THE INFLUENCE OF IN-CYLINDER FLOWS ON THE FLAME KERNEL GROWTH IN NATURAL GAS FUELLED SPARK IGNITION ENGINES

by

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A thesis submitted in conformity with the requirements for the degree of Doctor of Philosophy

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Dedicated to my wife Lisa and daughter Elena. The satisfaction I receive from my work is greatly enhanced by your presence in my home life.
The influence of in-cylinder flows on the flame kernel growth in natural gas fuelled spark ignition engines

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Doctor of Philosophy, 2000
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Abstract

The objective of this work was to determine the influence of in-cylinder flows on the natural gas combustion process in configurations representative of light duty vehicle spark ignition engines. Specifically, the influence of the flow field on the flame kernel development period was of interest. The interactions between the flow field and the combustion process were quantified by correlating characteristic in-cylinder fluid flow and mass burn rate parameters. The fluid motion, flame kernel development and the overall combustion duration were characterised using: (i) a two-component laser Doppler velocimeter for the in-cylinder fluid flow, (ii) a fibre-optic instrumented spark plug for the flame kernel growth rate, and (iii) a piezo-electric pressure transducer for the in-cylinder pressure from which the overall mass burn rate data is obtained. The measurements were made on an individual cycle basis. Results from a single cylinder V6 optical engine are reported. Physical interpretations of: (i) the flow field evolution and (ii) the interactions between the flow field and the natural gas combustion process are presented.

The main contributions of this work are through: (i) the application of novel
data processing techniques, and (ii) the exploration of the fibre-optic instrumented spark plug as a measurement technique in high swirl engines. For example, the discrete wavelet transform is used to show: (i) the energy cascade process in the non-stationary engine flow field over crank angle phase and frequency; and (ii) how the flow field evolution influences the early stages of the natural gas combustion process. The fibre-optic instrumented spark plug is shown to measure a mass-weighted velocity. Data from the fibre-optic instrumented spark plug data are shown to be biased by the large convection velocities in high swirl engines. This work provides the foundation upon which further investigations into the influence of different in-cylinder flows on the natural gas combustion process in spark ignition engines can be based.
Acknowledgements

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Nomenclature

Acronyms

AC      auto correlation
ADO     allowable drop out
ATDC    after top dead centre
BDC     bottom dead centre
BEI     BEI sensor and motions detectors system company, Inc.
CAD     crank angle degree
CCV     cycle to cycle variations
CDM     crank degree marker
$\text{COV}_{\text{MEP}}$ coefficient of variance (\%)
CWT     continuous wavelet transform
DAQ     data acquisition
DC      direct component (mean)
DFT     discrete Fourier transform
DWT     discrete wavelet transform
EDW     effective drop out width (crank degrees)
EFA     early flame arrival
EFKD    early flame kernel development
EGR     exhaust gas recirculation
ESI     effective sampling interval (crank degrees)
<table>
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<td>EVC</td>
<td>exhaust valve closing (crank degrees)</td>
</tr>
<tr>
<td>EVO</td>
<td>exhaust valve opening (crank degrees)</td>
</tr>
<tr>
<td>FAT</td>
<td>flame arrival time (crank degrees)</td>
</tr>
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<td>FFT</td>
<td>fast fourier transform</td>
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<td>FOSP</td>
<td>fibre-optic instrumented spark plug</td>
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<td>HP-DAQ</td>
<td>Hewlett Packard data acquisition system</td>
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<td>HWA</td>
<td>hot wire anemometry</td>
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<tr>
<td>IC</td>
<td>individual cycle</td>
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<tr>
<td>IMEP</td>
<td>indicated mean effective pressure (kPa)</td>
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<td>IVC</td>
<td>intake valve closing (crank degrees)</td>
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<tr>
<td>IVO</td>
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<td>LDV</td>
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<td>LPP</td>
<td>location of peak pressure (crank degrees)</td>
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<td>MAP</td>
<td>manifold absolute pressure (kPa)</td>
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<tr>
<td>MBT</td>
<td>minimum spark advance for maximum brake torque (crank degrees)</td>
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<tr>
<td>NG</td>
<td>natural gas</td>
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<tr>
<td>NI-DAQ</td>
<td>National Instruments data acquisition</td>
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<td>OD</td>
<td>outer diameter (m)</td>
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<td>OF</td>
<td>overlap factor (° m(^{-1}))</td>
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<td>PMEP</td>
<td>pumping mean effective pressure (kPa)</td>
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<td>PMT</td>
<td>photo multiplier tube</td>
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Symbols

$C_{ur}(\theta)$ normalised cross correlation coefficient function (-)

$Da$ Damköhler number (-)

$F_d$ Flame distortion parameter (-)

$FAT(i, k)$ flame arrival time at fibre i in cycle k (crank degrees)

$FAT_m(k)$ mean flame arrival time in cycle k (crank degrees)

$G_{rr}$ cross-spectral density function($m^2 s^{-3}$)

$k$ cycle number

$l_i$ Integral scale (m)

$l_k$ Kolmogorov scale (m)
$l_m$ Taylor microscale (m)

$Le$ Lewis number (–)

$m$ mass (kg)

$N$ total number of cycles

$p$ pressure (kPa)

$r_c$ compression ratio (–)

$R_s$ fibre mounting radius of fibre-optic spark plug (mm)

$Re$ Reynold’s number (–)

$Re_{cl_m}$ Taylor scale based Reynold’s number (–)

$S_g$ flame expansion speed (mm s$^{-1}$)

$S_L$ stretched Laminar flame speed (m s$^{-1}$)

$S_n$ unstretched Laminar flame speed (m s$^{-1}$)

$S_p$ mean piston speed (m s$^{-1}$)

$S_T$ turbulent flame speed (m s$^{-1}$)

$U$ velocity (m s$^{-1}$)

$u$ fluctuating velocity (m s$^{-1}$)

$\dot{u}$ instantaneous velocity (m s$^{-1}$)

$\ddot{u}$ turbulence intensity (m s$^{-1}$)

$\bar{U}$ mean velocity (m s$^{-1}$)

$\bar{V}_r$ flame convection velocity corrected for mass-weight averaging (m s$^{-1}$)

$\vec{V}_r$ flame convection velocity (m s$^{-1}$)
\[ V \] volume (l)
\[ V_d \] displaced volume (m³)
\[ x_{bf} \] final mass burn fraction (-)
\[ x_{bf}^c \] final mass burn fraction corrected for late burn.
\[ x_b \] mass burn fraction (-)
\[ x_{res} \] residual burned gas fraction (-)
\[ D \] mass diffusivity (m² s⁻¹)
\[ K_a \] Karlovitz strain parameter
\[ L \] Markstein length (m)
\[ L_c \] Markstein length associated with curvature (m)
\[ L_s \] Markstein length associated with aerodynamic strain (m)
\[ L_{cr} \] Markstein length associated with burned gas zone curvature (m)
\[ L_{sr} \] Markstein length associated with burned gas aerodynamic strain (m)
\[ \Delta t_{delay} \] fibre-optic probe enable time delay (ms)
\[ \Delta \theta_b \] crank degree range or bin width
\[ \Delta \theta_{0-1\%} \] 0 to 1% burn duration (crank degrees)
\[ \Delta \theta_{0-50\%} \] 0 to 50% burn duration (crank degrees)
\[ \Delta \theta_{0-5\%} \] 0 to 5% burn duration (crank degrees)
\[ \Delta \theta_{0-80\%} \] 0 to 80% burn duration (crank degrees)
\[ \Delta \theta_{0-80\%} \] overall burn duration (crank degrees)
\[ \Delta \Theta_{EFKD} \] early flame kernel development duration (crank degrees)
\[ \delta_L \quad \text{flame front thickness (m)} \]

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

\[ \gamma_{ur} \quad \text{coherence function (-)} \]

\[ \gamma_{w,ur}(f, \theta) \quad \text{normalised DWT coherence function (-)} \]

\[ \lambda \quad \text{excess air-fuel ratio (-)} \]

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

\[ \tau_r \quad \text{characteristic chemical reaction time (s)} \]

\[ \tau_m \quad \text{characteristic mixing time (s)} \]

\[ \theta \quad \text{crank degree} \]

\[ \psi \quad \text{polar coordinate angle (deg.)} \]

\[ \psi(i) \quad \text{polar coordinate angle of fibre i (deg.)} \]

\[ \psi_r \quad \text{early flame kernel convection direction (deg.)} \]
Chapter 1

Introduction

To make improvements in spark ignition engine fuel economy and emissions, a better fundamental understanding of the interaction between the in-cylinder flows and the combustion process is needed. With this goal in mind, Nagayama et al. (1977), Witze (1977), Rask (1979) and Liou and Santavicca (1982) began probing in-cylinder flow processes within spark ignition engines. Others, such as Witze et al. (1984), Fraser et al. (1986), Arcoumanis et al. (1987) and Corcione and Valentino (1994), did further investigations. The results from these early investigations lead to a better fundamental understanding of the in-cylinder flow processes, but also highlighted the complex nature of the non-stationary in-cylinder flows. Among others, Lancaster et al. (1976) and Ting et al. (1995) investigated the interactions between the in-cylinder flow and combustion processes in research spark ignition engines. Their results indicate that the in-cylinder fluid motions, for example, the bulk charge motion and turbulence intensities near the spark plug at the time of ignition, strongly influence the early flame kernel development period. EFKD. The review of Ozdor et al. (1994) shows that the EFKD has a strong influence on spark ignition engine performance and emission characteristics. The engine configurations in the above cited studies were, however, based on research engines designed specifically for ease of in-cylinder access for measurements. The engine configurations were not based on realistic production engine configurations. Investigations into the influence of fluid motions on the combustion process in configurations representative of spark ignition production engines are
on-going at research facilities throughout the world: see, for example, the work of Endres et al. (1992), Bradley et al. (1996), Johansson and Söderberg (1996), Trigui et al. (1996), Auriemma et al. (1998), Arcoumanis et al. (1998) and Klingmann and Johansson (1998).

Spark ignition engines fuelled by natural gas have the potential of lower regulated exhaust emissions. Meyer et al. (1991), Howes and Rideout (1995) and Bradley et al. (1996) report that engines converted from gasoline to natural gas have reduced non-methane hydrocarbon and carbon monoxide emissions, but results with respect to NO_x emissions vary. Meyer et al. (1991) report slight increases in NO_x emissions for natural gas fuelled engines, whereas Arcoumanis et al. (1998) report substantial decreases. The results of Meyer et al. (1991), Arcoumanis et al. (1998) and Bradley et al. (1996) indicate that natural gas fuelled engines have improved cold start performance and a lower lean operating limit, but less power. Bradley et al. (1996) attribute the differences between gasoline and natural gas fuelled engine performance to: (i) natural gas is a low molecular weight gas while gasoline is a high molecular weight liquid, and (ii) natural gas has slower reaction chemistry, characterised by a lower laminar burning speed. Bradley et al. (1988), Daneshyar et al. (1983) and Arcoumanis et al. (1998) report that the interaction of the in-cylinder flow with natural gas combustion is different than with that of gasoline, suggesting that changing the fuel requires new measurements.

The objective of this work was to determine the influence of in-cylinder flows on the natural gas combustion process in configurations representative of light duty vehicle spark ignition engines. The interactions between the fluid motion, early flame kernel development and the overall combustion duration were measured using: (i) a two-component laser Doppler velocimeter for the in-cylinder fluid flow, (ii) a fibre-optic instrumented spark plug for the EFKD growth rate, and (iii) a piezo-electric pressure transducer for the in-cylinder pressure from which the overall combustion duration data is obtained. The measurements were made on an individual cycle basis and the interactions were quantified by correlating characteristic in-cylinder fluid flow parameters with characteristic combustion parameters.
This thesis documents the investigation. An overview of the current understanding of the physics of the in-cylinder flows, the combustion process, and how they interact is presented in Chapter 2. Chapter 3 outlines: (i) the experimental apparatus developed to measure the in-cylinder fluid flow, EFKD and the overall combustion duration; and (ii) the analysis techniques used to extract characteristic parameters from the data. The results from a single cylinder v6 optical engine are discussed in Chapter 4. Physical interpretations of: (i) the flow field evolution and (ii) the interactions between the flow field and the natural gas combustion process are presented.
Chapter 2

In-cylinder flow and the combustion process

In spark ignition engines, air and fuel mix together in the intake system and are drawn into the cylinder through the intake port. The inducted charge is compressed and combustion is initiated with a spark discharge towards the end of the compression stroke. Combustion increases the in-cylinder pressure which provides $pdV$ work in the power stroke. Heywood (1988) states that the phasing of the combustion process with respect to the piston motion ultimately determines the performance and emission characteristics of SI engines. The rate at which combustion occurs, and hence the phasing at constant spark timing, depends on the thermochemical properties of the charge and the in-cylinder flow field. Engine design has an effect on the in-cylinder flow field at the time of combustion through: (i) intake port design, (ii) combustion chamber shape, and (iii) piston crown design. This chapter provides an overview of in-cylinder flows and the current understanding of how they influence the combustion process in SI engines. A brief overview of how turbulent flows and fundamental combustion processes are characterised is presented in Appendices A and B, respectively.
2.1 Spark ignition engine in-cylinder flows

Systematic improvements in spark ignition engine operation require a better understanding of the in-cylinder flow structures and how they interact with the combustion process. The in-cylinder flows are complex because: (i) mean and turbulence are non-stationary due to the piston and valve motion; (ii) the turbulence intensity is high, typically $>> 10\%$ (Hill & Zhang, 1994); and (iii) there are cyclic variations in the flow structure, where swirl and tumble are examples of large scale structures within the flow which exhibit cyclic variations (Arcoumanis & Whitelaw, 1987).

2.1.1 The development of in-cylinder flow fields

The four general categories of engine in-cylinder flows are (a) disorganised, (b) swirling, (c) tumbling, and (d) squish flows, as shown in Figure 2.1. The intake port design and engine geometry determine the initial in-cylinder flow pattern: (i) low tumble and swirl if the intake flow is directed in the axial direction, as in Figure 2.1(a); (ii) swirl if the flow is directed in the tangential direction, as in Figure 2.1(b); and (iii) tumble if the flow is directed to roll about an axis perpendicular to the cylinder axis, as in Figure 2.1(c). The flow pattern changes during the compression stroke through interaction with the combustion chamber walls and piston crown. The combustion chamber shape and piston crown design determine whether or not squish flows are generated. Hill and Zhang (1994) report that the initial flow pattern, which is modified during the compression stroke, significantly influences the mean and turbulent flow fields at the time of ignition and during combustion.

In all cases, the charge is inducted as a jet through the intake valve. There are a number of mechanisms of turbulence generation, including the large velocity gradients in the shear layers of the intake jet and interactions of the flow field with the combustion chamber walls. The intake jet turbulence decays rapidly during the latter stages of the induction stroke following the rapid drop-off of the jet momentum. Tabaczynski (1983) and Hill and Zhang (1994) report that, in the absence of any intake system generated bulk motions, for example, Figure 2.1(a), a disorganised flow
Figure 2.1: Schematic of the different flow fields in internal combustion engines.
pattern results at the time of ignition and the turbulence energy is relatively low.

For intake systems which induce a swirling flow, the in-cylinder flow at the time of ignition is significantly different than the initially disorganised case because the cylinder wall geometry transforms the intake flow into a long-lived, rotating structure. Arcoumanis and Whitelaw (1987) report that the velocity distribution approaches that of solid body rotation by the end of the intake stroke. The centre of the swirling motion is not necessarily lined up with the cylinder axis and may move about at a characteristic frequency, termed swirl precession by Arcoumanis et al. (1987). This swirling flow persists throughout the intake and compression strokes because there is little internal shear to extract energy from the bulk flow motion. There are frictional losses at the cylinder and combustion chamber walls, spark plug and piston crown. The angular momentum of the swirling flow may decay by 30-50% by the end of the compression stroke. A variety of sources, including Uzkan et al. (1983), Arcoumanis et al. (1987), Kent et al. (1989), Whitelaw and Xu (1995), Kim (1995) and Floch et al. (1995), report that at the time of ignition there are high swirling bulk fluid motions and low to middle level turbulence energy depending on the amount of turbulence produced from the wall interactions.

Kent et al. (1989) and Arcoumanis et al. (1998) show that the flow field evolves differently for intake systems in which tumbling flows are induced. The tumbling vortices also survive through IVC but eventually break-up during the compression stroke because they are compressed along an axis perpendicular to the tumbling axis. The break-up of the tumbling structures enhances the turbulence energy by a factor of up to $O(2)$ relative to non-tumbling intake flows with similar engine geometry. The time at which the tumbling vortex begins to break-up is dependent on engine geometry: the flow at the time of ignition may vary from a high bulk flow with relatively weak turbulence to a low bulk flow with high turbulence levels.

Fansler and French (1988) and Catania and Spessa (1996) show that squish flows are generated by interactions between the piston crown and combustion chamber shape. For example, with a compact combustion chamber shape and a flat-topped piston crown there are squish areas where the charge is forced in an inward radial di-
rection as the piston approaches the cylinder head: Figure 2.1(d) contains a schematic of this process. Squish jets induce large mean velocity gradients which are responsible for the production of turbulence energy as TDC is approached. The influence of these interactions dominate the charge motion as the piston approaches TDC. Fansler and French (1988) report that, if swirl or tumble are present, they interact with the squish flow to produce complex flow patterns with high mean shear rates which contribute to the production of turbulence.

It is evident that a wide range of turbulent flow fields are possible at the time of ignition. The turbulent flow field is of interest at this stage of the engine cycle because of its influence on the combustion process. The following list summarises the general features of SI engine flows near TDC of the compression stroke:

1. The turbulence velocity \( u'_i \) is proportional to the engine speed. The magnitude of \( u'_i \) is dependent on the intake system and the piston geometry: the variation with engine speed is dependent on the piston geometry rather than on the intake system (Witze, 1977; Nagayama et al., 1977; Corcione & Valentio, 1994; Ting et al., 1995; Catania & Spessa, 1996).

2. The mean velocity \( U'_i \) scales with engine speed (Witze, 1977; Catania & Spessa, 1996).

3. The Eulerian integral length scales at the end of the compression stroke are a function of combustion chamber geometry (Tabaczynski, 1983; Fraser et al., 1986; Fansler & French, 1988; Hadded & Denbratt, 1991; Ting et al., 1995; Dimopoulos & Boulouchos, 1997) and the size is independent of engine speed. The integral scales are on the order of the clearance gap in disc chambers and of the clearance height in squish chambers (Arcoumanis & Whitelaw, 1987; Hill & Zhang, 1994). The size of the integral length scales controls the dissipation rate of the turbulence near the time of combustion (Tabaczynski, 1983).

4. The in-cylinder flows are inhomogeneous at the large scales (Witze, 1977), but homogeneous and isotropic in the small scales at high \( Re \) numbers (Ting et al., 1995).
5. There is fine scale structure in the flow field (Tabaczynski, 1983). The size of the fine scale structures decrease with engine speed as $u'_l$ and $Re_T$ increase.

### 2.2 Combustion in spark ignition engines

The combustion process in spark ignition engines begins with a spark discharge towards the end of the combustion stroke. A turbulent flame develops and propagates through the mixture and extinguishes itself at the walls. The combustion process consists of the flame kernel development and the rapid burn phases. The flame kernel development period is defined as the time from spark discharge to the point at which 1% of the charge has burned. This definition historically has depended on the resolution of the measurement technique used to measure the duration of the EFKD, where typical values range from $x_{bf}$ of 1 to 10%. The EFKD period can account for up to 50% of the overall 0-90% mass burn duration: Keck et al. (1987), Hall (1989), Baritaud (1989), Cho and Santavicca (1993), Johansson (1993) and Ting et al. (1995) show that the mixture thermodynamic state, composition and motion in the vicinity of the spark plug influence the EFKD. The rapid burn phase is defined as the period over which 1-90% of the mass charge burns. The results of Lancaster et al. (1976), Nagayama et al. (1977) and Witze et al. (1984) indicate that the average mixture properties and turbulence intensity throughout the cylinder affect this stage of combustion.

#### 2.2.1 Flame kernel development

The flame kernel development period is defined to be the time over which the initial flame kernel burns out from the spark gap and begins to interact fully with the turbulent flow field. The flame kernel development is initiated with a spark discharge. After the spark breakdown event, a self-sustaining flame kernel is formed during the glow discharge phase and is trapped within an eddy; the outer boundary of the flame kernel is roughly spherical in shape (Ozdor et al., 1994). The spark discharge continues to deliver energy to the gas mixture during the glow discharge
phase, and enhances the local flame speed relative to the unstretched laminar flame speed. At this stage of the combustion process, Bianco et al. (1991) have measured initial flame expansion speeds 2 to 3 times the theoretical unstretched laminar flame speed. The thermochemical properties of the fluid near the spark plug gradually take over from the spark discharge process as one of the governing factors of flame kernel development. That is, the mass burn rate becomes limited by the local unstretched laminar flame speed (Bianco et al., 1991). There is no clear separation between the glow discharge and early flame kernel development phases. As the flame kernel grows larger, it begins to interact more fully with the turbulent flow field in the vicinity of the spark gap. A schematic of this process is shown in Figure 2.2. Ting et al. (1995) and Johansson and Olsson (1995) report that the turbulent eddies larger than the flame kernel convect the flame kernel, and eddies smaller than the flame kernel begin to wrinkle the flame front. Ting et al. (1995) show that the flame surface remains simply connected, whereas Keck et al. (1987) and Cho and Santavicca (1993) report a multiply-connected flame front. The type of flame front depends on the conditions local to the spark plug at the time of ignition. The end of the early flame kernel development period occurs once a turbulent flame is fully developed. Whether a fully developed turbulent flame is ever formed in SI engines is debatable because Checkel and Ting (1993) have shown that the turbulent flame speed continues to increase over time.

