<td>A3</td>
<td>SW3</td>
<td>B3</td>
<td>SP3</td>
</tr>
<tr>
<td>A4</td>
<td>SW4</td>
<td>B4</td>
<td>SP4</td>
</tr>
<tr>
<td>A5</td>
<td>SW5</td>
<td>B5</td>
<td>SP1, 2, 3 and 4</td>
</tr>
<tr>
<td>A6</td>
<td>SW6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A7</td>
<td>SW8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A8</td>
<td>SW9</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A9</td>
<td>SW11</td>
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</table>

<table>
<thead>
<tr>
<th>Figure No.</th>
<th>Location No.</th>
<th>Depth from Gas Face (mm)</th>
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<td>C1</td>
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<td>-10</td>
</tr>
<tr>
<td>C2</td>
<td>TN30P00</td>
<td>-10</td>
</tr>
<tr>
<td>C3</td>
<td>TP00P00</td>
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<tr>
<td>C4</td>
<td>TN15P00</td>
<td>-20</td>
</tr>
<tr>
<td>C5</td>
<td>TN30P00</td>
<td>-20</td>
</tr>
<tr>
<td>C6</td>
<td>TP00P00</td>
<td>-20</td>
</tr>
<tr>
<td>C7</td>
<td>TN15P00</td>
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</tr>
<tr>
<td>C8</td>
<td>TN30P00</td>
<td>-30</td>
</tr>
<tr>
<td>C9</td>
<td>TP00P00</td>
<td>-30</td>
</tr>
<tr>
<td>C10</td>
<td>TP15P00</td>
<td>-30</td>
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<tr>
<td>C11</td>
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</tr>
<tr>
<td>C17</td>
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</table>

Table 4.4. Summary of Measurement Locations and Corresponding Figures