The EFKD period duration depends on the flow field and the laminar flame speed, $S_L$, of the mixture in the vicinity of the spark plug at the time of spark discharge. Cho and Santavicca (1993). Stone et al. (1996) and Sodre (1997) show that mixtures with lower $S_L$ have longer EFKD periods. The influence of the flow field is through: (i) the large convective scales associated with coherent structures larger than the flame kernel (Nagayama et al., 1977; Pischinger & Heywood, 1990); (ii) the turbulent eddies larger than the flame kernel (Bradley et al., 1988; Ting et al., 1995); and (iii) the small turbulent mixing scales (Bradley et al., 1988; Johansson, 1993; Ting et al., 1995). The decomposition of the turbulent flow field into large and small scales depends on the dynamic size of the flame kernel. The techniques used to decompose
Figure 2.2: Schematic of the early flame kernel development.

(a) The early flame kernel burns within an eddy in the spark gap.

(b) The flame kernel burns out from the spark gap and begins to interact with the turbulent flow field.

(c) The flame kernel has grown large enough to interact fully with the turbulent flow field.
the flow field are discussed in Chapter 3.

The large flow field convective scales stretch the flame kernel when it is anchored on the spark plug: larger mean velocities increase the amount of stretch. This may decrease the EFKD period duration:

1. by providing a larger flame front surface area. Since mass diffusion into the reaction zone is the rate limiting step, increasing the flame front surface area increases the rate of unburned charge entrainment into the reaction zone. Thus, the mass burn rate increases.

2. by reducing the heat losses to the electrodes by pushing the flame kernel out of the spark gap. This may lead to a larger $S_L$ because the flame temperature is higher due to lower heat losses (Pischinger & Heywood, 1990; Dulger & Sher, 1995), and

3. by providing a source for turbulence energy to maintain a high $u'$.

This holds true as long as the flame stretch is not large enough to decrease the stretched laminar flame speed $S_n$ relative to the non-strained $S_L$ and consequently decrease $S_T$. The results of Bradley et al. (1988) show that if the aerodynamic strain rate is too high the flame kernel may be quenched. Appendix B contains details regarding the influence of aerodynamic strain on the laminar flame speed. Dulger and Sher (1995) report a convection velocity which maximises the turbulent flame speed.

The turbulence intensity and length scales of the in-cylinder flow field affect the EFKD period duration. Hires et al. (1978) summarise the influences through

$$
\Delta \Theta_{EFKD} = C \left( \frac{l_I}{u'} \right)^{1/3} \left( \frac{l_m}{S_n} \right)^{1/3}
$$

(2.1)

where $C$ is determined by engine geometry: the integral, $l_I$, and micro, $l_m$, scales are defined in Appendix A. $\Delta \Theta_{EFKD}$ increases with larger turbulence length scales and decreases with $u'$ and $S_n$. There are interactions between these parameters. For example, increasing $u'$ decreases $l_m$, both of which influence $S_n$, as described in
Appendix B.3. Checkel and Ting (1993) investigated the separate effects of turbulence intensity and scale: their results indicate that small scale turbulence is more effective in enhancing the burning velocity of a growing flame than large scale turbulence. They also report that the turbulent flame speed increases linearly with $u'$.

The large scale turbulence influences $\Delta \Theta_{EFKD}$ by determining the size of the first eddy burned: after the glow discharge phase, the flame kernel grows at $\approx S_L$ until the fuel in the first eddy is consumed. Keck et al. (1987). Ting et al. (1995) and Sodre (1997) state that if the average size of the large scale turbulence is increased then the expected $\Delta \Theta_{EFKD}$ increases. The characteristic scale in Equation 2.1 is a weighted average of the integral and micro scales, with more weight given to the micro scale. As discussed in Appendix A, the integral scale is determined by engine geometry, and the micro scale is dependent on the magnitude of $u'$ and turbulence dissipation rate.

The turbulence scales larger than the flame kernel cause random movement of the flame kernel about the spark plug, where flame kernel displacements up to distances of several millimeters have been observed (Ozdor et al., 1994). The random walk of the flame kernel results in variations in the instantaneous contact area of the flame kernel with the spark electrodes. The flame front surface area available for mass transfer of fresh charge into the flame reaction zone is affected, which influences the mass burn rate. The magnitude of the cycle to cycle variations, CCV, in the combustion process duration are influenced by the random walk of the flame kernel. The influence of CCV on engine operation is discussed ahead in subsection 2.2.3.

A number of sources, including Bradley et al. (1988), Ozdor et al. (1994), Ting et al. (1995), Stone et al. (1996), Whitelaw and Xu (1995) and Sodre (1997), show that enhanced small scale mixing speeds up turbulent flame development. The small scale turbulence influences $\Delta \Theta_{EFKD}$ by increasing the mass burn rate by increasing the mass transfer rate of fresh charge into the flame reaction zone. The effect is seen through the dependence of $\Delta \Theta_{EFKD}$ on $u'$ and $l_m$, where a larger $u'$ results in more turbulence energy at the smaller scales; the influence of $u'$ on the dissipative scales is seen in Equations A.5. and B.9–B.10.

As for the influence of specific in-cylinder flow patterns, the work of Kent et al.
(1989). Hadded and Denbratt (1991). Floch et al. (1995) and Arcoumanis et al. (1998) indicate that tumble decreases the EFKD period because either: (i) $u'$ is increased at the time of ignition through the mean motion break-down into turbulence; (ii) the bulk flow velocity is increased if the mean motion break-down occurs after the time of spark discharge; or (iii) a combination of the increase in $u'$ and bulk flow velocity if the break-down process of mean motion to turbulence is on-going at the time of ignition. The results of Uzkan et al. (1983), Arcoumanis et al. (1987), Kent et al. (1989), Whitelaw and Xu (1995), Kim (1995) and Floch et al. (1995) indicate that swirl decreases the EFKD period by increasing the bulk flow velocity, and to a lesser extent, by enhancement of the $u'$. Note that for squish flows, the increase in $u'$ occurs after the EFKD period unless the ignition event occurs near TDC.

### 2.2.2 Rapid burn phase

The rapid burn phase, RBP, begins once the flame kernel has developed into a turbulent flame. Arcoumanis and Whitelaw (1987). Hill and Zhang (1994) and Johansson and Olsson (1995) show that the average turbulence intensity throughout the combustion chamber has the strongest influence on the RBP. This is in marked contrast to the EFKD period where the local conditions near the spark have the strongest influence on the combustion process. The turbulence intensity and laminar flame burning velocity govern the duration of the RBP. Increasing either the $u'$ or $S_L$ decreases the RBP by increasing the combustion rate. The RBP duration for fuels with different laminar flame burning speed is, however, similar in length because the burn rate is dominated by the magnitudes of the turbulence velocity, $u'$.

The magnitude of $u'$ during the RBP is dependent on engine speed and the convection velocity. Swirl, tumble and squish flows increase, to varying degrees, $u'$ during the RBP which decreases the RBP duration. The turbulence velocity $u'$ also increases with engine speed, but the RBP duration increases slowly with engine speed in terms of crank degrees.
2.2.3 Cycle to cycle variations

Cycle to cycle variations, CCV, in the combustion phasing with respect to piston motions degrade SI engine performance because the optimum spark timing is set for an average phasing. Heywood (1988) states that the CCV's in the combustion phasing result in unstable engine performance and increased hydrocarbon emissions which limit the use of dilute mixture strategies to achieve low NO\textsubscript{x} emissions and improved fuel economy. Keck et al. (1987) and Ozdor et al. (1994) show that the cyclic variations in the duration of the EFKD period, ΔΘ\textsubscript{EFKD}, are the main source of combustion phasing CCV. Engine configurations with longer ΔΘ\textsubscript{EFKD} are more susceptible to CCV. The cause of these variations in ΔΘ\textsubscript{EFKD} are variations in the flow field and mixture composition around the spark plug at the time of spark discharge. Ozdor et al. (1996) theorise that the problem arises because a point ignition source is used. The entire combustion development depends on a slow EFKD. The degree to which CCV in combustion phasing occurs depends on engine design and operating conditions.

During the EFKD period the flame kernel is stretched in a consistent direction by the largest, coherent scales of motion such as swirl or tumble. Within the engine, the large scale turbulent eddies, however, tend to 'kick' the flame kernel about in a random direction. Keck et al. (1987) and Ting et al. (1995) show that the effect of the random walk is to vary the flame front surface area contact with electrodes and walls from one cycle to the next. Depending on the relative location of the quench surfaces to the mean convection and random walk direction of the flame kernel, the combustion duration can be very different from one cycle to the next. The factors influencing CCV in ΔΘ\textsubscript{EFKD} are: (i) the mean velocity \( U \); (ii) the turbulence velocity \( u' \); (iii) the distribution of the turbulence energy among the scales; and (iv) the combustion chamber shape and spark plug location. The effects of the fluid flow field on the combustion process are summarised here.

1. Increasing \( U \) improves combustion stability (Nagayama et al., 1977; Floch et al., 1995; Baby, 1997; Johansson & Söderberg, 1996; Hill & Zhang, 1994; Arcouma-
nis & Whitelaw. 1987). The reasons are: (i) a larger $U$ provides a consistent convection direction for the flame kernel, and (ii) the $\Delta \Theta_{EFKD}$ is reduced through enhanced development of the turbulent flame. A large $U$ may, however, quench the flame, especially under light load conditions where the laminar flame speed is relatively low (Dulger & Sher. 1995).

2. Increasing $u'$ with constant engine geometry improves combustion stability (Johansson & Söderberg. 1996; Stone et al., 1996; Whitelaw & Xu. 1995); constant engine geometry implies constant integral scales, and enhanced small scale mixing (Hill & Zhang. 1994; Tabaczynski. 1983). The enhanced small scale mixing shortens $\Delta \Theta_{EFKD}$ and thus the effect of the random motion of large scale eddies is reduced (Johansson & Söderberg. 1996; Ting et al., 1995). The influence of increasing $u'$ is approximately independent of excess air–fuel ratio (Johansson. 1993).

3. It is expected that configurations with larger integral scales have longer EFKD durations and, therefore, are more susceptible to CCV.

4. The combustion stability is strongly influenced by the combustion chamber shape (Johansson & Olsson. 1995). This is due to variations in $U$, $u'$ and turbulence length scales from one configuration to the next.

The laminar flame speed of the local mixture also strongly influences the CCV in $\Delta \Theta_{EFKD}$ because (i) $S_n$ determines the rate of growth of the flame kernel, and (ii) the susceptibility of the flame kernel to flame stretch quenching is proportional to $S_L$. Cho and Santavicca (1993) report that the inhomogeneity in the local mixture composition with respect to air, fuel and residual gas also strongly influences the CCV in $\Delta \Theta_{EFKD}$.

**Natural gas versus gasoline** Natural gas fuelled engine susceptibility to CCV is expected to be different than gasoline fuelled engines. This is because natural gas has a lower $S_L$ than gasoline; the $\Delta \Theta_{EFKD}$ is longer and may be more susceptible to CCV than gasoline engines. However, natural gas as a gas tends to mix better in the
intake system than gasoline: the improved mixture homogeneity may result in smaller \( \text{CCV} \). This investigation concentrates on the effect of the flow field on the combustion process. One of the design criteria which must be investigated is the degree of \( \text{CCV} \) in engine operation.

**Summary**  From the above discussion, it is evident that the interactions between the flow field, the combustion process, and engine operating characteristics are complex. To investigate the potential of fuelling spark ignition engines with natural gas, a better fundamental understanding of the interactions between the flow field and the natural gas combustion process is needed. In this work, the influence of the flow field on the early flame kernel development is specifically of interest. To obtain a better understanding of the interactions between the flow field, combustion process and engine operating characteristics, simultaneous measurements of the in-cylinder flow near the spark plug, the early flame kernel development and the overall mass burn rates were made on an individual cycle basis in a configuration representative of a production spark ignition engine. The experimental apparatus and analysis techniques used in this investigation are outlined in the next chapter. The results are discussed in Chapter 4.
Chapter 3

Experimental apparatus and analysis techniques

In this chapter, the measurement and analysis techniques used to investigate the influence of the flow field on the natural gas combustion process are discussed. The methodology used in this investigation was to characterise: (i) the in-cylinder flow near the spark plug with single point velocity measurements; (ii) the early flame kernel development period with a fibre-optic instrumented spark plug; and (iii) the combustion process through a mass burn analysis of the measured in-cylinder pressure. Simultaneous velocity-pressure measurements were made to characterise the influence of the flow on the combustion process, and simultaneous FOSP-pressure measurements were made to characterise the influence of the EFKD period on the engine operation. When discussing the analysis techniques used in this work particular attention is paid to the velocity and FOSP data analysis techniques as there are many relevant, unresolved issues. For example, the limitations in the standard techniques used to decompose engine flows into mean and turbulence are highlighted in section 3.2, and a new decomposition technique is introduced which gives a different view of the flow. As for the FOSP data analysis procedures, solutions which minimise the introduction of bias into the FOSP data in high swirl flows are described in section 3.4. This discussion, however, begins with a description of the experimental apparatus.
3.1 Experimental Apparatus

The main components of the apparatus used in this investigation are (i) the single cylinder engines, (ii) a two-component laser Doppler velocimeter, (iii) a piezo-electric pressure transducer, and (iv) a fibre-optic instrumented spark plug. These four components are described in the following subsections.

3.1.1 Single cylinder engines

Measurements were made in four single cylinder engines: these include an L-head research engine, non-optical GM 2.8 and 3.1 L SCV6 engines, and an optical GM 3.1 L SCV6 engine. The engine specifications are listed in Table 3.1. The reason the L-head research engine was included in this work was because the optical access needed for the velocity measurements could be set up in a much shorter time relative to the SCV6 optical engine. The L-head optical engine facilitated the development of both the LDV and pressure measurement and analysis systems by the author while Mr. H. Jääskeläinen set-up the SCV6 engine. The non-optical SCV6 2.8 and 3.1 L builds were included in this study because the FOSP measurement and analysis system was initially tested in these engines. The optical SCV6 3.1 L engine was solely used for the simultaneous velocity-pressure and FOSP-pressure measurements.

The optical access needed by the LDV system to make measurements was achieved through a window in the cylinder head for the L-head engine, and using a Bowditch piston in the production SCV6 3.1 L engine. Schematics of the optical access design and coordinate system for these two engines are contained in Figure 3.1.

A schematic of the engine control systems is contained in Figure 3.2: the same general control scheme was used for both the L-head and SCV6 engines. The control scheme for the L-head engine was developed by the author, while Mr. H. Jääskeläinen developed the SCV6 engine control scheme.
Figure 3.1: Schematic of optical access and coordinate system for single cylinder engines.
Table 3.1: Single cylinder engine specifications.

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<td>3.1 l Optical</td>
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</tbody>
</table>

Figure 3.2: Schematic diagram of the single cylinder L-Head research engine.
3.1.2 In-cylinder velocity measurements

As mentioned previously, a two component laser Doppler velocimetry (LDV) system was used to measure the in-cylinder velocity. LDV was chosen to make the velocity measurements because it has a large dynamic range, high spatial resolution and is non-intrusive. A short description of the underlying principles of LDV and the tests carried out to validate the velocity data is presented next.

Laser Doppler velocimetry is a single point measurement technique which uses two crossed coherent laser beams to measure one velocity component of a flowing fluid. A schematic of the LDV system used in this study is contained in Figure 3.3(a). To measure the velocity, the flow must be seeded with particles. When a seed particle passes through the intersection, or probe, volume of the laser beams the modulated frequency of the scattered light is observed at a photo-multiplier tube and converted to a velocity by multiplication with a linear conversion factor. Velocity data cannot be acquired continuously using LDV. Rather, data is acquired only when a particle in the probe volume scatters enough light to give a high signal-to-noise ratio (SNR) as seen at the photo-multiplier tube. High LDV data rates can only be achieved with a well designed seeding and optical system.

Haghgoie et al. (1982, 1986) measured Eulerian fluctuating frequencies \( f_{\text{max}} = \mathcal{O}(10 \, \text{kHz}) \) in engine flows. The Lagrangian velocity fluctuating frequencies to which the seed particles must respond are expected to be less than the Eulerian frequencies by a factor of \( R_t^{1/4} \) (Tennekes & Lumley, 1972) where \( R_t \) is \( \mathcal{O}(10^3) \) in SI engines (Heywood, 1988). Based on these order of magnitude estimates, accurate velocity measurements require: (i) LDV data rates in excess of 20 kHz per channel, and (ii) seed particles small enough to follow the highest spatial flow accelerations of \( \partial U_i/\partial x_i = \mathcal{O}(2 \, \text{kHz}) \). The LDV optical and seeding systems designed to meet these needs are summarised in Table 3.2 and Figures 3.3(a) and 3.3(b), respectively. The main features to note for the optical system are: (i) the 2.6X beam expander, which improved the SNR substantially; (ii) that a frequency shift of 40 MHz was used to by-pass the noisy TSI down-mixers; and (iii) the laser and scattered light are trans-
Figure 3.3: Schematic of the LDV optical and seed systems. The optical system parameters are listed in Table 3.2. The seed system uses 0.2μm median particle diameter TiO₂ and Aerosil Degussa R972 desiccant.
Table 3.2: LDV optical system. A 5 W Coherent Ar-Ion laser is used with a 2.6× beam expander. The scattered light is collected in back-scatter mode and the signal is processed using the IFA750 digital burst correlator.

<table>
<thead>
<tr>
<th></th>
<th>Channel 1</th>
<th>Channel 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wavelength (nm)</td>
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<td>514.5</td>
</tr>
<tr>
<td>Front focal lens length (mm)</td>
<td>480.0</td>
<td>480.0</td>
</tr>
<tr>
<td>Beam spacing (mm)</td>
<td>67.6</td>
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</tr>
<tr>
<td>Fringe spacing (μm)</td>
<td>3.65</td>
<td>3.47</td>
</tr>
<tr>
<td>Beam crossing half angle (deg.)</td>
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<td>-1.04</td>
</tr>
<tr>
<td>Frequency shift (MHz)</td>
<td>40.0</td>
<td>-40.0</td>
</tr>
<tr>
<td>Probe volume major axis (mm)</td>
<td>0.62</td>
<td>0.65</td>
</tr>
<tr>
<td>Probe volume minor axis (μm)</td>
<td>43.7</td>
<td>46.1</td>
</tr>
<tr>
<td>Velocity component</td>
<td>u</td>
<td>v</td>
</tr>
</tbody>
</table>

mitted via fibre–optic cables. The main features to note for the seed system, shown in Figure 3.3(b), are that 0.2 μm TiO2 particles are used, and that the intake air to the engine is dried to a dew point < 0° C to prevent agglomeration of the seed particles. Validation tests, described below, were carried out to ensure that the seed particles followed the accelerations in the flow.

Validation tests were carried out on the LDV data to determine whether the velocity is measured accurately. The issues to resolve were whether the seed particles were able to follow the accelerations in the flow and whether the presence of seed particles affected the combustion process. The test results are summarised here.

1. Measurements were taken in the L-head research engine in the x=0 plane in a grid with 5 mm spacing between data points. One hundred cycles of the velocity data were ensemble averaged. The measured in-cylinder pressure and intake air temperature were used to calculate the gas density ρ. The average mass inducted per engine cycle was calculated as

\[ m = \int_{\theta=15}^{\theta=210} \int_{z=-35mm}^{z=35mm} \int_{y=-35mm}^{y=35mm} \rho(\theta)U(\theta, x = 0, y, z) dy dz d\theta \]

The results agreed to within 5% of independent mass flow measurements using a calibrated hot wire sensor and lend credibility to the measured velocity.
2. Seed particle tests were carried out in the L-head engine to optimise the choice of seed particles. The issues to resolve were which seed particles were (i) small enough to follow the accelerations in the flow, and (ii) large enough to scatter enough light to maintain high SNR. Three different types of seed particles were tested: 1.0 µm median diameter Alumina and 0.2 and 1.0 µm TiO₂ seed particles. The results indicate that the 1.0 µm TiO₂ and alumina seed particles produced 10–15% larger data rates, but lagged the intake flow accelerations by up to 5% relative to the 0.2 µm TiO₂ seed particles. It was concluded that 0.2 µm TiO₂ seed particles should be used even though the resulting data rates were lower.

3. The quality of the LDV signal was examined. Clean Doppler bursts were observed. The minimum number of crossings was set to eight to reduce the measurements from multiple bursts per particle.

4. The pressure measurement and analysis system was used to determine whether the combustion process was affected by the presence of the seed particles in the L-head research engine. The results indicate that the combustion process is unaffected by the presence of the seed particles (Ancimer, Wallace, & Jääskeläinen, 1999).

3.1.3 In-cylinder pressure measurements

The design of the in-cylinder pressure measurement system, used to characterise the combustion process in the engines, is discussed next. A Kistler Model 6121A piezoelectric pressure transducer was chosen to make the in-cylinder pressure measurements. The Kistler Model 6121A is a small, fast responding pressure transducer with a 0.4-70 bar measurement range; Randolph (1990b) shows that these characteristics are needed to make accurate in-cylinder pressure measurements. The underlying principle of piezo-electric pressure transducers is that the quartz crystal, confined within the device, produces an electrical charge proportional to a pressure input. The charge output from the piezo-electric transducer is converted to a pressure change by first
passing the charge through a charge amplifier. Where the input charge is converted to an output voltage with a known gain. The output voltage is converted to a pressure change using a linear conversion factor. These transducers, however, only respond to changes in pressure (Kistler, 1993). Absolute pressures are obtained by referencing the transducer to the measured absolute intake or exhaust manifold pressure.

One of the main benefits of using the Kistler 6121A pressure transducer is its small size. This feature allows in-cylinder access without the need for major modifications to the cylinder head. When designing access to the combustion chamber, the recommendations of Randolph (1990a, 1990b) were followed to minimise thermal shock effects. Thermal shock effects result from the large and rapid in-cylinder temperature changes. The measures taken to minimise thermal shock effects were two-fold: (i) a slotted pressure transducer mounting adaptor was built based on the design of Randolph (1990a), and (ii) the piezo-electric transducer output was referenced to the intake manifold pressure on an individual cycle basis. Further details regarding the mounting scheme are discussed by Burcar (1996).

A data acquisition system was developed to measure the in-cylinder, intake and exhaust manifold pressures along with the spark timing. The intake and exhaust manifold pressures were measured using GM manifold absolute pressure MAP sensors, and the spark timing was measured through a conditioned signal from an inductive pick-up on the ignition system secondary spark plug lead. Data acquisition was triggered on the exhaust top dead centre pulse and five samples were acquired per CAD. A National Instruments AT-MIO-16E-1 data acquisition board was used to make the measurements.

Motored and fired diagnostic tests, as recommended by Lancaster et al. (1975), were performed to ensure that the measured pressures were accurate. For example, one of the motored tests involved acquiring motored pressure data and comparing the measured pressure data with a motored simulation. One of the fired tests involved checking for realistic polytropic compression and expansion coefficients. Problems, such as (i) incorrect phasing of the data acquisition, (ii) inaccurate mass flow measurements, and (iii) intake manifold leaks, were pinpointed using these diagnostic
tools: the problems were subsequently corrected. Minimising these types of errors is important. For example, the error is linearly dependent on the accuracy of the intake mass flow rate. Realistic IMEP over a wide range of operating points lend further confidence to the measured pressure results.

3.1.4 Early flame kernel development measurements

Ting et al. (1994) show that, in general, the in-cylinder pressure measurement and analysis systems are not sensitive enough to give accurate early mass burn growth rates because of the relatively small pressure rise attributable to combustion in the EFKD period. A more sensitive instrument is the fibre-optic instrumented spark plug. The FOSP is a spark plug with a ring of fibre-optic probes mounted on the periphery. When the flame passes over a fibre-optic probe the light emitted by the flame is seen by the fibre-optic probe and passed to a photo-multiplier tube. The output voltage from the photo-multiplier tube is amplified. When the output voltage exceeds a threshold level the flame is assumed to have arrived at the fibre-optic probe. The eight flame arrival times are recorded on an individual cycle basis and are used to calculate the early expansion rate of the flame kernel, $S_g$, and the velocity at which the flame kernel is convected, $V_c$. The two models generally used to calculate $S_g$ and $V_c$ from the eight flame arrival times are the cubic spline model of Bianco et al. (1991) and the ellipse model of Kerstein and Witze (1990). Both models are outlined in Appendix G.

The fibre-optic instrumented spark plug was developed in-house based on the design of Witze, Hall, and Wallace (1988). The FOSP was made from a Champion RS15LYC 14 mm spark plug with a 22 kΩ resistor element. The electrode diameter is 2.54 mm and the spark gap is 1.4 mm. Eight equally spaced 750 μm fibre-optic probes are flush mounted at a radius of 5.35 mm on the periphery of the FOSP and have a 10° cone of acceptance. The light seen by the probes is short pass filtered at 450 nm using a Newport 10SWF-450 filter before being passed to a Hamamatsu IP28 photomultiplier tube. The output voltage from the photo-multiplier tubes is amplified and passed to the data acquisition system. To maintain approximately equal responsivity
between the eight fibre-optic probes. the eight PMT amplifiers were adjusted to give
the same average peak output voltage (within 0.5 V) over 20 engine cycles. The
amplification was maintained at a low enough level to not saturate the DAQ system
on any individual cycle.

The PMT output voltages were acquired simultaneously with the in-cylinder pressure
data using a National Instruments AT-MIO-16E-1 data acquisition board. A
unity-gain differential to single-ended converter was built to allow the AT-MIO-16E-1
to acquire the 12 differential channels. This converter degrades the analogue to digi-
tal signal settling time of the DAQ system such that the maximum engine speed at
which data can be acquired is 2200 RPM. Data acquisition for one engine cycle was
triggered by an exhaust TDC pulse from a BEI shaft encoder. The data were sampled
five times per CAD and written out to files for post-processing.

**Simultaneous measurements** The technique used to make the simultaneous velocity-
pressure-EFKD measurements is presented in Appendix C. A control system had to
be developed because the FOSP-pressure and velocity data are acquired on separate
PC’s.

### 3.2 Velocity analysis

Although in-cylinder velocity measurements have been made for over two decades (Hill
& Zhang, 1994), the issue of how to decompose the flow into mean and turbulence
velocities remains unresolved. In stationary flows, the mean flow is associated with
the energy in the DC component of the flow, and the flow energy in the higher fre-
quency components is associated with the turbulence. Engine flows are, however,
non-stationary such that the frequencies associated with the mean and turbulent flow
processes overlap. Engine flows are also characterised by high turbulence intensities
and quasi-periodic variations in the mean. Because of the overlapping scales and the
quasi-periodic variations in the mean, there is no clear cut method to differentiate
between mean and turbulent flow energy.
The velocity data decomposition techniques generally used in engine flows include the ensemble and cyclic averaging techniques. Previous work by Sullivan, Ancimer, and Wallace (1999), however, has shown that these methods do not decompose the flow into physically meaningful components, but an arbitrary mean and turbulence. Ensemble averaging uses a statistical Reynolds based filtering operator. Liou et al. (1984) were the first to show that ensemble averaging over-estimates the turbulence energy by assigning the quasi-periodic bulk flow energy as turbulence. Cyclic averaging, on the other hand, uses a low pass filtering scheme to calculate the turbulence. A scale separating the bulk flow and fluctuating velocities is used to define the low pass filter. Liou et al. (1984). Lorenz and Prescher (1990) and Catania and Mittica (1990) have all introduced different methods to filter the measured velocity into mean and turbulent flow. By virtue of the low pass filtering used to define the mean flow by cyclic averaging techniques, no low frequency events are included as turbulence. The choice of filtering scale, therefore, biases the turbulence scale information. Neither the ensemble nor cyclic based techniques is entirely satisfactory in decomposing engine flows because of the biases they introduce.

In the following subsections these two decomposition techniques are discussed in detail. The different techniques are compared and critiqued based on the degree to which the turbulence is biased. A new decomposition technique, based on wavelet transforms, and developed to identify the quasi-periodic bulk flow variations, is also introduced: wavelet transform theory is outlined in Appendix D.

### 3.2.1 Ensemble based averaging

Ensemble averaging defines the bulk, or mean, velocity as an exactly repeatable phase average for all engine cycles. The mean velocity is calculated as the ensemble average over N cycles:

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

(3.1)

where \(\bar{U}(\theta)\) is the mean velocity. \(\tilde{u}(\theta, k)\) is the instantaneous velocity in cycle \(k\), and \(\theta\) is the crank degree or phase. The fluctuating velocity is the difference between the
instantaneous and mean velocities

\[ u(\theta, k) = \bar{u}(\theta, k) - \bar{U}(\theta). \] (3.2)

and the statistical properties of the turbulent flow are calculated either over the \( N \) cycles.

\[ [u'(\theta)]^2 = \frac{1}{N} \sum_{k=1}^{N} [u(\theta, k)]^2. \] (3.3)

or within an engine cycle \( k \) over a crank angle range where the statistical properties of the flow are assumed to be stationary.

\[ [u'(\theta_o, k)]^2 = \frac{1}{\Delta \theta} \int_{\theta_o - \frac{\Delta \theta}{2}}^{\theta_o + \frac{\Delta \theta}{2}} [u(\theta, k)]^2. \] (3.4)

The crank angle range is centred on \( \theta_o \) and is of extent \( \Delta \theta \).

If the phase-conditioned turbulence velocities are calculated using Equation 3.3, a bias is introduced due to inclusion of the energy of the quasi-periodic mean flow variations. Arcoumanis and Whitelaw (1987) give: (i) precessing swirl, (ii) tumble and (iii) flapping intake or squish jets, as examples of energetic events present in engine flows which exhibit cyclic variations in their phasing, amplitude and orientation. To show how the turbulence is biased, first consider, in Figure 3.4(a), the ensemble average mean velocities at one point in the L-head research engine during the intake stroke. The non-stationary mean velocities associated with the intake jet are evident. It is the cycle to cycle variations in the phasing and orientation of this intake jet with respect to the measurement point that introduce the bias into the turbulence. To show how these cyclic variations in the intake jet flow might bias the turbulence, the fluctuating velocity calculated from the ensemble average for Cycle 15 of the data is shown in Figure 3.4(b). Two large scale, relatively high energy events at 30–40 CAD and 60–80 CAD are evident. Superimposing the events at 30–40 and 60–80 CAD onto the mean flow indicates that they are in phase with the large accelerations in the mean. Jitter in the phase of the intake jet might be the cause of these large scale
drifts.

![Graphs showing velocity fluctuations](image)

(a) Ensemble mean velocities  
(b) Ensemble average based filtering

![Graph showing cyclic filtering](image)

(c) Cyclic based filtering. Low pass filter at 120 (engine cycle)$^{-1}$.

Figure 3.4: Ensemble mean velocities and the fluctuating u-component fluctuating velocity in cycle no. 15. Motored operation at $\tau = (0, -25, 15)$, 900 RPM and $N = 200$ cycles.

To demonstrate this possibility, the bias introduced by phase jitter is estimated through a high energy event modelled as a sine wave with a 36 CAD period, as shown in Figure 3.5(a). The 36 CAD period is associated with a frequency of $f = 20$ (engine cycle)$^{-1}$. That is, it is an event whose period can repeat itself 20 times in one 720 CAD engine cycle. Random jitter in the phase, that is, the starting crank angle, over 18 CAD was allowed, parallel to variations in real engines. The turbulence was
modelled as white noise. One hundred cycles of data were generated and ensemble averaging was applied. The results are compared with the input signal statistics in Figure 3.5. Figure 3.5(a) demonstrates that the ensemble average fails to capture the individual cycle mean. The ensemble average is of significantly lower magnitude and longer duration. The resulting fluctuating velocity of modelled cycle number 10, plotted in Figure 3.5(b), shows a large scale local event. This event is the result of the cyclic variations in the phase of the energetic event, and belongs to the mean flow. Ensemble averaging over-estimates the turbulence velocity where cyclic variations occur, as shown in Figure 3.5(c). The energy associated with these cyclic variations are, within a frequency band width of 10-100 (engine cycle)$^{-1}$. The ensemble average turbulence, PSD, over-estimates the energy content of the turbulence by a factor of approximately 50, as shown in Figure 3.5(d). The turbulence is biased in a broad range of frequencies, which is due to the fact that a large number of Fourier frequencies are need to represent a transient, local event. This simple model demonstrates that the quasi-periodic variations in the mean can significantly bias the turbulence energy, and that the bias can extend over a broad range of frequencies.

Similar biasing of the turbulence occurs if the orientation of the intake jet with respect to the measurement volume changes from one cycle to the next. For example, consider if the phase of the intake jet remained constant, but the orientation of the intake jet changed from one cycle to the next. The proportion of the mean velocity of the intake jet projected onto the $u$ and $v$ component measurement directions would vary from one cycle to the next. These quasi-periodic variations in the magnitude of the individual cycle $u$ and $v$ component mean velocities would be seen as turbulence by the ensemble averaging technique. The ensemble average turbulence velocity is therefore biased by the quasi-periodic variations in the mean flow.

There is an additional bias introduced because the intake jet is non-stationary, non-homogeneous and anisotropic. In this case, the statistics of the turbulence depend (i) on the phase of the measurement, and (ii) on the location and orientation of the measurement within the intake jet. The phase jitter and variations in the orientation of the jet with respect to the measurement location, therefore, cause quasi-
(a) Comparison of the input mean and Ensemble average for cycle 10.

(b) The Ensemble average fluctuating velocity in cycle 10.

(c) Comparison of the input and the Ensemble average turbulence.

(d) Comparison of the input and the Ensemble average turbulence power spectrum.

Figure 3.5: Ensemble average results. The input signal is one period of a sine wave with frequency 20 (engine cycle)$^{-1}$ and cyclic variations in the phase of the event. Statistics were calculated for $N = 100$ cycles.
periodic turbulence broadening biases. The ensemble based turbulence statistics at a particular phase \( \theta \) and location \( \mathcal{F} \) are statistically averaged over time and space. The extent of the average over time depends on the amount of phase jitter. The extent of the average over space depends on the variations in the orientation of the jet with respect to the measurement point. To remove these quasi-periodic turbulence broadening biases, a conditioned ensemble average is needed. Conditioning with respect to both intake jet phasing and orientation is required.

### 3.2.2 Cyclic based averaging

Cyclic based averaging techniques estimate the mean by low pass filtering the instantaneous velocity either in the temporal or the frequency domain. The scale based averaging techniques proposed by Liou et al. (1984), Lorenz and Prescher (1990) and Catania and Mittica (1990) use a cut-off frequency where the energy below this frequency is assigned to the mean and the energy above is the turbulence. Fansler and French (1988) argue that the cut-off frequency should be chosen based on the physical scales of interest in the flow. Liou and Santavicca (1985) suggest an alternate method to choose the cut-off frequency based on the highest frequency with significant energy in the power spectral density function, PSD, of the ensemble average of the velocity data. With rapid variations in the mean flow, such as during the intake or exhaust stroke, the ensemble mean velocity may contain energy at high frequencies. Choosing a cut-off frequency using the second technique may, therefore, result in too severe a filtering. Regardless of how the cut-off frequency is chosen, Liou et al. (1984) and Catania and Mittica (1990) state the ensemble average of the individual cycle mean velocity should coincide with the overall ensemble average velocity. A value of \( \chi^2 = 1 \) is expected (Press et al., 1980).

To illustrate the effect of the low pass filtering scheme, the velocity data were filtered using a cut-off frequency of 120 (engine cycle)^{-1} for both the \( u \) and \( v \) velocity components. The cut-off frequency was chosen to filter out the energy associated with the mean flow variations. There are no significant differences between the ensemble average of the low pass filtered velocity and the ensemble average velocities plotted
in Figure 3.4(a). A $\chi^2$ value of 0.8 is obtained for both the $u$ and $v$ velocity components. The differences between the ensemble and cyclic based averaging techniques become evident when the fluctuating velocities are compared. The local large scale, high energy events at 30–40° and 60–80° seen in Figure 3.4(b) are not present in Figure 3.4(c).

Comparison is made between the ensemble and cyclic based turbulence velocities calculated from Equation 3.3 and plotted in Figure 3.6. As expected, the ensemble based turbulence velocities are greater than the cyclic based results. The cyclic based turbulence velocities are lower in magnitude, but still follow the same general pattern as the ensemble based results: an initially high turbulence velocity begins to decay before rising and falling again. In both cases, the $v$ component turbulence velocity is 10–15% larger than the $u$ component of the turbulence velocity.

![Graphs](image)

(a) Ensemble average based filtering  
(b) Cyclic based filtering. Low pass filter at 120 (engine cycle)$^{-1}$.

Figure 3.6: Turbulence velocity after filtering. Motored operation at $\bar{T} = (0, -25, 15)$. 900 RPM and $N = 200$ cycles.

The main problem with cyclic averaging techniques arises from applying a constant low pass filter to data with a non-stationary convection and turbulence velocity. As the convection and turbulence velocity vary with crank angle, the frequency associated with a length scale will also change. More than one length scale can be associated with a single frequency. Whether the filtering scale is defined in the frequency or
crank angle domain is irrelevant. The physical interpretation of what scales are being filtered is difficult and it is uncertain whether the energy near the cut-off frequency is associated with large or small scale motions. Subsequent calculations of auto-correlation, power spectral density functions or integral time scales are biased.

3.2.3 Identification of quasi-periodic bulk flow variations

Neither the ensemble nor cyclic based techniques is entirely satisfactory in decomposing engine flows because of the biases they introduce into determination of the turbulence statistics. Ideally, the technique used to decompose engine flows should be able to separate the quasi-periodic bulk flow variations from the turbulence. Conditional ensemble averaging could then be used to remove the biases introduced by the quasi-periodic bulk flow events. The investigation of coherent structures in stationary turbulence has focused on the identification of energetic, quasi-periodic events. The techniques used to identify coherent structures in stationary flows can also identify the quasi-periodic events in engine flows. Most of these methods of structure identification, however, require multi-point measurements. For example, Hayakawa and Hussain (1987, 1989) identify coherent structures using multi-point velocity measurements. These techniques cannot, however, be applied to single point LDV velocity measurements.

For single point velocity measurements, Higuchi et al. (1994) have shown wavelet analysis to be a very effective method of identifying coherent, quasi-periodic events. Wavelet basis functions are used to decompose the flow energy over the frequency-crank degree plane: details of wavelet basis function properties are outlined in Appendix D. Wavelet basis functions are effective in identifying quasi-periodic events because they can represent transient events locally in the crank degree-frequency plane. This is in contrast to Fourier basis functions which require a large number of coefficients to model a transient event in the crank domain. Wavelet based decomposition methods have been successfully applied to single point data in a number of flows including the wake of two flat plates by Higuchi et al. (1994) and the three-dimensional wall jet by Sullivan and Pollard (1996).
Wiktorsson et al. (1996) and Sullivan et al. (1999) applied wavelet based filtering to SI engine flows. Their technique, however, effectively divides the flow into high and low frequencies, and is the equivalent to the cyclic based technique in that a filtering scale is introduced; the wavelet based decomposition is, however, more efficient numerically.

In an attempt to look at the turbulent flow without introducing a filtering scale a new technique to decompose the flow into mean and turbulence, based on discrete-time wavelet transforms, is proposed. It uses an energy based filtering technique to separate high energy events within the turbulence. This procedure is similar to that proposed by Lancaster et al. (1976) where the turbulence velocity calculated by ensemble averaging was interpreted to consist of low and high frequency components associated with the cyclic variations in the mean and turbulence, respectively. Here, the ensemble average turbulence velocity is interpreted to consist of high and low energy events, where the high energy events are associated with the quasi-periodic variations in the mean. The detailed development and preliminary results from this technique are presented in Appendix E.

### 3.3 In-cylinder pressure analysis

Next, the better established techniques used to calculate the characteristic engine operating and mass burn rate parameters from the measured in-cylinder pressure are described. The characteristic engine operating parameters, such as the location of peak pressure and indicated mean effective pressure, are obtained directly from the pressure trace and are used to characterise the performance of the engine. For example, IMEP characterises the amount of work available in a particular cycle. The characteristic mass burn rate parameters, such as the final mass burn fraction and the 0-5% mass burn duration, are calculated using a thermodynamic analysis of the measured pressures: $r_{fb}$ and $\Delta\theta_{0.5\%}$ give a more detailed picture of the combustion process on an individual cycle basis. There are two zero-order models used to calculate the mass burn fraction as a function of CAD from the measured pressure data: the three-zone model of Krieger and Borman (1966) and the one-zone model of Cheung
and Heywood (1993). An overview of both models is presented and the advantages and disadvantages of each model discussed: the thermodynamic sub-models used to account for heat transfer, crevice and residual gas effects in the mass burn analysis are discussed in Appendix F.

The three-zone model of Krieger and Borman (1966) treats: (i) the compression phase as consisting of a homogeneous unburned charge: (ii) the combustion phase as an unburned and burned zone, where the burned zone is further subdivided into an adiabatic core and boundary layer across which heat transfer from the burned zone to the cylinder walls occurs, and (iii) the expansion phase as a continuation of the combustion phase, with no mass transfer between the unburned and burned zones. The three-zone model ignores mass transfer, and, therefore, does not account for crevice effects: crevice effects are discussed in Appendix F.2. Figure 3.7(a) contains a schematic of the three-zone model during the combustion phase: the pressure is assumed to be uniform throughout all three zones. The thermodynamic properties of the burned and unburned gases are evaluated using best fit curves to the actual thermodynamic data, where temperature is the only independent variable. The ideal gas law is used to evaluate the mixture thermodynamic properties.

The one-zone model was developed by Cheung and Heywood (1993) to perform real time pressure analysis to replace the Rassweiler and Withrow model in combustion analysers. The development goals were to improve the accuracy of the mass burn rate calculations relative to the Rassweiler and Withrow model while maintaining the high computational speed needed for real-time analysis. The one-zone model treats the in-cylinder charge as a homogeneous mixture at the measured pressure. The one-zone system is open to heat and mass transfer. Since the one-zone model is open to mass transfer, crevice effects can be modelled, as shown in Figure 3.7(b). The thermodynamic properties of the mixture are calculated based on the ratio of specific heats where the ratio of specific heats during the compression, combustion
Figure 3.7: Schematic representation of the one and three zone models.

(a) Three-zone system is open to heat transfer, but closed to mass transfer.

(b) One-zone system is open to both mass and heat transfer.
and expansion phases are modelled as

\[ \gamma_{\text{comp}} = a_e T + b_e \]
\[ \gamma_{\text{comb}} = a_b \]
\[ \gamma_{\text{exp}} = a_c T + b_c. \]  

(3.5)

respectively. The specific heat ratio models of Equation 3.5 are based on the observed variation, by Chun and Heywood (1987), of the specific heats with respect to the mass averaged temperature during the compression, combustion and expansion phases (Figure 3.8). Values for \((a_e, b_e), (a_b)\) and \((a_c, b_c)\) are obtained by matching the one- and three-zone mass burn profiles when the input pressure trace is generated by an engine simulation programme where there are neither heat losses nor crevice effects.

Figure 3.8: Ratio of specific heats as a function of the mass averaged in-cylinder gas temperature as calculated by the three-zone model for iso-octane at \(\lambda = 1.0, r_e = 8.5\) and MAP=100 kPa.

The main differences between the one- and three-zone models are:

1. The thermodynamic property evaluation is more accurate for the three-zone model.

2. Crevice effects are readily incorporated in the one-zone model, whereas crevice effects cannot be easily incorporated into the three-zone model. This is because
Table 3.3: Engine test points used to compare the one and three zone thermodynamic models. GM V6 turbo-charged 3.1 L engine with 2.8 L cam shaft fuelled with natural gas was used for the tests.

<table>
<thead>
<tr>
<th>Test point</th>
<th>λ</th>
<th>Speed (RPM)</th>
<th>Spark timing (deg. ATDC)</th>
<th>Load (N·m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>XACTSRT1</td>
<td>1.53</td>
<td>2000</td>
<td>-50</td>
<td>65</td>
</tr>
<tr>
<td>YA301491</td>
<td>1.49</td>
<td>1200</td>
<td>-30</td>
<td>40</td>
</tr>
<tr>
<td>ZA401571</td>
<td>1.57</td>
<td>1500</td>
<td>-40</td>
<td>130</td>
</tr>
<tr>
<td>ZA401531</td>
<td>1.53</td>
<td>1500</td>
<td>-40</td>
<td>130</td>
</tr>
</tbody>
</table>

The thermodynamic properties of the mass flowing into the crevices in the one-zone model are unambiguously the thermodynamic properties of the one-zone system: this is not the case for the three-zone model.

3. The computation time for the three-zone model is approximately 10 s per engine cycle on a 133 MHz Pentium, an order of magnitude longer than that of the one-zone model. This is significant when hundreds of engine cycles are analysed.

The choice of thermodynamic model to use in the mass burn analysis depends on whether the inaccuracies introduced by the thermodynamic property approximations in the one-zone model are out-weighed by inaccuracies introduced by ignoring crevice effects in the three-zone model. To help in making the decision, in-cylinder pressure measurements were made in a GM V6 3.1 l turbo-charged engine. The engine specifications are equivalent to those listed in Table 3.1 for the SCV6 non-optical 3.1 L build engine: details of the system set-up are documented by Burcar (1996). Based on the diagnostic tests, as discussed in section 3.1.3, the quality of the pressure data was deemed to be high. The results from the one- and three-zone models are compared at the four engine test points listed in Table 3.3. Typical individual mass burn profiles are shown in Figure 3.9 and the average $x_{bf}$, $\Delta \theta_{0-5\%}$, and $\Delta \theta_{0-50\%}$ are listed in Table 3.4.

The results from the low load and speed condition, test point YA301491, show insignificant differences between the one- and three-zone mass burn profiles and the mass burn analysis statistics, see Figure 3.9(b) and Table 3.4 respectively. There are,
Table 3.4: Comparison of results from the one- and three-zone model. GM V6 turbo-charged 3.1 l engine with 2.8 l cam shaft fuelled with natural gas was used for the tests.

<table>
<thead>
<tr>
<th>Test point</th>
<th>( x_{bf} )</th>
<th>( \Delta \theta_{0-5%} ) (CAD)</th>
<th>( \Delta \theta_{0-50%} ) (CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>XACTSRT1</td>
<td>0.94</td>
<td>14</td>
<td>30</td>
</tr>
<tr>
<td>YA301491</td>
<td>0.82</td>
<td>30</td>
<td>58</td>
</tr>
<tr>
<td>ZA401571</td>
<td>0.91</td>
<td>15</td>
<td>40</td>
</tr>
<tr>
<td>ZA401531</td>
<td>0.91</td>
<td>15</td>
<td>32</td>
</tr>
</tbody>
</table>

however, significant differences at the remaining three test points: these three tests points are at higher load and more advanced spark timing conditions.

The individual cycle mass burn profiles, as shown in Figures 3.9(a), 3.9(c) and 3.9(d), indicate that there are problems with the one-zone model results. The one-zone mass burn profiles show an early rise and late overshoot in the mass burn profile when compared with the three-zone mass burn profiles. The early rise in mass burn fraction is unexpected during the early flame-kernel growth period especially for natural gas, as it has a relatively low laminar flame speed. The early rise in the one-zone model is attributed to the dependence of \( \gamma_{comb} \) on temperature during the early stages of combustion. This is seen as the variation in the \( \gamma_{comb} \) with respect to the mass averaged temperature in the range of 500–1000 K shown in Figure 3.8.

The assumptions used to develop the one-zone model are invalidated, probably due to the lower mass averaged temperatures at more advanced spark timing. Cheung and Heywood (1993) report that the best-fit parameters were not sensitive to spark timing: results with spark advances of > 30° were not, however, presented. The decay in the final mass burn fraction has been attributed by Cheung and Heywood (1993) to thermal shock effects. This would explain the increase in overshoot with increasing load. This conclusion, however, is not supported here as the three-zone model does not show a similar trend.

It is concluded that the one-zone model does not perform adequately for the SI engine mass burn analysis programme, and that the three-zone model should be used
for the mass burn analysis. Further investigation into the early rise and late decay in the one-zone model mass burn profiles is warranted.

![Graphs of mass burn analysis](image)

(a) Engine test point XACTSTR1  
(b) Engine test point YA301491  
(c) Engine test point ZA401571  
(d) Engine test point ZA401531

Figure 3.9: Typical individual mass burn profiles calculated by the one- and three-zone models for natural gas in the GM 3.1 l V6 turbo-charged engine.

### 3.4 Early flame kernel development analysis

The analysis procedures used to characterise the EFKD period from the FOSP measurements are now described. Specifically, the method by which the eight flame arrival times are determined, and the models used to calculate the characteristic EFKD parameters $S_g$ and $\tilde{V}_c$ are outlined. $S_g$ is the mean expansion rate of the flame kernel, and $\tilde{V}_c$ is the velocity at which the flame kernel is convected. The two models generally used to calculate $S_g$ and $\tilde{V}_c$ from FOSP data are the cubic spline model of Bianco et al. (1991), and the ellipse model of Kerstein and Witze (1990). Both models are
outlined in Appendix G. Lord et al. (1993), however, showed that both the cubic spline and the ellipse models gave unexpectedly high flame kernel expansion rates when investigating the early flame kernel development in engines with high swirl in-cylinder flows. This is of concern here because the SCV6 GM 3.1 l and 2.8 l engines have relatively high swirl.

Lord et al. (1993) noted that problems were first observed when the mean flame kernel convection velocities exceeded 5 m s⁻¹. The legitimacy of the large \( S_g \) values under these high convection conditions came into question. The problem manifests itself as early flame arrival times at the fibre-optic probes downstream in the convection direction. The problem was further investigated by Lord et al. (1993) using a bench top simulation of the ignition event outside the engine under strong convective flows. The early flame arrival times were observed once the mean convection velocity over the spark plug was increased above 8 m s⁻¹. Specific information regarding the type of spark plug used in the investigation was not provided. Lord et al. (1993) concluded that the spark kernel was highly stretched and that the analysis models could not account for this phenomenon.

In later work, Kim and Anderson (1995) developed a spark anemometry technique, based on the method of Maly, Meinel, and Wagner (1983), to measure the mean gas convection velocity in the spark gap. They measured the ignition system secondary voltage and current in a bench top simulation of the SI engine ignition process. The results indicated that ignition system restrikes were observed once the bulk gas velocity over the spark plug increased above 6 m s⁻¹: the frequency of restrikes increased as the bulk gas velocity increased. A Motorcraft AWF34C spark plug with a gap of 1.27 mm was used in their study. An ignition system restrike implies that a new flame kernel has formed in the spark gap. The fate of the original stretched flame kernel is indeterminate.

Based on the results of Kim and Anderson (1995) it was hypothesised that the problem observed by Lord et al. (1993) may be marked by restrikes. The present work was carried out: (i) to establish that restrikes occur in GM 2.8 l production engines; (ii) to find a solution to the FOSP data analysis problem in high swirl engines; and
(iii) to provide a preliminary examination of the interaction between the characteristic FOSP and combustion parameters in the GM 2.8 l and high swirl 3.1 l engines before installing the Bowditch piston for optical access is presented next.

### 3.4.1 Experiments into FOSP measurements in high swirl engines

Simultaneous in-cylinder pressure and FOSP measurements were made in the single cylinder GM 3.1 l engine. These tests were followed by simultaneous pressure, FOSP and ignition system secondary voltage measurements in the SCV6 GM 2.8 l engine. The engine specifications are listed in Table 3.1 and the nine engine test points are listed in Table 3.5. The engine test points are all at maximum brake torque and were chosen to cover a range of engine speeds and loads. A GM high energy ignition system was used with a glow discharge phase duration of \( \approx 1.5 \) ms. The resulting mass burn analysis statistics for the test points are listed in Table 3.6 and indicate that the 2.8 l engine has a slower burning combustion chamber relative to the 3.1 l engine.

A high voltage Model 6015B Tektronics probe was used to measure the ignition system secondary voltage. This probe attenuates the signal by a factor of 1000 and has a 4.5 ns response time. The ignition system secondary voltage was not acquired using the NI-DAQ system because of sampling rate limitations: the AT-MIO-16E-1 can only sample at 1.25 MHz or 85 kHz per channel, whereas sampling rates >100 kHz per channel were needed to capture the bandwidth of the ignition system secondary voltage variations during the glow phase. A Model HP34615B Hewlett-Packard digital oscilloscope was used to acquire the data. The same exhaust TDC trigger used to trigger the NI-DAQ system was used as an external trigger for the HP-DAQ system: 5000 data points were acquired at a 1 MHz sampling rate and were written out to files for post-processing. The actual start of the HP-DAQ was delayed by a preset amount so that the glow discharge phase of the ignition process was centred on the scope.

Tests were carried out to ensure that the individual cycle data acquired by the NI-DAQ and HP-DAQ systems was from the same engine cycle. The tests involved
acquiring fired in-cylinder pressure data on both DAQ systems. The correlations between the individual cycle peak pressure were >0.995 for all test cases. Since there were significant cycle to cycle variations in peak in-cylinder pressures, it was concluded that the NI-DAQ and HP-DAQ systems were acquiring data from the same engine cycle.

Table 3.5: Engine test points.

<table>
<thead>
<tr>
<th>Cylinder head</th>
<th>Test point</th>
<th>No. of engine cycles</th>
<th>λ</th>
<th>MAP (kPa)</th>
<th>Speed (rpm)</th>
<th>Spark timing (ATDC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1 l</td>
<td>F120020</td>
<td>100</td>
<td>1.00</td>
<td>95</td>
<td>1200</td>
<td>-20</td>
</tr>
<tr>
<td></td>
<td>F126528</td>
<td>100</td>
<td>1.00</td>
<td>55</td>
<td>1200</td>
<td>-28</td>
</tr>
<tr>
<td></td>
<td>F124540</td>
<td>100</td>
<td>1.00</td>
<td>35</td>
<td>1200</td>
<td>-40</td>
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<tr>
<td></td>
<td>F180025</td>
<td>100</td>
<td>1.00</td>
<td>85</td>
<td>1800</td>
<td>-25</td>
</tr>
<tr>
<td></td>
<td>F186529</td>
<td>100</td>
<td>1.00</td>
<td>55</td>
<td>1800</td>
<td>-29</td>
</tr>
<tr>
<td>2.8 l</td>
<td>G120025</td>
<td>100</td>
<td>1.00</td>
<td>90</td>
<td>1200</td>
<td>-25</td>
</tr>
<tr>
<td></td>
<td>G126530</td>
<td>100</td>
<td>1.00</td>
<td>55</td>
<td>1200</td>
<td>-30</td>
</tr>
<tr>
<td></td>
<td>G180030</td>
<td>90</td>
<td>1.00</td>
<td>85</td>
<td>1800</td>
<td>-30</td>
</tr>
<tr>
<td></td>
<td>G186535</td>
<td>100</td>
<td>1.00</td>
<td>55</td>
<td>1800</td>
<td>-35</td>
</tr>
</tbody>
</table>

Table 3.6: Characteristic engine operating and mass burn parameters.

<table>
<thead>
<tr>
<th>Test point</th>
<th>ηe (%)</th>
<th>IMEP (kPa)</th>
<th>COV_{IMEP} (%)</th>
<th>LPP (CAD)</th>
<th>x_{bf}</th>
<th>Δθ_{0-1%} (CAD)</th>
<th>Δθ_{0-5%} (CAD)</th>
<th>Δθ_{0-50%} (CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>F120020</td>
<td>66</td>
<td>722</td>
<td>1.5</td>
<td>17.2</td>
<td>0.92</td>
<td>11.9</td>
<td>16.7</td>
<td>31.0</td>
</tr>
<tr>
<td>F126528</td>
<td>36</td>
<td>403</td>
<td>1.0</td>
<td>14.8</td>
<td>0.90</td>
<td>15.4</td>
<td>20.8</td>
<td>37.6</td>
</tr>
<tr>
<td>F124540</td>
<td>22</td>
<td>241</td>
<td>3.3</td>
<td>12.2</td>
<td>0.87</td>
<td>21.3</td>
<td>27.6</td>
<td>45.6</td>
</tr>
<tr>
<td>F180025</td>
<td>62</td>
<td>697</td>
<td>1.8</td>
<td>18.4</td>
<td>0.94</td>
<td>15.2</td>
<td>20.7</td>
<td>38.0</td>
</tr>
<tr>
<td>F186529</td>
<td>36</td>
<td>406</td>
<td>1.9</td>
<td>16.0</td>
<td>0.91</td>
<td>16.9</td>
<td>22.5</td>
<td>39.7</td>
</tr>
<tr>
<td>G120025</td>
<td>74</td>
<td>764</td>
<td>1.2</td>
<td>16.8</td>
<td>0.94</td>
<td>13.2</td>
<td>18.4</td>
<td>35.4</td>
</tr>
<tr>
<td>G126530</td>
<td>41</td>
<td>417</td>
<td>1.3</td>
<td>16.9</td>
<td>0.85</td>
<td>18.7</td>
<td>22.6</td>
<td>40.9</td>
</tr>
<tr>
<td>G180030</td>
<td>68</td>
<td>729</td>
<td>1.6</td>
<td>16.5</td>
<td>0.84</td>
<td>15.4</td>
<td>21.3</td>
<td>40.4</td>
</tr>
<tr>
<td>G186535</td>
<td>40</td>
<td>424</td>
<td>1.4</td>
<td>16.7</td>
<td>0.85</td>
<td>19.5</td>
<td>26.4</td>
<td>47.3</td>
</tr>
</tbody>
</table>

3.4.2 FOSP data analysis

Simultaneous in-cylinder pressure-FOSP data was first acquired in the single cylinder 3.1 l engine and the cubic-spline model data processing technique was applied. The
cubic-spline model constructs a 2-D flame contour of the 3-D flame kernel from the eight flame arrival times and is detailed in Appendix G.1. The EFKD flame growth rates were calculated relative to the 2-D flame contour centroid, based on the technique of Lord et al. (1993). Unexpectedly large flame growth rates were obtained in a large number of individual engine cycles. The mean expansion speed is expected to be $O(S_L(\rho_u/\rho_b))$, where $S_L$ is the unstretched laminar flame speed. Mean expansion rates 4-5 times larger than this value might be expected due to enhancement of the burn rate via turbulence and flame front stretching. For natural gas fuelled engines $S_L$ is $O(0.3) \text{ m s}^{-1}$ for $\lambda = 1$ at the time of ignition. $\rho_u/\rho_b \approx 4$, putting an upper limit on $S_g$ of 10 m s$^{-1}$.

A closer look at the data revealed similar early flame arrival problems to those observed by Lord et al. (1993). Early flame arrivals, EFA, are defined when the flame arrival time is equal to the enable time delay. Enable time delays are imposed to prevent triggering of the fibre-optic probes off light reflected from the ground electrode. Enable time delays of 0.25 ms are usually used. The data from test point G120025 is used to demonstrate the influence of the early flame arrivals on the calculated flame expansion speed. The flame contours for the engine cycles 1 and 8 are plotted in Figures 3.10(a) and 3.10(b), respectively, and the characteristic EFKD parameters for these two engine cycles are listed Table 3.7. The 13.4 m s$^{-1}$ centroid base flame expansion rate of engine cycle 8 is unrealistically high. This engine cycle has early flame arrival at 3 fibres. In contrast, engine cycle 1 has a realistic flame expansion speed, and no early flame arrivals.

A possible solution to the high $S_g$ value problem is to give less weight to the early flame arrival times when calculating $S_g$. The perimeter based method calculates the flame expansion rate relative to the ignition point and is outlined in Appendix G. The perimeter based method gives greater weight to the later flame arrival times relative to the centroid based method. The results using both techniques for engine cycles 1 and 8 are given in Table 3.7. For engine cycle 1, where the flame kernel is nearly circular, the two techniques give similar results. However, for engine cycle 8, where the flame kernel is highly expanded, the perimeter-based method estimates the
Table 3.7: Individual engine cycle EFKD results from data set G1200025.

<table>
<thead>
<tr>
<th>Engine cycle</th>
<th>$V_c^2$ (m s$^{-1}$)</th>
<th>$\theta_c$ (deg.)</th>
<th>$S_g$ (m s$^{-1}$)</th>
<th>$F_D$ (-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.9</td>
<td>100</td>
<td>4.3</td>
<td>4.0</td>
</tr>
<tr>
<td>8</td>
<td>11.4</td>
<td>68</td>
<td>13.4</td>
<td>9.5</td>
</tr>
<tr>
<td>10</td>
<td>7.6</td>
<td>23</td>
<td>8.8</td>
<td>4.5</td>
</tr>
<tr>
<td>11</td>
<td>3.3</td>
<td>125</td>
<td>5.2</td>
<td>4.4</td>
</tr>
</tbody>
</table>

expansion rate to be 9.5 m s$^{-1}, 4$ m s$^{-1}$ lower than the centroid-based method.

The centroid and perimeter based method $S_g$ statistics are compared in Table 3.8 for three flame contour types.

1. *Expanded flame contour*. This category consists of engine cycles with 2 or more early flame arrivals. Figure 3.10(b) contains a plot of a typical flame contour for an engine cycle in this category. The flame kernel has expanded rapidly transverse to the convection direction, hence the term expanded flame contour.

2. *Stretched flame contour*. This category is defined for engine cycles with 1 early flame arrival time. Figure 3.10(c) contains a plot of a typical flame contour for an engine cycle in this category.

3. *Near circular flame contour*. This category contains all engine cycles with no early flame arrivals. Figures 3.10(a) and 3.10(d) contain plots of typical flame contours for engine cycles in this category.

For expanded flame contours the perimeter method gives an average flame expansion speed 4 m s$^{-1}$ lower than the centroid based method. The perimeter based estimate of $S_g = 8.7$ m s$^{-1}$ is a more reasonable value relative to the 12.5 m s$^{-1}$ centroid based estimate. For stretched flame contours the perimeter method estimates the average flame expansion speed to be 5.7 m s$^{-1}, 3$ m s$^{-1}$ lower than the centroid based method. There are statistically insignificant differences between the two techniques for near circular flame kernels.
Table 3.8: Conditionally averaged expansion rate, $\bar{S}_g$, as calculated using the centroid and perimeter methods for data set G120025. Error bounds are the 95% confidence intervals.

<table>
<thead>
<tr>
<th></th>
<th>Expanded</th>
<th>Stretched</th>
<th>Near circular</th>
</tr>
</thead>
<tbody>
<tr>
<td>Centroid (m s$^{-1}$)</td>
<td>$12.5 \pm 0.5$</td>
<td>$8.6 \pm 0.4$</td>
<td>$4.7 \pm 0.3$</td>
</tr>
<tr>
<td>Perimeter (m s$^{-1}$)</td>
<td>$8.6 \pm 0.4$</td>
<td>$5.8 \pm 0.3$</td>
<td>$4.8 \pm 0.2$</td>
</tr>
<tr>
<td>Engine cycles</td>
<td>25</td>
<td>46</td>
<td>29</td>
</tr>
</tbody>
</table>

The perimeter based technique is used to calculate $S_g$ in this work because realistic $S_g$ values are obtained under high convection conditions, and equivalent results to the centroid technique are obtained under quiescent conditions. The correlation between the centroid and perimeter based $S_g$ values is 0.94 for data set G120025. The convection velocity magnitude and direction are the same for the centroid and perimeter based methods.

**Choice of threshold voltage** A threshold voltage is used to determine when the flame arrives at a fibre-optic probe. Data set G120025 is used to demonstrate the effect of varying the threshold voltage on the characteristic EFKD parameters $S_g$ and $\bar{V}_r$. The variation of the mean convection velocity magnitude and the flame growth rate with respect to threshold voltage is shown in Figure 3.11. Note that the mean convection velocity magnitude is a simple average that does not take direction into account. The $\bar{V}_r$ magnitude starts near zero at 0 THV, passes through a maximum near 0.10 V and then decreases at a constant rate of $\approx 15$ m s$^{-1}$ per V. The average flame expansion speed starts high and drops rapidly until a threshold voltage of 0.2 V where $S_g$ also decreases at a rate of $\approx 15$ m s$^{-1}$ per V. The threshold voltage selected has a significant impact on the EFKD parameter statistics. A systematic method is needed to minimise threshold voltage selection biases.

One method found to work well was to examine the correlation of $|\bar{V}_r|$ and $S_g$ with the mass burn duration parameters $\Delta \theta_{0-1\%}$, $\Delta \theta_{0-5\%}$, and $\Delta \theta_{0-50\%}$. Figures 3.11(c) and 3.11(d) contain plots of the variation of the correlation coefficients with respect to threshold voltage. The threshold voltage is chosen to maximise the magnitude of
(a) Engine cycle 1 - near circular flame kernel. \( \dot{V}_c = (1.9 \text{ m s}^{-1}, 100.4^\circ) \). \( S_g = 4.0 \text{ m s}^{-1}, \quad F_d = 1.05 \).

(b) Engine cycle 8 - expanded flame kernel. \( \dot{V}_c = (11.4 \text{ m s}^{-1}, 66.7^\circ) \). \( S_g = 9.5 \text{ m s}^{-1}, \quad F_d = 1.32 \).

(c) Engine cycle 10 - stretched flame kernel. \( \dot{V}_c = (7.6 \text{ m s}^{-1}, 22.7^\circ) \). \( S_g = 4.5 \text{ m s}^{-1}, \quad F_d = 1.31 \).

(d) Engine cycle 11 - near circular flame kernel. \( \dot{V}_c = (3.3 \text{ m s}^{-1}, 125.2^\circ) \). \( S_g = 4.4 \text{ m s}^{-1}, \quad F_d = 1.11 \).

Figure 3.10: Flame contours for cycles 1, 8, 10 and 11 of data set G120025 at 345 CAD. The spark timing is 335 CAD. \( THV = 0.25 \text{ V} \). \( \Delta t_{delay} = 0.25 \text{ ms} \). The EFKD parameters are calculated using the perimeter based cubic-spline model.
Figure 3.11: Variation of the characteristic early flame kernel development statistics with common threshold voltage. Test point G120025.
Table 3.9: Common threshold voltages and the associated mean flame arrival times. The threshold voltage $\bar{S}_g$ and $|\vec{V}_c|$ biases are estimated as the rate of change with respect to a $\pm 50$ mV threshold voltage variation. The mass burn fraction at $F.AT$ is estimated as $F.AT / \Delta \theta_{0-1\%}$.

<table>
<thead>
<tr>
<th></th>
<th>F120020</th>
<th>F126528</th>
<th>F124540</th>
<th>F180025</th>
<th>F186529</th>
</tr>
</thead>
<tbody>
<tr>
<td>THV (V)</td>
<td>0.25</td>
<td>0.40</td>
<td>0.50</td>
<td>0.35</td>
<td>0.35</td>
</tr>
<tr>
<td>$F.AT %$ (CAD)</td>
<td>9.2</td>
<td>9.2</td>
<td>13.0</td>
<td>12.0</td>
<td>10.0</td>
</tr>
<tr>
<td>$\Delta \theta_{0-1%} %$ (CAD)</td>
<td>11.9</td>
<td>15.4</td>
<td>21.1</td>
<td>15.2</td>
<td>16.9</td>
</tr>
<tr>
<td>$\approx x_b$ at $F.AT$ (%)</td>
<td>0.77</td>
<td>0.60</td>
<td>0.62</td>
<td>0.80</td>
<td>0.60</td>
</tr>
<tr>
<td>$\langle S_g \rangle$ bias (m s$^{-1}$ V$^{-1}$)</td>
<td>±0.5</td>
<td>±0.2</td>
<td>±0.2</td>
<td>±0.4</td>
<td>±0.3</td>
</tr>
<tr>
<td>$\langle</td>
<td>\vec{V}_c</td>
<td>\rangle$ bias (m s$^{-1}$)</td>
<td>±0.5</td>
<td>±0.3</td>
<td>±0.2</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>G120025</th>
<th>G126530</th>
<th>G180030</th>
<th>G1860035</th>
</tr>
</thead>
<tbody>
<tr>
<td>THV (V)</td>
<td>0.25</td>
<td>0.25</td>
<td>0.20</td>
<td>0.30</td>
</tr>
<tr>
<td>$F.AT %$ (CAD)</td>
<td>9.2</td>
<td>10.5</td>
<td>14.0</td>
<td>12.4</td>
</tr>
<tr>
<td>$\Delta \theta_{0-1%} %$ (CAD)</td>
<td>13.0</td>
<td>17.5</td>
<td>15.1</td>
<td>19.5</td>
</tr>
<tr>
<td>$\approx x_b$ at $F.AT$ (%)</td>
<td>0.70</td>
<td>0.60</td>
<td>0.93</td>
<td>0.65</td>
</tr>
<tr>
<td>$\bar{S}_g$ bias (m s$^{-1}$)</td>
<td>±0.4</td>
<td>±0.5</td>
<td>±0.4</td>
<td>±0.3</td>
</tr>
<tr>
<td>$\bar{V}_c$ bias (m s$^{-1}$)</td>
<td>±0.3</td>
<td>±0.5</td>
<td>±0.3</td>
<td>±0.3</td>
</tr>
</tbody>
</table>

The correlation for both $\langle S_g \Delta \theta_{0-1\%} \rangle$ and $\langle |\vec{V}_c| \Delta \theta_{0-1\%} \rangle$. For this particular data set, 0.25 V is a reasonable choice. Figure 3.11(b) contains a plot of the average flame arrival time against threshold voltage for data set G120025. At the threshold voltage of 0.25 V, the mean flame arrival time is 9.2 CAD compared to the average 0-1% mass burn duration of 13.0 CAD.

The remaining data sets behave in a similar manner. The threshold voltages which maximise $\langle S_g \Delta \theta_{0-1\%} \rangle$ and $\langle |\vec{V}_c| \Delta \theta_{0-1\%} \rangle$ are listed in Table 3.9 along with the mean flame arrival times and the average 0-1% mass burn durations. The selected threshold voltages are associated with mean flame arrival times between 60-95% of the respective 0-1% mass burn duration.

The threshold voltage bias is the result of uncertainty in the choice of threshold voltage. The threshold voltage $\langle S_g \rangle$ and $\langle |\vec{V}_c| \rangle$ biases are estimated from the rate of change of the respective EFKD parameter with respect to a $\pm 25$ mV threshold voltage variation. The biases are listed in Table 3.9 and range from ±0.02 to ±0.5 m s$^{-1}$ and are on the order of the statistical uncertainty: see Table 3.8 for a comparison.
### 3.4.3 Ignition system secondary voltage data analysis

The discussion of ignition system secondary voltages by Kim and Anderson (1995) with respect to the assessment of the convective flow behaviour at the spark gap during the EFKD period leads to the hypothesis that examining the ignition system secondary voltages might lead to further insights into the early flame kernel development in high swirl engines. A brief look at the measured ignition system secondary voltages is presented next.

First, the ignition system secondary voltages for engine cycles 1, 10 and 11 from data set G120025, displayed in Figure 3.12, are examined. The data from engine cycle 1, a near circular flame kernel with $|V_c| = 1.9 \text{ m s}^{-1}$, shows a common ignition system secondary voltage behaviour for an engine cycle with a near circular flame kernel in data set G120025. The initial spark discharge event occurs at 334 CAD where there is a rapid rise and fall in the secondary voltage. The actual breakdown voltage is not known because the HP-DAQ system is not configured to capture the details of the breakdown event: the voltage range on the HP scope was set at -300 to 500 V. There is an initial gradual rise and fall in the secondary voltage after the breakdown event, which is followed by a gradual rise in the secondary voltage until the glow mode ends at 346 CAD: the glow phase duration is $\approx 1.5 \text{ ms}$ for the ignition system used with this engine and the end of the glow discharge phase, $\theta > 346 \text{ CAD}$, is marked by large scale secondary voltage variations.

The secondary voltage data for engine cycle 11, defined as a near circular flame kernel with $|V_c| = 3.3 \text{ m s}^{-1}$, is plotted in Figure 3.12(d) and indicates the initial discharge event occurs at 334.5 CAD. This is followed by a gradual secondary voltage rise until a sharp drop in voltage occurs at 344 CAD. This saw-tooth shaped voltage trace is indicative of a restrike. The restrike occurs once the voltage required to maintain current flow through the flame kernel rises to the breakdown voltage of the spark gap. A new flame kernel is formed in the spark gap. The fate of the original flame kernel is unclear: it may be extinguished, blown off the spark plug, or the spark plug might act as a bluff body flame stabiliser. After the restrike, the secondary
(a) Engine cycle 1 - near circular with $S_d = 0.594$.  
(b) Engine cycle 8 - expanded flame contour with $S_d = 1.868$.  
(c) Engine cycle 10 - stretched flame contour with $S_d = 1.219$.  
(d) Engine cycle 11 - near circular with $S_d = 0.728$.  

Figure 3.12: Ignition system secondary voltages for engine cycles 1, 8, 10 and 11 of data set G120025. The spark timing is 335 CAD.
voltage begins to rise until the end of the glow phase at 347 CAD. The slope of the secondary voltage is approximately the same before and after the restrike.

The secondary voltage data from engine cycle 10, a stretched flame contour with \( |V_c| = 7.6 \) m s\(^{-1}\). is plotted in Figure 3.12(c) and shows restrikes during the glow discharge phase at 343, 347 and 350 CAD: the distinctive saw-tooth pattern is evident. The slope of the secondary voltage is approximately the same immediately before and after the first restrike at 343 CAD, whereas the slope after the second restrike at 347 CAD is significantly higher than before 347 CAD.

The physical justification for using the perimeter based method, which calculates the flame expansion rate relative to the ignition point to calculate \( S_g \), is the presence of restrikes. The implication of the restrikes is that the centre of growth for stretched flame kernels is at the ignition point, and not at the flame kernel centroid.

Kim and Anderson (1995) show that the rate of secondary voltage rise and the number of restrikes is related to the magnitude of the convection velocity. Their empirical model estimates the secondary voltage as

\[
V_p(t) = 40.46 \rho^{-0.32} p^{0.51} \left( 1 + \frac{2|V_c|}{d_g} \right)
\]

where \( V_p \) is the secondary voltage corrected for constant electrode fall voltages, \( d_g \) is the spark gap size (mm), \( i \) is the current (A), \( p \) is the in-cylinder pressure (kPa) and \( t \) is time (s). The shape of the spark kernel is assumed to be piecewise rectangular and gas temperature and composition effects are ignored. The secondary voltage increases with higher in-cylinder pressure, lower discharge current, an increase in the velocity, and an increase in time as the discharge stretches. Kim and Anderson (1995) also use a simplified model where they show that the influence of the convection velocity has the strongest influence on the secondary voltage. The simplifications reduce the relationship between the rate of change of secondary voltage and the convection velocity through the electric field \( E \) to

\[
|V_c| = \frac{\partial V_p/\partial t}{2E}
\]
As long as a good estimate for $E$ is used the variations in the electric field with respect to time have an insignificant influence on the convection velocity estimate. Kim and Anderson (1995) show this relation to hold in their bench-top simulation of the engine spark ignition event when the convecting flow is perpendicular to the spark gap. They found the frequency of restrikes to be proportional to the magnitude of the convection velocity. Note that the secondary voltage theoretically responds only to the flow velocity magnitude, independent of the flow direction. Any significant deviations from the linear relationship between the magnitude of the convection velocity and the slope of the secondary voltage may be the result of the discharge shape not following the idealised shape: Maly et al. (1983) show the shape of the spark kernel to be more irregular and fluctuating.

Assessment of the secondary voltage data from engine cycles 1, 10 and 11 allow for the following statements:

1. Based on constant secondary voltage slopes, engine cycles 1 and 11 have approximately constant convective velocity magnitude throughout the glow discharge phase.

2. Based on the change in slope after the second restrike in engine cycle 10, the convection velocity magnitude has changed after the second restrike. This holds true unless the electric field has gone through a significant change at this point.

3. Based on the number of restrikes, the increasing order of convection velocity magnitudes is engine cycle 1, 11, 10. This is in agreement with the FOSP results.

4. Through visual inspection, the maximum secondary voltage slope in the first 10 CAD of the glow discharge phase is 60, 70 and 50 V CAD$^{-1}$ for engine cycles 1, 11, and 10, respectively. If the electric field is approximately constant from one engine cycle to the next, these results would suggest that the increasing order of convection velocity magnitudes is 11, 1, 10 which would contradict the FOSP results. The electric field, therefore, more than likely varies significantly from one engine cycle to the next. Measurement of the ignition system secondary
current is needed to get a good estimate of the electric field. Unfortunately, the secondary current was not measured in these experiments.

5. The $S_y$ values for engine cycles 1, 11 and 10 are 4.0, 4.5 and 4.4 m s$^{-1}$, respectively. These are all reasonable values and the variations may be the result of the different amounts of flame stretch and turbulence from one cycle to the next. The influence of the restrikes in engine cycles 11 and 10 does not appear to be significant on the estimation of the flame expansion speed.

Next consider the secondary voltage data from engine cycle 8 which is plotted in Figure 3.12(b). Recall that engine cycle 8 has an expanded flame kernel. The secondary voltage rises rapidly after the initial spark discharge event at 335 CAD until a saw-tooth shaped restrike occurs at 339 CAD. The secondary voltage then begins to rise again, even more rapidly, and restrikes again at 340 CAD. This second restrike is not saw-tooth shaped. After the second restrike, no more restrikes occur: the glow discharge phase ends at 346 CAD.

The initial rapid rise in voltage suggests a high initial convection velocity, higher than that in engine cycles 1, 10 and 11. The cubic spline model estimates the magnitude of the convective velocity as 11.2 m s$^{-1}$, but this might be a low estimate because it is near the maximum value which can be calculated by the cubic-spline model with a 0.25 ms enable time delay. The rapid rise in voltage after the first restrike suggests that the second flame kernel is blown out of the spark gap immediately and that the ignition event must be re-initiated. Most surprising, though, is that no restrike events occur after the second restrike. The secondary voltage slope does not increase rapidly any longer. This strongly suggests that the high convective velocities are not present near the spark plug over the entire glow discharge phase.

Spark anemometry is a promising technique which can be used to characterise the flow field in the spark gap during the early stages of combustion. Measurement of both the secondary voltage and current is necessary. Further investigations into spark anemometry are, however, beyond the scope of this work. Next, the results from the FOSP are examined.
### 3.4.4 Non-optical build FOSP results

**Mass weighted velocity**

The convection velocity and expansion velocities measured during the EFKD period in the GM 2.8 l and 3.1 l single cylinder engines are now examined. Figures 3.13(a) and 3.13(b) contain plots showing the variation of the convection velocity magnitude with engine speed and the phase of the measurement, respectively: the convection velocity is normalised with respect to mean piston speed, $S_p$. No clear pattern emerges from the data. If, however, the FOSP measures the *mass-weighted velocity*, $|V_c| \propto (\rho V_c)$, the influence of the volumetric efficiency variations is still buried within these data: the volumetric efficiency data is contained in Table 3.6 in Section 3.4.2. The correction factor to account for the density variations is $\rho_u/\rho_o$ where $\rho_u = m_e/V(\theta_{ST})$ is the gas density at the time of ignition and $\rho_o$ is the gas density at intake bottom dead centre with 100% volumetric efficiency. Figure 3.13(c) shows the variation of $\bar{V}_c = (\rho_o |V_c|)/(\bar{S}_p \rho_u)$. The $\bar{V}_c$ is approximately independent of engine speed, indicating that the appropriate normalisation factor is $\bar{S}_p \rho_u/\rho_o$. Figure 3.13(d) contains a plot of $\bar{V}_c$ variations with respect to the phase of the measurement. The results indicate that $\bar{V}_c$: (i) is higher in the 3.1 L engine than in the 2.8 L engine, and (ii) decays with respect to CAD.

**Conditional Averages** The data from data sets G186535 and F180025 are separated, as discussed in section 3.4.2, based on flame contour type. These data sets were chosen because they showed the strongest correlations with LPP and $\Delta \theta_{0-15\%}$: the correlations were near -0.5. Conditional averages were calculated for the $S_g$ and $\bar{V}_c$. The results are listed in Table 3.10. This separation of data based on flame contour type clearly separates $|\bar{V}_c|$ and $S_g$ into high, mid, and low ranges. The conditional averages for LPP, $\Delta \theta_{0-15\%}$, and $\Delta \theta_{0-50\%}$ were also calculated. The results from both data points indicate that the LPP of the expanded flame contours is earlier than those of the stretched or near circular flame contours. The expanded flame contours also shorten the $\Delta \theta_{0-15\%}$ for data set F180025. The results indicate that the magnitude of
$S_g$ and $\bar{V}_c$ have some effect on overall engine performance. It should be kept in mind that there are factors other than $S_g$ and $\bar{V}_c$ which also affect the engine performance.

**Conclusions** The following important findings are obtained by this preliminary investigation

1. Restrikes occur in the GM 2.8 l single cylinder engine. No conclusions regarding whether or not restrikes occur in the 3.1 l can be made because ignition system secondary voltage data was not acquired in the 3.1 l engine.

2. The use of ignition system secondary voltages as an anemometry technique is promising. The secondary voltages indicate that the flow velocity around the spark plug can vary significantly during the glow phase. However, due to the influence of high in-cylinder turbulence on the secondary voltage, the development of spark anemometry requires further work. Simultaneous measurement of the ignition system secondary voltage and current along with in-cylinder velocity via LDV would be useful.

3. The flame growth rate should be calculated as the average expansion rate away from the ignition point. The justification is that under high convective flow conditions restrikes occur, where the restrikes keep the centre electrode as the reference point of the flame kernel expansion. In quiescent flows the difference between using the flame kernel centroid or ignition point as the expansion rate reference point is insignificant.

4. The choice of threshold voltage has a significant influence on the resulting EFKD parameters. The EFKD parameter threshold voltage bias ranges from $\pm 0.03$ to $\pm 0.5$ m s$^{-1}$.

5. The FOSP measures a mass-weighted convection velocity. Normalisation by the mean piston speed and density at the time of ignition accounts for a large part of the variation of $\bar{V}_c$ with respect to engine speed and load. The results indicate
Table 3.10: Conditionally averaged parameters for data set G120025. Error bounds are the 95% confidence intervals.

|               | $|\bar{V}_c|$ (m s$^{-1}$) | $\xi_g$ (m s$^{-1}$) | LPP (CAD) | $\Delta\theta_{0-1\%}$ (CAD) | $\Delta\theta_{0-50\%}$ (CAD) |
|---------------|----------------------------|----------------------|------------|-------------------------------|-------------------------------|
| G186535       |                            |                      |            |                               |                               |
| Expanded      | 10.2                       | 7.9                  | 16.4       | 19.4                          | 35.3                          |
| Stretched     | 7.4                        | 6.5                  | 17.2       | 19.6                          | 36.0                          |
| Near circular | 3.5                        | 5.4                  | 18.3       | 20.3                          | 35.8                          |
| All           | 9.3                        | 7.5                  | 16.7       | 19.5                          | 35.8                          |
| Error bound   | $\pm 0.4$                  | $\pm 0.3$            | $\pm 0.4$  | $\pm 0.35$                    | $\pm 0.8$                     |
| F180025       |                            |                      |            |                               |                               |
| Expanded      | 10.1                       | 9.6                  | 17.8       | 14.6                          | 47.0                          |
| Stretched     | 5.8                        | 6.1                  | 18.8       | 15.6                          | 48.2                          |
| Near circular | 1.9                        | 4.8                  | 18.9       | 15.8                          | 49.7                          |
| All           | 6.3                        | 7.2                  | 18.4       | 15.2                          | 47.3                          |
| Error bound   | $\pm 0.4$                  | $\pm 0.3$            | $\pm 0.4$  | $\pm 0.35$                    | $\pm 0.8$                     |

that the so-normalised $\bar{V}_c$ magnitude decays with respect to CAD during the compression stroke, and is higher in the 3.1 l engine.

6. Conditional sampling of the data with respect to flame contour type indicates that there can be a significant effect of flame contour shape on the engine operating or mass burn parameters for any one data set.

Summary  This chapter outlined the experimental apparatus used to measure: (i) the in-cylinder velocity using a two-component LDV system; (ii) the early flame kernel development using the fibre-optic instrumented spark plug, and (iii) the in-cylinder pressure. The analysis techniques used to characterise the in-cylinder flow field, the early flame kernel development period, and the mass burn rate and engine operating characteristics through the measured in-cylinder pressure, were also overviewed. In the next chapter, the results from the velocity, fibre-optic instrumented spark plug, and pressure measurements made in the optical single cylinder engine are presented. The interactions between the flow field and the natural gas combustion process are examined in detail.
(a) $|V_c|$ vs. engine speed. Normalising factor $= \bar{S}_p$

(b) $|V_c|$ vs. crank degree. Normalising factor $= \bar{S}_p$

(c) $\hat{V}_c$ vs. engine speed. Normalising factor $= \bar{S}_p \rho(\theta)/\rho_0$

(d) $\hat{V}_c$ vs. crank degree. Normalising factor $= \bar{S}_p \rho(\theta)/\rho_0$

Figure 3.13: Normalised convection velocity at the spark plug.
Chapter 4

Results and Discussion

The goal of this work is to obtain a better fundamental understanding of how the flow field influences the combustion process, and through the combustion process, the engine operating characteristics. To this end, three sets of measurements were made in the SCV6 3.1 L engine: stand alone motored and fired velocity measurements, and fired, simultaneous velocity-pressure and pressure-FOSP measurements. These types of measurements have been made in the past at other research facilities. Here, however, novel data processing techniques are used to obtain more insight into: (i) the character of the flow, and (ii) the influence of the flow on the combustion process. For example, the stand alone velocity measurements were made to characterise the general behaviour of the flow during the combustion period. The data are first examined using the standard velocity data processing techniques outlined in section 3.2, and then through the discrete wavelet transform, DWT. The DWT allows the in-cylinder energy transfer process to be visualised. The stand alone velocity results are presented in section 4.2. Similarly, the simultaneous velocity-pressure measurements were made to characterise the interaction between the flow field and the EFKD period. Again, the DWT is applied as a data processing tool in a novel way by looking at the correlations between the turbulent flow and the mass burn rates over the DWT plane. The results show the location in time and the length scales of turbulence energy which enhance the mass burn rate in the EFKD period. The simultaneous pressure-velocity results are discussed in section 4.4. Information regarding the influence of the EFKD period
on the engine operation is extracted from the simultaneous pressure–FOSP measurements. The results are presented in section 4.3. The FOSP data processing techniques developed in section 3.4.2 are used.

4.1 Measurements

The stand alone velocity, and simultaneous pressure–FOSP and velocity–pressure data were taken at nine operating points, listed in Table 4.1, in the SCV6 3.1 L engine. The engine was fuelled with natural gas at an excess air ratio of 1. Prior to acquiring these data, the baseline operation of the 3.1 L optical engine was characterised through pressure and emissions measurements. The purpose of these tests was three-fold: (i) the MBT spark timing at each of the operating points was located and are given in Table 4.1; (ii) the excess air ratio, as calculated from the emission measurements, was compared with the results from the excess air-fuel ratio sensor; the results agreed to within 3%, within the limitations of the equipment; and (iii) the mass burn analysis results were used as a baseline for comparison with subsequent measurements. The mass burn analysis results remained consistent over the baseline, simultaneous pressure–FOSP and simultaneous velocity–pressure measurements.

The range of speed and load conditions covered by the test matrix was constrained by physical limitations of the engine. For example, the Bowditch piston crown was found to overheat under WOT conditions at engine speeds above 1000 RPM. To circumvent this problem, an upper limit of 90 kPa MAP and 1200 RPM was imposed on the test range. A second problem encountered was the Bowditch piston expansion. As the engine warmed up, the piston expanded, thereby increasing the compression ratio over time, with the final value of the compression ratio depending on speed and load condition. The increase in the compression ratio over time meant that the operating characteristics of the engine also varied over time. This is of considerable concern when making the LDV measurements as they needed to be made in the first few minutes of engine operation before the window was coated with seed particles. Thus, in most cases, the LDV measurements were started before the engine was fully
Table 4.1: Test matrix for measurements made in the optical scv6 3.1 L engine. Listed spark timings are for MBT: $\lambda = 1.0$; fuelled with natural gas.

<table>
<thead>
<tr>
<th>Operating point</th>
<th>Speed (RPM)</th>
<th>MAP (kPa)</th>
<th>Spark advance (CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7A1</td>
<td>925</td>
<td>70</td>
<td>-20</td>
</tr>
<tr>
<td>7B1</td>
<td>1050</td>
<td>70</td>
<td>-21</td>
</tr>
<tr>
<td>7C1</td>
<td>1200</td>
<td>70</td>
<td>-20</td>
</tr>
<tr>
<td>8A1</td>
<td>925</td>
<td>80</td>
<td>-15</td>
</tr>
<tr>
<td>8B1</td>
<td>1050</td>
<td>80</td>
<td>-18</td>
</tr>
<tr>
<td>8C1</td>
<td>1200</td>
<td>80</td>
<td>-18</td>
</tr>
<tr>
<td>9A1</td>
<td>925</td>
<td>90</td>
<td>-16</td>
</tr>
<tr>
<td>9B1</td>
<td>1050</td>
<td>90</td>
<td>-16</td>
</tr>
<tr>
<td>9C1</td>
<td>1200</td>
<td>90</td>
<td>-18</td>
</tr>
</tbody>
</table>

stabilised. Appendix J contains more details of the problems encountered during the set-up of the scv6 3.1 L engine.

**Velocity data – choice of measurement location** The purpose of measuring the in-cylinder velocity was to investigate the influence of the flow field near the spark plug on the EFKD period. As these measurements are single point, the velocity measurement location was chosen carefully. Of paramount importance is (i) that the measurement location is close enough to the spark plug such that the measured velocity characterises the flow to which the flame kernel is exposed, and (ii) that the LDV data rates are high enough to obtain an accurate characterisation of the flow.

Johansson (1993) has shown that significant correlations between the mean and turbulence with the EFKD mass burn rate were found when measurements were made within 5 mm of the spark gap. With this 5 mm limit, motored velocity measurements were made in this work, at $\bar{r} = (15.0.0)$, (15.0.2.5) and (15.5.2.5) mm to determine the LDV data rates: this region in the combustion chamber was chosen because all three points are within 5 mm of the spark gap, where $\bar{r} = (15.0.0)$ mm is 4-5 mm below the spark plug. The motored data rates at (15.0.2.5) and (15.5.2.5) were significantly lower relative to $\bar{r} = (15.0.0)$ mm because these measurement points are closer to the cylinder head wall. Since $\bar{r} = (15.0.0)$ is within 5 mm of the spark
gap and the data rates are greatest at this point. The velocity measurements for all data sets were made at \( T = (15.0, 0.0) \).

4.2 Flow field in the 3.1 L single cylinder engine

The primary goal of this work is to determine how the flow field influences the combustion process. To achieve this end, the stand-alone velocity data are examined next. First, the results from applying standard velocity data processing techniques to the fired and motored data are presented. The mean and turbulent flow energy variations during the combustion process are characterised. Results from applying the DWT to the velocity data are then used to better visualise the in-cylinder energy transfer process. Regions in the DWT plane where larger scale structures break-up into smaller scales are highlighted.

4.2.1 The mean and turbulence velocities

Motored flow field

Examining the ensemble average mean velocities of the motored data in Figure 4.1(a,c,e), the variations in the mean velocity magnitudes are \(<2.5 \text{ m s}^{-1}\) in the 335-385 CAD range. The \( u \) component velocity is approximately constant and the \( v \) component velocity slowly increases in this crank degree window. Similar mean flow variations are seen at all engine speed and load conditions considered in this study. Note that uncertainty in the mean velocity in this crank degree window is \( \approx \pm 0.5 \text{ m s}^{-1} \), and the uncertainty in the turbulence velocity is \( \approx \pm 10\% \). Both uncertainties are at a 95% confidence level; the uncertainty levels are quantified in Appendix I.

The ensemble averaged turbulence energy data in Figure 4.2 is of the same order of magnitude as the mean flow energy at all operating conditions. For example, the average \( u \) component turbulence energy is about \( 3.0 \text{ m}^2 \text{ s}^{-2} \) at 925 RPM (Figure 4.2(a)) versus a \( u \) component mean flow energy of \( (1.0)^2 = 1 \text{ m}^2 \text{ s}^{-2} \) (Figure 4.1(a)). That the ensemble mean and turbulence flow energies are of the same order of magnitude
is expected in engine flows (Hill & Zhang, 1994). Note also that the turbulence energy decays with respect to CAD in the 335–385 CAD window. Runs and reverse arrangements tests, carried out as suggested by Bendat and Piersol (1986), indicate the decay is significant. Similar results are observed for the v component energy.

The cyclic turbulence energy for the motored engine, where the energy for $f > 28.8$ (engine cycle)$^{-1}$ is defined as turbulence, is plotted in Figure 4.3(a,c,e). The cyclic average turbulence energy (i) does not decay significantly with respect to CAD. (ii) is isotropic. (iii) varies little with respect to MAP; and (iv) increases with increasing engine speed. The dependence on engine speed is examined by plotting the turbulence energy $u^2 + v^2$ against mean piston speed $S_p$ in Figure 4.4. The results show that there is a linear dependence of turbulence energy on $S_p$. independent of load, at 70 and 90 kPa MAP. The turbulence energy appears to behave differently for the 80 kPa MAP operating points, where the increase in turbulence energy with respect to $S_p$ is low relative to the increase at the 70 and 90 kPa MAP operating points. The cause of this discrepancy is not immediately clear and needs to be investigated in future work.

Linear regression is performed, using the model $u'/S_p = C$. where the operating point 80 kPa MAP at 1200 RPM is excluded as an outlier. The results give $C = 0.72 \pm 0.02$ at a 95% confidence interval and $R^2 = 0.96$. Thus, we conclude that $u'/S_p$ is approximately constant for the SCV6 engine.

**Ensemble vs cyclic averaging results**  The average turbulence energy calculated by ensemble averaging are about 1.5 times that calculated by the cyclic averaging technique. The uncertainty in the turbulence velocity is $\approx \pm 10\%$ at a 95% confidence level: the uncertainty levels are quantified in Appendix I. Thus, the differences between the ensemble and cyclic averaging results are significant. This significant difference is expected in engine flows (Hill & Zhang, 1994).

**Comparison of the motored and fired flow fields**

The motored and fired mean velocities are compared in Figure 4.1 in columns 1 (a,c,e) and 2 (b,d,f), respectively. The velocity component normal to the flame front should
Figure 4.1: Ensemble average mean velocities. Motored results - a.c.e. Fired results h.d.f.
Figure 4.2: Turbulence energy. Ensemble averaging technique was used to calculate the turbulence. Motored results – a.c.e. Fired results – b.d.f.
Figure 4.3: Turbulence energy. Energy for $f > 28.8$ (engine cycle)$^{-1}$ is defined as turbulence. Motored results - a.c.e. Fired results - b.d.f.
Figure 4.4: Motored total average turbulence energy in 335–385 CAD range versus mean piston speed. Energy for $f > 28.8$ (engine cycle)$^{-1}$ is defined as turbulence.

increase as the flame approaches the measurement volume, followed by a reversal in direction after the flame front passage, as mandated by conservation of mass. In the measurement system the flame front approaches the measurement plane from above. Since the measurement point is directly below the spark plug, if the flame kernel propagates spherically centred on the spark gap, there should not be a large influence of flame passage on the $u$ and $v$ component velocities. Comparison of the motored and fired velocities, shown in Figures 4.1(a) and Figure 4.1(b), respectively, show, however, that the fired mean velocities first diverge from the motored results at $\approx 360$ CAD where the $u$ and $v$ component mean velocities first increase in the negative direction. At $\approx 370$ CAD, after the passage of the flame front, both velocity components increase rapidly in the positive direction. The results imply that the flame front approaches the measurement volume from the positive direction, and bisects the $u$ and $v$ component velocity directions. This is confirmed by the FOSP results, see section 3.4, where large flame kernel convection velocities are seen due to the swirling velocity flow field. Similar results are seen at all engine speed and load conditions.
The arrival of the flame front at the measurement volume can also be observed through the LDV data rates. The average LDV data rate at each crank degree is quantified as the total number of engine cycles with at least one data point. The average motored and fired LDV data rates, compared in columns 1 and 2 of Figure 4.5, show that the average motored data rates are constant throughout the 335–385 CAD window, whereas the average fired data rates decrease gradually, by a factor of 2–3, between 345–370 CAD. The decrease in LDV data rate is caused by a combination: (i) of the destruction of the seed particles as they pass through the flame front, and (ii) by the reduction in SNR ratio due to beam steering by index of refraction gradients at the thermal boundary layer of the quartz window. The flame front arrival time at the measurement point is, therefore, marked by the decrease in data rate. The extent of the CAD range over which the data rate drops is a measure of the cycle to cycle variations in the flame arrival time. The cyclic variations in the flame arrival times are caused (i) by variations in the turbulent flame speed resulting (a) from changes in the charge composition near the spark plug, and (b) from changes in the turbulent flow field from one cycle to the next; and (ii) by variations in the magnitude and direction of the large scale convection velocities, which serve to advect the flame kernel away from or closer to the measurement point, hence varying the distance the flame must burn before reaching the measurement point.

Conditional ensemble averaging based on flame arrival time should be used when calculating the turbulence velocity. That is, the engine cycles should be separated based on flame arrival times at the measurement volume, where only engine cycles with similar flame arrival times are ensemble averaged. The phase averaged turbulence velocities are different for each set of flame arrival times. For example, Foster and Witze (1988) mark the arrival of the flame at the measurement volume using a laser beam, and Whitelaw and Xu (1995) use an ionization probe. Both use conditional ensemble averaging on flame arrival times to calculate the turbulence velocity. If conditional ensemble averaging is not used, cyclic variations in the flame arrival times results in biased turbulence velocities. No specific technique was used here to mark the flame arrival time in these data sets, however, the CAD range where a sharp
Figure 4.5: Average data rates. Motored results - a.c.e. Fired results - b.d.f.
drop off in LDV data rates is observed for an individual cycle can be used as a flame arrival time marker.

If the ensemble average fired turbulence energy is computed without conditional averaging with respect to flame arrival time, a dramatic increase in the turbulence energy after TDC is seen (Figure 4.2). This is in marked contrast to the motored data results where there is a steady decay in the motored turbulence energy throughout the 335-385 CAD range. The increase in turbulence intensity after the passage of the flame front is in agreement with the work of others (Foster & Witze. 1988: Whitelaw & Xu. 1995). Prior to TDC the motored and fired turbulence energies are of the same magnitude; note the different ordinate scales of the motored and fired turbulence energy plots. If, instead, we use cyclic averaging, where the individual cycle mean is calculated from the flow energy below the cut-off frequency of 28.8 (engine cycle)$^{-1}$. Figure 4.3 shows that there is still a significant, but much lower, increase in turbulence energy after TDC.

Cyclic variations in the mean flow

The ensemble and cyclic fired turbulence velocities both show a significant increase in turbulence energy after TDC. The ensemble based results are especially difficult to interpret in the 360-370 CAD range, where spikes in the turbulence energy are observed. This increase in turbulence energy may be biased by the inclusion of cyclic variations in the mean. One contributing factor to the cyclic variations in the mean remaining as turbulence is the rapid change in velocity as the flame front passes through the measurement plane. This rapid change in velocity is represented by a broad bandwidth of Fourier frequencies. If the entire band of frequencies needed to represent the mean flow variations is not used, then energy from the mean flow remains as turbulence. However, turbulence energy is also lost to the mean flow by assigning all the energy below the cut-off frequency to the mean flow. This overlap of mean and turbulence flow scales is at the heart of the problem in defining turbulence levels within IC engine flows.

Physically, the cyclic variations in the flame arrival time and flame kernel convect-
tion velocity are the probable sources of the increased cyclic variations in the mean. The influence of cyclic variations in the flame arrival time is explained by imagining that the flame kernel convection velocity magnitude is zero, such that the centre of expansion of the flame front remains at the spark gap for each cycle. In this case, for cycles with different flame arrival times at the measurement volume, the phase of the changes in the $u$ and $v$ component velocities vary from one cycle to the next. The phase shift extends throughout the entire post-flame zone. Note that the phase shift would be a repeatable coherent event. This phasing bias implies that the velocity data should be conditionally averaged based on the flame arrival time, as done by Foster and Witze (1988) and Whitelaw and Xu (1995), and discussed previously in subsection 4.2.1. Conditional averaging was not done in this work because the flame arrival time at the measurement volume was not recorded.

Next, consider the influence of cyclic variations in the flame kernel convection, where the flame arrival times are assumed to remain constant, but the flame kernel is advected away from the spark gap by the bulk velocity. In this case the centre of the flame kernel is no longer at the spark plug. This type of flame kernel movement is expected in a high swirl engine. For example, the displacement of the flame kernels for the high swirl scv6 3.1 L non-optical build, displayed in Figure 3.10 on page 43, is evident. When there are cyclic variations in the flame kernel convection, the cyclic variations in the angle of approach of the flame front would vary the magnitude of the changes in the $u$ and $v$ component velocities as the flame front passed through the measurement volume. For example, in one cycle the flame front expansion velocity might add 10% to the $u$ component and 90% to the $v$ component, whereas in another cycle where the flame kernel approaches from a different angle, the reverse might hold. The effect of this would be that the apparent turbulence intensity in a particular cycle would be dependent on the mean flow process advecting the flame kernel. One way to get around this problem might be to conditionally ensemble average the velocity data based on both the flame arrival time and flame kernel convection direction, where simultaneous velocity–pressure–FOSP measurements could be used. That is, the engine cycles with similar flame arrival times and flame kernel convection directions
would be averaged together. A different ensemble averaged turbulence velocity would be obtained for each (flame arrival time, flame kernel convection direction) set of engine cycles. Conditional averaging was not done in this work because (i) the flame arrival time at the measurement volume was not recorded, and (ii) simultaneous LDV–FOSP data was not acquired.

4.2.2 Power spectral density functions

Next, the flow field characteristics are examined through the Fourier and wavelet based power spectral density functions. The Fourier based PSD function can be used to decompose the flow energy with respect to frequency: so that the spectral dynamics of the flow, that is, the cascade of energy from low to high frequency processes, can be seen. The energy exchange process in Fourier PSD functions are, however, averaged over all times at each frequency, and, as such, are most useful in stationary flows. The wavelet based PSD function, on the other hand, decomposes the flow energy with respect to crank degree and frequency, as described in Appendix D. This allows the evolution of the flow energy to be visualised over time and frequency. Only the PSD functions of the motored data are examined because: (i) the motored flow field prior to combustion was shown to be similar to the fired flow field; and (ii) the motored data rates are higher, which allows a better frequency characterisation of the flow field.

Fourier spectra decay rates

The Fourier PSD functions of the u component velocity are plotted in Figure 4.6. Note that the spectral energy decay depends on the operating point. The decay rates, calculated as the slope of the PSD function over \( f = 56.6 \rightarrow 300 \) (engine cycle\(^{-1} \), range from \(-0.5\) to \(-0.9\) and \(-0.35\) to \(-0.75\) for the u and v component velocities, respectively, and are plotted against mean piston speed and intake charge mass flow rate in Figure 4.7. The correlation coefficient between the decay rates and the abscissa values are listed in the sub-figure captions. The decay rates increase
with engine speed, that is, they become more negative. There is no significant effect of increasing mass flow rate. Note that the data point associated with (70 kPa, 1200 RPM) is an outlier in this data set. It is not clear why this operating point has significantly higher decay rates relative to the remaining operating points.

To interpret these results, compare the observed decay rates with the decay rate of \(-5/3\) expected in the inertial–advection sub range (Hinze, 1987). The inertial–advection sub range is associated with a range of frequencies in a temporal power spectra through which the energy cascades from the energy containing frequencies down through the dissipation frequencies: the low and high frequencies are assumed to be associated with large energy containing and dissipation scales, respectively\(^*\). The inertial–advection sub range exists only if there is a large separation in size between the large, energy containing scales and the dissipation scales. As the turbulence Reynolds number of the flow increases, the separation between these scales increases. An inertial–advection sub range is observed in flows with large turbulence Reynolds numbers. Since the decay rates seen here are less than \(-5/3\), it is concluded that the Reynolds numbers are not high enough to exhibit an inertial–advection sub range and that only the large scale spectrum is seen. Note, however, that as the engine speed is increased, Figure 4.7(a) shows that the observed decay rates become more negative, implying that the PSD functions at higher engine speeds might exhibit an inertial–advection sub range. This is not surprising since it was shown previously in this work that \(|\tilde{u}'| = 0.72S_p\) such that the turbulence Reynolds number of the flow increases with mean piston speed. These results agree with those of Catania et al. (1996), who did not find an inertial–advection sub range during the compression stroke in their engine flows at engine speeds \(\leq 2000\) RPM at a distance of 5 mm from the cylinder head, whereas an inertial–advection sub range was observed at 3000 RPM.

Note that although we do not see the frequencies at which dissipation occurs, aliasing biases are expected to be very low at \(f < 360\) (engine cycle)\(^{-1}\): there is a detailed discussion regarding aliasing in Appendix I.

\(^*\)Note that the \(-5/3\) slope is expected only in stationary flows. No similar \textit{standard} slope exists for non–stationary flows
Figure 4.6: Motored turbulence PSD functions. Energy for $f > 28.8$ (engine cycle)$^{-1}$ is defined as turbulence.
Figure 4.7: Motored turbulence PSD decay rates. Energy for $f > 28.8 \text{ (engine cycle)}^{-1}$ is defined as turbulence.
Figure 4.8: Ensemble averaged contour plots for motored turbulence at $f > 28.8$ (engine cycle)$^{-1}$ - operating point 7A1. The light shaded regions have more energy than dark shaded regions. The regions with energy cascades are highlighted. The third highlighted region is a cascade of energy deficit.

Discrete wavelet power spectral density functions

The energy cascade process can be examined through the calculation of the discrete wavelet power density spectra. One of the benefits of using the DWT is that the evolution of the flow field over time and frequency, or scale, can be visualised. To visualise the variation of energy in the crank degree-frequency plane, the contour plot of the ensemble average of the DWT energy is shown in Figure 4.8. To interpret these plots, note: (i) that the light coloured regions are associated with high energy and the dark regions with low energy, and (ii) the discrete time-frequency plane tiling is associated with a two-band wavelet. At low frequencies the discrete tiling shows a fine frequency resolution and a coarse time resolution: at high frequencies the discrete tiling shows a coarse frequency resolution, and a fine time resolution. For example, at $f = 28.8$ (engine cycle)$^{-1}$ there are four translations over the 51.2 CAD window: the translations are in integer steps of the $12.8$ CAD = $720/f$ scale associated with this frequency. At $f = 900$ (engine cycle)$^{-1}$ there are 64 0.8 CAD translations.

The contour plot of the DWT energy allows one to visualise the dynamic evolution of the distribution of energy with respect to frequency over CAD. It is possible to consider the distribution of energy at a particular crank degree over frequency as a
snap shot, or static, view of the energy distribution. Tracking this static view over CAD, one obtains a dynamic view of the distribution of energy with respect to crank degree. Alternately, it is possible to locate regions of high energy in the crank degree-frequency plane and track the evolution of energetic events. This alternate technique is used to describe the dynamics of the turbulence energy.

For example, it is possible to locate an energetic event in the u component contour plot (Figure 4.8(a)) at $f = 56.6$ (engine cycle)$^{-1}$ over the $\Delta \theta_b = 330-360$ CAD range. This energetic event disappears immediately after TDC; new energy, lower in magnitude, reappears at this frequency band in the $\Delta \theta_b = 375-385$ CAD range. The v component contour plot, Figure 4.8(b), is very similar in character to the u component contour plot.

The u component energy, as shown in Figure 4.8(a), is found to cascade from $(\Delta \theta_b = 334-346, f = 56.6)$ down to $(\Delta \theta_b = 335-345, f = 112.5)$ and then to $(\Delta \theta_b = 342.5-347.5, f = 250)$. A second cascade can be seen where the energy from $(\Delta \theta_b = 346-358, f = 56.6)$ appears to cascade down to $(\Delta \theta_b = 352-372, f = 112.5)$ and then to $(\Delta \theta_b = 355-375, f = 250)$ with a local peak at $(\Delta \theta_b = 365-367, f = 250)$. A region of energy deficit is seen immediately after TDC, where the energy from $(\Delta \theta_b = 350-360, f = 56)$ range drops, such that there is significantly less energy at $(\Delta \theta_b = 360-370, f = 56)$. This region of energy deficit extends down through $(\Delta \theta_b = 372-380, f = 112.5)$ and $(\Delta \theta_b = 375-385, f = 250)$. The v component contour plot shows cascades of energy from the same $(\Delta \theta_b, f)$ regions.

The u and v component energy cascades before TDC and the energy deficit immediately after TDC together suggest that, on average, there is steady transfer of energy before TDC from the large scales to the small scales. but a rapid destruction of the large scale structure occurs as the expansion process begins. A vortex instability might set in, such that, as the swirling structure associated with this scale is expanded, it breaks apart. This wouldn’t show up in the mean flow because the energy of this event is an order of magnitude less than that of the mean flow.

After the region of energy deficit, there is new energy introduced at $(\Delta \theta_b = 370-385, f = 56.6)$. This energy could have originated from larger scale processes breaking
down, or from the advection of energy from outside the measurement point. This new energy then begins to cascade through \((\Delta \theta_b=380-385. f = 112.5)\) and \((\Delta \theta_b=382.5-385. f = 225)\) the mid-range scales.

The u and v component DWT energy contour plots, ensemble averaged, of all nine operating points are similar in character to Figures 4.8(a) and 4.8(b). Specifically, they all have the same energy cascade prior to TDC, and the energy deficit region immediately after TDC.

**Discrete wavelet coherence functions**

We can also look at the flow through the DWT coherence function \(\gamma_{w,uv}(f, \theta)\) contour plot where \(\gamma_{w,uv}(f, \theta)\) is defined by.

\[
\gamma_{w,uv}(f, \theta) = \frac{\langle uv(f, \theta) \rangle^2}{\langle u^2(f, \theta) \rangle \langle v^2(f, \theta) \rangle}
\]  

(4.1)

The DWT coherence function \(\gamma_{w,uv}(f, \theta)\) is defined here in an analogous manner as the regular frequency domain coherence function \(\gamma_{uv}\) (Equation E.6). Areas in \((\theta, f)\) with high coherence would suggest coherent u and v motions which can be interpreted as being caused by the presence of organised structures. This would be a useful technique to employ, but there are insignificant \(C_{uv}(\theta)\) correlations in the motored data. It would be instructive to use this technique on the fired data, except that, unfortunately, the fired data rates are too low in the post-flame regions.

**4.2.3 Individual cycle wavelet spectra**

All the observations made in the ensemble averaged DWT energy maps are an average throughout the crank degree-frequency plane. Unless the events repeat themselves exactly from cycle to cycle, this ensemble averaging procedure will smear out the energy cascade process. For example, if the first cascade described above is characteristic of half the engine cycles then the averaging procedure will reduce the average energy of this event: the same holds true for the other cascade processes described. Individual cycle data are examined next to see if the energy cascade process can be
directly observed. The discrete wavelet spectra for a number of engine cycles from operating point 7A1 are examined (Figure 4.9). Three cycles were chosen to show typical wavelet spectra from individual cycle data.

The u and v DWT contour plots for cycle number 50 of data set 7A1 are plotted in Figures 4.9(a) and 4.9(b). The v component plot shows a high energy event ($\Delta \theta_b=335-347, f = 56.6$), the energy cascades down into ($\Delta \theta_b=347-352, f =112.5-250$). This energy cascade is associated with the first energy cascade region in the ensemble results: the u component has a moderate energy peak at ($\Delta \theta_b=352-360, f =112.5-250$): this energy might come from the v-component energy cascade. This energy cascade region in this cycle tracks the energy of a local high energy event over time and frequency. Note that only the first ensemble average energy cascade is realised in this cycle.

Notice also that u component has an energetic event ($\Delta \theta_b=360-372, f =56.6$): some of this energy cascades down to the ($\Delta \theta_b=372-378, f =112.5$) region in the u and v component velocities, but most simply disappears. Similarly, the high energy v component event at ($\Delta \theta_b=365-368, f =250$) appears from nowhere and disappears. This event is the result of energy being advected into and out of the measurement volume by the mean flow, and is a cyclic variation in the energy at ($\Delta \theta_b, f$). If the energy of this event is high relative to the other events in the ($\Delta \theta_b, f$) plane, then the event might be associated with a quasi-periodic variation of the mean flow. In this case, high energy events might rightfully belong with the mean.

Next, the DWT contour plots of cycle number 86, shown in Figures 4.9(c) and 4.9(d), are examined. The u and v component energies both show an energetic event at ($\Delta \theta_b=350-360, f =56.6$) and the subsequent transfer of this energy to ($\Delta \theta_b=360-370, f =112.5$). This cycle shows a local energetic event which breaks up after TDC into smaller scale. The energy transfer process associated with second energy cascade region of the ensemble results is seen in this engine cycle.

The contour plots of cycle number 694 are shown in Figures 4.9(e) and 4.9(f). The v component shows a clear energetic event at ($\Delta \theta_b=352-360, f =112.5-250$) with a similar, high energy event in the u component energy at the ($\Delta \theta_b=352-360, f =56.6$–
Figure 4.9: Individual cycle contour plots for motored fluctuating velocity at $f > 28.8 \text{ (engine cycle)}^{-1}$ – operating point 7A1. The light shaded regions have more energy than dark shaded regions.
region. This structure then proceeds to break up after TDC. This cycle shows the energy transfer process associated with the second energy cascade region of the ensemble results and is associated with the break up of a swirling vortex.

The individual cycle contour plots of Figure 4.9 indicate that there are significant cyclic variations in the energy distribution with respect to scale and frequency. The influence of cyclic variations in the distribution of the flow energy among the frequency–crank degree plane on the natural gas combustion process is examined in detail in section 4.4.2.

4.3 The influence of the early flame kernel development on engine operation

The simultaneous FOSP–pressure data were acquired to characterise the influence of the EFKD period on the SCV6 3.1 L optical engine operation. Of interest is how the EFKD period influences the cyclic variations in the engine operating parameters because these cycle to cycle variations are the limiting factors in improving SI engine operation. In this section, the EFKD period is first characterised through the mean expansion rate $S_g$ and flame kernel convection velocity $\vec{V}_c$. The relationship between $S_g$ and $\vec{V}_c$ in the 3.1 L SCV6 optical engine is then examined along with the influence of these two parameters on cyclic variations in the mass burn rate and the IMEP in the 3.1 L SCV6 optical engine.

4.3.1 $S_g$ and $\vec{V}_c$ in the optical 3.1 L SCV6 engine

The threshold voltages were selected using the criteria outlined in section 3.4. Table 4.2 lists the threshold voltages, mean flame arrival time, 0-1% mass burn duration and the approximate mass burn fraction associated with the mean flame arrival time. These threshold voltages were used to calculate the mean flame expansion speed $S_g$ and the flame kernel convection velocity $\vec{V}_c$ on an individual cycle basis.
Table 4.2: Common threshold voltages and the associated mean flame arrival times. $\Delta \theta_{0-1\%}$ is obtained from the mass burn analysis of the measured in-cylinder pressure. The mass burn fraction at $\overline{FAT}$ is estimated as $\overline{FAT} / \Delta \theta_{0-1\%}$.

<table>
<thead>
<tr>
<th>MAP (kPa)</th>
<th>RPM</th>
<th>THV (V)</th>
<th>$\overline{FAT}$ (CAD)</th>
<th>$\Delta \theta_{0-1%}$ (CAD)</th>
<th>$\approx x_b$ at $\overline{FAT}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>925</td>
<td>0.20</td>
<td>10.0</td>
<td>16.5</td>
<td>0.006</td>
</tr>
<tr>
<td>70</td>
<td>1050</td>
<td>0.32</td>
<td>11.9</td>
<td>16.9</td>
<td>0.007</td>
</tr>
<tr>
<td>70</td>
<td>1200</td>
<td>0.275</td>
<td>9.6</td>
<td>16.8</td>
<td>0.006</td>
</tr>
<tr>
<td>80</td>
<td>925</td>
<td>0.20</td>
<td>9.9</td>
<td>15.0</td>
<td>0.007</td>
</tr>
<tr>
<td>80</td>
<td>1050</td>
<td>0.32</td>
<td>11.1</td>
<td>15.6</td>
<td>0.007</td>
</tr>
<tr>
<td>80</td>
<td>1200</td>
<td>0.25</td>
<td>10.5</td>
<td>16.3</td>
<td>0.006</td>
</tr>
<tr>
<td>90</td>
<td>925</td>
<td>0.25</td>
<td>8.0</td>
<td>15.4</td>
<td>0.005</td>
</tr>
<tr>
<td>90</td>
<td>1050</td>
<td>0.25</td>
<td>8.4</td>
<td>14.3</td>
<td>0.006</td>
</tr>
<tr>
<td>90</td>
<td>1200</td>
<td>0.15</td>
<td>7.9</td>
<td>15.0</td>
<td>0.005</td>
</tr>
</tbody>
</table>

**Mean flame convection velocity**

To begin the analysis, it is possible to compare the results from the SCV6 3.1 L optical and non-optical builds. One of the main findings in the non-optical build was the dependence of $\langle |\overline{V_c}| \rangle$ on engine speed and gas density, see section 3.4. The same relationship is obtained in the optical build, namely $\langle |\overline{V_c}| \rangle = C_1 S_p \rho_n$, where $S_p$ is the mean piston speed and $\rho_n$ is the normalised density of the unburned gas at the time of ignition. The variations in the parameters $S_p$ and $\rho_n$ together account for 90% of the variance in $\langle |\overline{V_c}| \rangle$, versus <40% for either parameter on its own. The dependence of $\langle |\overline{V_c}| \rangle$ on $\rho_n$ implies that the FOSP measures a mass-weighted velocity.

To determine whether the FOSP measures a true mass-weighted velocity, the linear model, $\langle |\overline{V_c}| \rangle = C_1 S_p^{A} \rho_n^{B}$. was applied to the data, with the null hypothesis $A = B = 1$. The results are that $A$ is not significantly different from unity, but that $B = 0.65 \pm 0.22$ is significantly different than one. That $B = 0.65$ implies that the FOSP does not measure a true mass-weighted velocity, but a fairly high mass-weighted velocity. The appropriate model is therefore $\langle |\overline{V_c}| \rangle = C_1 S_p \rho_n^{0.65}$ and accounts for 94% of the variation in $\langle |\overline{V_c}| \rangle$.  

Table 4.3: Characteristic parameters at the time of ignition. Parameters \((a, b)\) are least square estimates via \(S_g = \alpha \vec{V}_c + b\)

<table>
<thead>
<tr>
<th>MAP</th>
<th>RPM</th>
<th>(x_{tr})</th>
<th>(\phi)</th>
<th>(S_L)</th>
<th>(S_p)</th>
<th>(\rho_n)</th>
<th>(\alpha)</th>
<th>(b)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>925</td>
<td>0.12</td>
<td>0.96</td>
<td>26.3</td>
<td>2.59</td>
<td>2.711</td>
<td>0.39 ± 0.02</td>
<td>2.44 ± 0.12</td>
</tr>
<tr>
<td>70</td>
<td>1050</td>
<td>0.11</td>
<td>0.95</td>
<td>25.5</td>
<td>2.94</td>
<td>2.559</td>
<td>0.35 ± 0.02</td>
<td>2.34 ± 0.17</td>
</tr>
<tr>
<td>70</td>
<td>1200</td>
<td>0.12</td>
<td>0.96</td>
<td>26.4</td>
<td>3.36</td>
<td>2.598</td>
<td>0.44 ± 0.02</td>
<td>3.30 ± 0.17</td>
</tr>
<tr>
<td>80</td>
<td>925</td>
<td>0.11</td>
<td>0.97</td>
<td>29.6</td>
<td>2.59</td>
<td>3.623</td>
<td>0.39 ± 0.02</td>
<td>2.43 ± 0.14</td>
</tr>
<tr>
<td>80</td>
<td>1050</td>
<td>0.10</td>
<td>0.96</td>
<td>29.9</td>
<td>2.94</td>
<td>3.293</td>
<td>0.43 ± 0.03</td>
<td>2.46 ± 0.24</td>
</tr>
<tr>
<td>80</td>
<td>1200</td>
<td>0.10</td>
<td>0.98</td>
<td>29.3</td>
<td>3.36</td>
<td>3.224</td>
<td>0.43 ± 0.03</td>
<td>3.01 ± 0.23</td>
</tr>
<tr>
<td>90</td>
<td>925</td>
<td>0.10</td>
<td>0.99</td>
<td>31.5</td>
<td>2.59</td>
<td>4.014</td>
<td>0.47 ± 0.02</td>
<td>2.79 ± 0.18</td>
</tr>
<tr>
<td>90</td>
<td>1050</td>
<td>0.09</td>
<td>0.99</td>
<td>31.6</td>
<td>2.94</td>
<td>3.806</td>
<td>0.49 ± 0.03</td>
<td>2.99 ± 0.23</td>
</tr>
<tr>
<td>90</td>
<td>1200</td>
<td>0.09</td>
<td>0.99</td>
<td>32.0</td>
<td>3.36</td>
<td>3.690</td>
<td>0.50 ± 0.03</td>
<td>3.71 ± 0.28</td>
</tr>
</tbody>
</table>

**Relationship between \(S_g\) and \(|\vec{V}_c|\) on an individual cycle basis**

The relationship between \(S_g\) and \(|\vec{V}_c|\) was examined on an individual cycle basis. A linear first order model was assumed to describe the relationship at each operating point. \(S_g = a \left| \vec{V}_c \right| + b\). The least squares estimates for \((a, b)\) are listed in Table 4.3. The parameter \(a\) describes the dependence of \(S_g\) on increasing \(|\vec{V}_c|\): the model \(a = C_2 S_L S_p\) was found to describe the variation of \(a\) with operating conditions with \(R^2 = 0.70\).

In the absence of a flame kernel convection velocity, that is, with \(|\vec{V}_c| = 0\), the variations in \(S_g\) with operating conditions are described by \(b\). Since the relationship between the flame expansion rate and turbulent flame speed is \(S_g/S_T = \rho_a/\rho_b \approx \) constant in the early stages of combustion, the parameter \(b\) also describes how the turbulent flame speed \(S_T\) varies with operating condition. Heywood (1988) notes that most of the turbulent flame speed models used are of the form \(S_T/S_L = 1 + C |\vec{n}| / S_L\), where the parameter \(C\) is constant and allows for different combustion chamber geometry. This empirical model forces \(S_T/S_L \rightarrow 1\) as \(|\vec{n}| \rightarrow 0\). The model chosen to describe the variations in \(S_g\) is, therefore, \((\rho_a/\rho_b)b - S_L = C_3 S_p\) and results in \(R^2 = 0.68\): \(C_3\) is expected to depend on \(u'\).

Based on these models, \(a = C_2 S_p S_L\) and \((\rho_a/\rho_b)b - S_L = C_3 S_p\), the relationship
between $S_g$ and $\Gamma_c$, over all nine operating points, is

$$\frac{S_g}{S_L} = 0.51 S_p \left| \overline{\Gamma_c} \right| + \rho_b \rho_u \left( 1 + \frac{0.56 S_p}{S_L} \right)$$

(4.2)

with $C_2 = 0.51 \pm 0.04$ and $C_3 = 0.56 \pm 0.06$. This model accounts for 74% of the variation in $S_g$ and shows that the EFDK mass burn rate, as quantified by $S_g$, is strongly dependent on the convection velocity magnitude at the spark plug. The magnitude of $S_g$ increases with $\left| \overline{\Gamma_c} \right|$ and is influenced by the unstretched laminar and mean piston speeds. The increase in $S_g$ with increasing $\left| \overline{\Gamma_c} \right|$ is due to: (i) reduced heat losses to the spark plug electrodes as the flame kernel is stretched out from the spark gap and hence larger $S_L$. (ii) enhanced aerodynamic strain such that there is a larger stretched $S_L$; and. (iii) possibly increased magnitude of $\overline{u'}$ with increasing $\left| \overline{\Gamma_c} \right|$. That the EFDK mass burn rate is strongly dependent on $\left| \overline{\Gamma_c} \right|$ agrees with the results of Ting et al. (1995), who found strong correlations between the mean convection velocity at the spark plug and the 0–0.5% mass burn duration.

Examining Equation 4.2 at $\left| \overline{\Gamma_c} \right| = 0$, it is clear that $S_T/S_L = (\rho_u/\rho_b)(S_g/S_L) = 1 + C_3 S_p/S_L$. Parameter $C_3$, which is independent of speed and load, is expected to depend on the turbulence intensity near the spark plug at the time of ignition. This implies that $C_3 = C(|\overline{u'}|/S_p)$, where $|\overline{u'}|/S_p = 0.72 \pm 0.02$ was shown to be independent of engine speed and load in section 4.2.1. Thus, the turbulent flame combustion rate model $S_T/S_L = 1 + 0.67 |\overline{u'}|/S_L$, where the $C = 0.67 \pm 0.04$ is the flame expansion rate enhancement factor during the EFDK period in the absence of a mean flame convection velocity; it is a characteristic parameter of the natural gas fuelled scv6 3.1 L optical engine. Ting et al. (1995) present similar results for methane–air flames at an excess air ratio of 1.1, where they plot the burning velocity $S_T/S_L - 1$ against the normalised turbulence intensity $u'/S_L$ for fully developed turbulent flames in flows with different integral length scales. Their values for $C$ vary from 0.35–0.70 for flows with integral length scales ranging from 8–2 mm. That is, the mass burn rate is enhanced in flows with smaller integral scales. Recall that the integral length scale characterises the energy containing scales of turbulent flows. From this, $C = 0.67$
suggests that integral length scales of \( \approx 2.5 \) mm are associated with the turbulent flow field.

To verify that the integral length scales are \( O(2.5) \) mm in the optical engine, note that the lowest frequency with significant energy in the turbulent flow field is 36.6 (engine cycle)\(^{-1}\). This is seen in the DWT PSD results presented in section 4.2.2. This lower frequency limit is due to the low pass filtering scheme used to calculate the mean velocity. Since the integral scales characterise the energy containing turbulence scales, \( f = 36.6 \) (engine cycle)\(^{-1}\) is expected to be associated with the integral scale. The size of the scale associated with \( f = 36.6 \) (engine cycle)\(^{-1}\) is determined through a Taylor's type hypothesis. \( \Lambda = \tau S_p = 2L/f \), where \( f \) has units of (engine cycle)\(^{-1}\), \( \tau = 720/f \) is the CAD scale. \( S_p \) is a characteristic velocity, which in this case is chosen as the mean piston speed, and \( L = 0.084 \) m is the stroke of the engine. The frequency of \( f = 36.6 \) (engine cycle)\(^{-1}\) is associated with scales \( O(3) \) mm at all operating points and, as expected, agrees with the integral scale. The empirical turbulent flame model results are, therefore, consistent with the results of Ting et al. (1995).

Note that, in fact, the low pass filtering scale \( f = 28.8 \) (engine cycle)\(^{-1}\) was chosen here so that the largest turbulence length scale of \( O(3) \) mm was \( \leq O(\text{flame kernel size}) \) through the flame kernel development period. In this way, the length scales associated with the turbulent eddies are, for the most part, smaller than the flame kernel, and wrinkle the flame front. The length scales associated with the mean flow processes are larger than the flame kernel during the EFKD period and advect the flame kernel.

The unaccounted variation in \( S_y \) from Equation 4.2 is attributed to cyclic variations in \( u' \) and the charge composition in the spark gap at the time of ignition. Simultaneous velocity–FOSP–spark spectroscopy–pressure measurements could be used to determine if the cyclic variations in \( u' \) and charge composition in the spark gap would account for some of the remaining dispersion: spark spectroscopy is a technique which measures the charge composition in the spark gap.
Direction of mean flame kernel convection

The interaction between the flame convection velocity magnitude and direction is examined through the joint PDF's. The joint magnitude–direction PDF's, plotted in Figure 4.10, show a transition in the shape of the joint PDF's as the load is increased at constant RPM: the load in the sub-figures of Figure 4.10 increases from left to right and the RPM increases from top to bottom. For example, the joint PDF at (70 kPa, 1050 RPM), see Figure 4.10(d), shows that the direction of the flame front convection remains within a band 0–70° while the convection velocity is distributed over a non-localised band of 0–12.5 m s⁻¹. At (90 kPa, 1050 RPM), see Figure 4.10(f), the \( \overline{V_r} \) is localised at 9–15.5 m s⁻¹ within the 0–70° band. The joint PDF at (80 kPa, 1050 RPM), see Figure 4.10(e), shows a transition between these two regimes. Similar transitions are seen in the 925 and 1200 RPM joint PDF's.

Note that as the mass flow rate increases, the joint PDF's move from a non-localised joint PDF with respect to velocity magnitude at low mass flow rates, for example, 4.9 g s⁻¹ at (70 kPa, 1050 RPM), to a localised joint PDF at high mass flow rates, for example, 6.6 g s⁻¹ at (90 kPa, 1050 RPM). For operating conditions with similar mass flow rates, such as (90 kPa, and 925 RPM) and (80 kPa, and 1050 RPM), the joint PDF's are similar in shape. The cyclic variations in the mean convection velocity, therefore, depend on the mass flow rate. At higher intake mass flow rates, the cyclic variations in the mean convection velocity is reduced. Since previous results from this work showed that the FOSP measures a mass-weighted velocity, the reduction in cyclic variations at higher mean convection velocities is interpreted as follows: when the swirling flow has higher momentum, that is, higher \( \dot{m} \propto \rho|V| \), the swirl flow precesses less.

4.3.2 Factors which influence the IMEP variations

The influence of the EFKD period on the operating characteristics of the engine is examined next. In particular, the factors which influence the cyclic variations in IMEP are examined. The IMEP cyclic variations are visualised through residual plots
(a) 70 kPa 925 RPM: $m_r = 4.6 \text{ g s}^{-1}$.

(b) 80 kPa 925 RPM: $m_r = 5.4 \text{ g s}^{-1}$.

(c) 90 kPa 925 RPM: $m_r = 6.2 \text{ g s}^{-1}$.

(d) 70 kPa 1050 RPM: $m_r = 4.9 \text{ g s}^{-1}$.

(e) 80 kPa 1050 RPM: $m_r = 6.0 \text{ g s}^{-1}$.

(f) 90 kPa 1050 RPM: $m_r = 6.6 \text{ g s}^{-1}$.

(g) 70 kPa 1200 RPM: $m_r = 5.6 \text{ g s}^{-1}$.

(h) 80 kPa 1200 RPM: $m_r = 6.8 \text{ g s}^{-1}$.

(i) 90 kPa 1200 RPM: $m_r = 7.7 \text{ g s}^{-1}$.

Figure 4.10: Joint flame kernel convection velocity magnitude–direction PDF.
where the residual of cycle \( k \), \( \text{IMEP}(k) - \langle \text{IMEP} \rangle \), is plotted versus cycle number. The IMEP residual plots are shown in Figure 4.11. The residual plots for operating points (90 kPa, 1050 RPM) and (70 kPa, 1050 RPM) show small drifts on the order of \( \pm 2\% \) and \(-1\% \) respectively. The reason for these monotonic drifts is that the mass flow rate into the engine varied over time, possibly due to (i) the chiller used to dry the intake air. and (ii) the Bowditch piston expansion were not fully stabilised. The residual plots for (90 kPa, 925 RPM) and (90 kPa, 1200 RPM) show large drifts with corrections: the corrections to engine operation were made during the experiment if the MAP or air-fuel excess ratio drifted from the set-point. The residual plot for (80 kPa, 1050 RPM) shows a non-linear excursion of about 5\% : perhaps due to fuel-injector instability. The large drifts are noticeable as up to 15 minutes were required to acquire the data at each operating point. This is in contrast to being able to acquire 100 engine cycles of stand alone pressure data in under one minute. One improvement to reduce the engine drift is to use a continuous air flow through the chiller ensuring that the outlet temperature remains constant and addition of a throttle valve dynamometer control system to maintain a constant MAP.

Normalisation of the IMEP data by \( x_{lb}^{-1} \), see Figure 4.12. removes the drifts in the IMEP residual plots. The parameter \( x_{lb} \) is the final mass burn fraction corrected for late burning. That is, \( x_{lb} \) is associated with the mass burn fraction at the end of the rapid burn phase when the flame front reaches the combustion chamber walls. The parameter \( x_{lb} \) is calculated as the mass burn fraction at the time at which the mass burn rate initially drops to near zero. The values for \( A \), listed in Table 4.4. appear to be smaller at 925 RPM. but about the same at 1050 and 1200 RPM. The strong correlation between IMEP and \( x_{lb} \) is in agreement with the results of others. see. for example, the work of Ishii et al. (1997).

It is possible to isolate the factors which influence the remaining variations in IMEP. First, it is necessary to look at the influence of \( S_g \) on the mass burn rate in the early stages of combustion and the combustion phasing through \( \theta_{L5\%} \) and \( \theta_{L50\%} \). respectively. The parameters \( \theta_{L5\%} \) and \( \theta_{L50\%} \) are the location of the 5\% and 50\% burn points. respectively. The correlations of \( S_g \) with \( \theta_{L5\%} \) and \( \theta_{L50\%} \) were
Figure 4.11: IMEP residual versus cycle number.
Figure 4.12: $\text{IMEP}/x_{bl}^4$ residual versus cycle number.
Table 4.4: Correlation between IMEP and $x_{lb}$. Parameter $A$ is least squares estimate from $IMEP = x_{lb}^A$.

<table>
<thead>
<tr>
<th>MAP</th>
<th>RPM</th>
<th>$X$</th>
<th>$R_{xlb,imep}$</th>
<th>$A$</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>925</td>
<td>445</td>
<td>0.86</td>
<td>0.69 ± 0.04</td>
</tr>
<tr>
<td>70</td>
<td>1050</td>
<td>449</td>
<td>0.96</td>
<td>0.91 ± 0.03</td>
</tr>
<tr>
<td>70</td>
<td>1200</td>
<td>447</td>
<td>0.89</td>
<td>0.83 ± 0.05</td>
</tr>
<tr>
<td>80</td>
<td>925</td>
<td>430</td>
<td>0.86</td>
<td>0.55 ± 0.05</td>
</tr>
<tr>
<td>80</td>
<td>1050</td>
<td>348</td>
<td>0.97</td>
<td>0.93 ± 0.03</td>
</tr>
<tr>
<td>80</td>
<td>1200</td>
<td>404</td>
<td>0.92</td>
<td>0.88 ± 0.05</td>
</tr>
<tr>
<td>90</td>
<td>925</td>
<td>416</td>
<td>0.55</td>
<td>0.63 ± 0.05</td>
</tr>
<tr>
<td>90</td>
<td>1050</td>
<td>408</td>
<td>0.96</td>
<td>0.92 ± 0.03</td>
</tr>
<tr>
<td>90</td>
<td>1200</td>
<td>343</td>
<td>0.96</td>
<td>0.96 ± 0.03</td>
</tr>
</tbody>
</table>

marginal, as shown in Table 4.5. If the influence of $S_{gn}$ on the combustion phasing is examined, where $S_{gn}$ is $S_g$ with the $|\vec{V}_r|$ correlation removed, we find significant negative correlations between $S_{gn}$ and the combustion phasing parameters (Table 4.5). Equation 4.2 is used to define $S_{gn} = S_g - S_L S_p (0.51 |\vec{V}_r|)$. Note that $S_{gn}$ is not necessarily independent of $|\vec{V}_r|$, because $S_{gn}$ is possibly still dependent on $|\vec{V}_r|$ through, for example, the influence of increasing $|\vec{V}_r|$ on the unstretched laminar flame speed due to reduced heat losses to the spark plug electrodes.

The parameter $S_{gn}$ was found to be the best EFKD parameter to account for variations in the combustion phasing. Since the variations in $\theta_{L,50\%}$ account for between 2-50% of the variations in IMEP, $S_{gn}$ also has a significant influence on the IMEP variations through $\theta_{L,50\%}$. An implication of this result is that the effect of increasing $|\vec{V}_r|$ on the magnitude of $S_g$ is over-predicted by the cubic-spline model. The over-prediction of $S_g$ in the high swirl engine is still a problem and is due to displacement of the spark kernel arc from the spark gap over the fibre-optic probe.

The influence of $|\vec{V}_r|$ on $\theta_{L,5\%}$ and $\theta_{L,50\%}$ was examined: marginally significant correlations were found. This result is not surprising given that (i) the main influence of $|\vec{V}_r|$ would be through its influence on $S_g$ during the early stages of the combustion process; and (ii) marginally significant correlations between $S_g$ and $\theta_{L,5\%}$ and $\theta_{L,50\%}$ were found. To investigate the influence of $|\vec{V}_r|$ further, we consider conditional
Table 4.5: Correlation of $S_{yn}$ with $\theta_{L5\%}$ and $\theta_{L30\%}$ and $\theta_{L50\%}$ with IMEP. The error is reported as the 95% confidence interval for the correlations.

<table>
<thead>
<tr>
<th>MAP</th>
<th>RPM</th>
<th>$R_{Sg.L5}$</th>
<th>$R_{Sg.L50}$</th>
<th>$R_{Sgn.L5}$</th>
<th>$R_{Sgn.L50}$</th>
<th>$R_{imep.L50}$</th>
<th>Error limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>925</td>
<td>-0.05</td>
<td>0.03</td>
<td>-0.20</td>
<td>-0.16</td>
<td>-0.16</td>
<td>±0.10</td>
</tr>
<tr>
<td>70</td>
<td>1050</td>
<td>-0.19</td>
<td>-0.12</td>
<td>-0.47</td>
<td>-0.51</td>
<td>-0.54</td>
<td>±0.09</td>
</tr>
<tr>
<td>70</td>
<td>1200</td>
<td>-0.08</td>
<td>0.00</td>
<td>-0.32</td>
<td>-0.28</td>
<td>-0.61</td>
<td>±0.10</td>
</tr>
<tr>
<td>80</td>
<td>925</td>
<td>-0.10</td>
<td>-0.02</td>
<td>-0.30</td>
<td>-0.28</td>
<td>-0.47</td>
<td>±0.10</td>
</tr>
<tr>
<td>80</td>
<td>1050</td>
<td>-0.12</td>
<td>-0.08</td>
<td>-0.34</td>
<td>-0.37</td>
<td>-0.52</td>
<td>±0.11</td>
</tr>
<tr>
<td>80</td>
<td>1200</td>
<td>-0.22</td>
<td>-0.12</td>
<td>-0.30</td>
<td>-0.33</td>
<td>-0.74</td>
<td>±0.10</td>
</tr>
<tr>
<td>90</td>
<td>925</td>
<td>-0.15</td>
<td>-0.05</td>
<td>-0.22</td>
<td>-0.19</td>
<td>0.39</td>
<td>±0.10</td>
</tr>
<tr>
<td>90</td>
<td>1050</td>
<td>-0.14</td>
<td>-0.11</td>
<td>-0.33</td>
<td>-0.31</td>
<td>0.61</td>
<td>±0.10</td>
</tr>
<tr>
<td>90</td>
<td>1200</td>
<td>-0.05</td>
<td>0.04</td>
<td>-0.18</td>
<td>0.19</td>
<td>0.57</td>
<td>±0.11</td>
</tr>
</tbody>
</table>

correlations, where the data are conditionally sampled based on low, mid and high $|\vec{V}_c|$ values. The resulting conditional correlations of $|\vec{V}_c|$ with $\theta_{L5\%}$ are given in Table 4.6 and indicate that:

1. For low $|\vec{V}_c| < 5$ m s$^{-1}$, there are marginal negative correlations at low and mid loads, but no significant correlations at high loads. Higher flame convection velocities in this range result in a marginally significant increase in the early mass burn rate.

2. For mid $|\vec{V}_c|$ 5-10 m s$^{-1}$, there are marginal positive correlations at (70 kPa, 925 RPM) and (80 kPa, 1050 RPM). The effect of too much aerodynamic strain is noted, where the early mass burn rates now slow down with increasing $|\vec{V}_c|$.

3. For high $|\vec{V}_c| > 10$ m s$^{-1}$, there are significant positive correlations at all but two of the operating points. In this $|\vec{V}_c|$ range, an increase in the flame convection velocity results in a slower early average mass burn rate.

The conditional correlations between the variations in the high range $|\vec{V}_c|$ with $\theta_{L50\%}$ and IMEP are also significant, see Table 4.7. The location of the 50% burn point occurs later in the cycle with increasing $|\vec{V}_c|$ given that $|\vec{V}_c| > 10$ m s$^{-1}$, indicating that the overall mass burn rate is significantly reduced. A decrease in IMEP is also noted with increasing $|\vec{V}_c|$ once $|\vec{V}_c|$ increases above 10 m s$^{-1}$. The fact that there is a limit to
Table 4.6: Conditional correlations of $|\vec{V}_e|$ with $\theta_{L57}$.

<table>
<thead>
<tr>
<th>Operating points</th>
<th>Conditional correlations</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAP RPM $</td>
<td>\vec{V}_e</td>
</tr>
<tr>
<td>70 925</td>
<td>$-0.12 \pm 0.13$</td>
</tr>
<tr>
<td>70 1050</td>
<td>$-0.12 \pm 0.14$</td>
</tr>
<tr>
<td>70 1200</td>
<td>$-0.12 \pm 0.15$</td>
</tr>
<tr>
<td>80 925</td>
<td>$-0.13 \pm 0.14$</td>
</tr>
<tr>
<td>80 1050</td>
<td>$-0.18 \pm 0.18$</td>
</tr>
<tr>
<td>80 1200</td>
<td>$-0.03 \pm 0.18$</td>
</tr>
<tr>
<td>90 925</td>
<td>$+0.04 \pm 0.17$</td>
</tr>
<tr>
<td>90 1050</td>
<td>$+0.12 \pm 0.18$</td>
</tr>
<tr>
<td>90 1200</td>
<td>$-0.12 \pm 0.25$</td>
</tr>
</tbody>
</table>

how much the $|\vec{V}_e|$ can be increased before a detrimental effect on the mass burn rate is observed is an important consideration when modelling the influence of the flame kernel convection velocity on the combustion process. For example, Anbarasu and Abata (1996) conclude that increasing $|\vec{V}_e|$ from 5 to 12 m s$^{-1}$ significantly increases the mass burn rate where the increase is due to a reduction in the heat losses to the spark electrode and an increase in the flame front area during the EFKD period. They do not, however, account for the possibility that increasing the aerodynamic strain might cause the flame front to be extinguished locally resulting in a lower mass burn rate.

### 4.4 The influence of the flow field on combustion

Understanding the influence of the flow field on the natural gas combustion process through the simultaneous velocity-pressure measurements is the goal of this work. More specifically, the influence of the flow field on the early mass burn rate was studied through examination of the correlations between the characteristic flow and early mass burn parameters on an individual cycle basis. In this section, the results from examining the simultaneous pressure–fired velocity data are presented. The velocity field is examined over the 329–355 CAD range. This range is used because
Table 4.7: Conditional correlations of $|\bar{V}_e| > 10$ m s$^{-1}$ with $\theta_{L50\%}$ and IMEP. The error is reported as the 95% confidence interval.

| MAP | RPM | N  | $\langle \theta_{L50\%} \rangle$ | $\langle |\bar{V}_e| > 10 \rangle$ | $\langle \text{IMEP} \rangle$ | $\langle |\bar{V}_e| > 10 \rangle$ | Error limits |
|-----|-----|----|-------------------------------|---------------------------------|------------------|------------------|---------------|
| 70  | 925 | 110| 0.21                          | -0.2                          | -0.20            | ±0.19            |
| 70  | 1050| 119| 0.30                          | -0.15                         | ±0.18            |
| 70  | 1200| 115| 0.22                          | -0.15                         | ±0.19            |
| 80  | 925 | 134| 0.15                          | -0.16                         | ±0.17            |
| 80  | 1050| 127| 0.15                          | -0.09                         | ±0.18            |
| 80  | 1200| 144| 0.34                          | -0.30                         | ±0.17            |
| 90  | 925 | 149| 0.13                          | -0.18                         | ±0.16            |
| 90  | 1050| 164| 0.27                          | -0.14                         | ±0.16            |
| 90  | 1200| 140| 0.08                          | -0.20                         | ±0.17            |

The fired LDV data rates are so low after TDC that the spectra become biased; see Appendix I for details regarding spectral biasing due to the allowable data rate drop-out.

4.4.1 Correlations between the flow field and combustion parameters

The influence of the overall mean and turbulence flow energy on the early mass burn rates are first presented. Cyclic based averaging was used to separate the mean and turbulent flow, where the mean flow energy is defined for $f < 28.8$ CAD. The cut-off frequency was chosen from the results of section 4.2.1, which indicate that the turbulence energy at $f > 28.8$ (engine cycle)$^{-1}$ is approximately constant in the 330–360 CAD range. We look at the correlations between the total mean flow energy and the characteristic early mass burn rate parameter $\theta_{L50\%}$ on an individual cycle basis. The total mean flow energy for an engine cycle $k$ is calculated as $U^2(k) + V^2(k)$, where the mean flow energy for each component is calculated as $U^2_i(k) = \sum_{\theta=330}^{\theta=355} U^2_i(\theta, k); i = 1, 2$. The correlation between the total turbulence energy $u^2(k) + v^2(k)$ and $\theta_{L50\%}$ is presented, where the total turbulent flow energy for each component is calculated as $u^2_i(k) = \sum_{\theta=330}^{\theta=355} u^2_i(\theta, k); i = 1, 2$. 
Table 4.8: Correlations between the mean and turbulence energy and \( \theta_{L5\%} \). Scale based averaging was used to define the mean flow with a low pass filter of \( f = 28.8 \) (engine cycle)\(^{-1} \). 95\% confidence intervals are listed.

<table>
<thead>
<tr>
<th>MAP (kPa)</th>
<th>RPM (RPM)</th>
<th>( \mathcal{U}^2 + \mathcal{V}^2 \theta_{L5%} )</th>
<th>( u^2 + \bar{v}^2 \theta_{L5%} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>925</td>
<td>-0.08 ± 0.11</td>
<td>-0.25 ± 0.11</td>
</tr>
<tr>
<td>70</td>
<td>1050</td>
<td>+0.01 ± 0.15</td>
<td>-0.06 ± 0.15</td>
</tr>
<tr>
<td>70</td>
<td>1200</td>
<td>-0.04 ± 0.13</td>
<td>-0.16 ± 0.13</td>
</tr>
<tr>
<td>80</td>
<td>925</td>
<td>+0.10 ± 0.14</td>
<td>+0.01 ± 0.14</td>
</tr>
<tr>
<td>80</td>
<td>1050</td>
<td>-0.05 ± 0.12</td>
<td>-0.11 ± 0.12</td>
</tr>
<tr>
<td>80</td>
<td>1200</td>
<td>-0.15 ± 0.10</td>
<td>-0.23 ± 0.10</td>
</tr>
<tr>
<td>90</td>
<td>925</td>
<td>-0.13 ± 0.10</td>
<td>-0.24 ± 0.10</td>
</tr>
<tr>
<td>90</td>
<td>1050</td>
<td>-0.06 ± 0.11</td>
<td>-0.18 ± 0.11</td>
</tr>
<tr>
<td>90</td>
<td>1200</td>
<td>-0.08 ± 0.12</td>
<td>-0.21 ± 0.12</td>
</tr>
</tbody>
</table>

The results in Table 4.8 show that two operating points have a marginally significant correlation between the total mean flow energy and \( \theta_{L5\%} \), namely operating points (80 kPa, 1200 RPM) and (90 kPa, 925 RPM). Note that the significance of the correlation is determined using the null hypothesis that the correlation is equal to one. If the listed 95\% confidence interval encompasses zero, then the correlation is statistically insignificant. The insignificant mean flow energy correlation with \( \theta_{L5\%} \) agrees with the results of Ting et al. (1995) for a similar analysis. These results also agree, in general, with the FOSP results. The FOSP results in Table 4.6 show little or no influence of \( |\overline{V}_c| \) on the early mass burn rate. The FOSP results did, however, show that values of \( |\overline{V}_c| > 10 \text{ m s}^{-1} \) have a detrimental effect on the early mass burn rate. A similar analysis was carried out on the mean flow energy where low, mid and high total mean flow energy regions were correlated with \( \theta_{L5\%} \). No significant detrimental influence of high mean flow energy on the early mass burn rate was found except for operating point (80 kPa, 925 RPM), where a marginal positive conditional correlation of \( \mathcal{U}^2_1(k) + \mathcal{U}^2_2(k) > 7.5 \text{ m}^2\text{s}^{-2} \) with \( \theta_{L5\%} \) was found. The fact that there is not complete agreement between FOSP results and LDV results may be because:

1. the velocity correlations are made based on the total energy in a relatively wide CAD range, whereas the FOSP measurements are made only during the EFKD
period for each cycle.

2. the velocity measurements were made \( \approx 5 \) mm away from the spark gap, whereas the FOSP convection velocity is measured directly around the spark plug, and

3. the definition of mean flow energy might not be correct in each cycle. For example, if an increase in turbulence energy serves to increase the mass burn rate, and an increase in mean flow energy serves to decrease the mass burn rate, then if turbulence energy is incorrectly assigned to the mean for a particular individual cycle, there are competing effects which might offset one another to some degree. The FOSP flame convection velocity has no such ambiguity in definition.

The influence of the turbulence energy on \( \theta_{L5\%} \) are shown in Table 4.8 where six of the nine operating points have significant negative correlations. The negative correlations imply that an increase in turbulence energy serves to increase the early mass burn rate such that \( \theta_{L5\%} \) occurs earlier; again, the significant negative correlations of turbulence flow energy correlation with \( \theta_{L5\%} \) found here agrees with the results of Ting et al. (1995).

4.4.2 Correlations between the DWT energy and combustion parameters

Next, the regions in the DWT transform plane where changes in the turbulence energy influence the mass burn rate are isolated. That is, the influence of changes in the turbulence energy with respect to scale (frequency) and phase is examined. To isolate the regions in the DWT transform plane where changes in the turbulence energy affect the combustion process, the flow energy is first separated into the crank-degree frequency plane using the DWT. The energy of each \( (\Delta \theta_b, f) \) region in the DWT plane is then correlated with the characteristic mass burn rate parameters. The benefit of this technique is that if a particular \( (\Delta \theta_b, f) \) is found to influence the combustion process, then the size of the scale relative to the flame kernel size at that point in
the cycle is used to attribute the effect as being due to either a turbulence or bulk flow process. For example, in the early stages of combustion, when the flame kernel is small, then only small scale events are seen as turbulence by the flame kernel. The larger scales act only to stretch and convect the flame kernel. The size of the scale is determined through a Taylor's type hypothesis. \( \lambda = \tau S_p = 2L/f \), where \( f \) has units of (engine cycle)\(^{-1} \). \( \tau = 720/f \) is the CAD scale. \( S_p \) is a characteristic velocity, which in this case is chosen as the mean piston speed, and \( L = 0.084 \) m is the stroke of the engine.

We begin by looking at the correlation between the DWT coefficient energy and the characteristic early mass burn rate parameter \( \theta_{L5\%} \). The correlation is made between the energy at each DWT location \( u^2(k, \theta, f) + v^2(k, \theta, f) \) and \( \theta_{L5\%}(k) \) on an individual cycle basis. The result, as shown in Figure 4.13, is a 2-dimensional correlation coefficient map. Only the statistically significant, at a 95% confidence level, negative correlations are displayed in Figure 4.13. There are no statistically significant regions of positive correlation: an increase in the local energy level at \( f > 28.8 \) (engine cycle)\(^{-1} \), therefore, is not detrimental to the early mass burn rate for the range of local energy magnitudes which occur under the operating conditions considered.

Next consider the correlation coefficient map of the 925 RPM operating points, see Figures 4.13(a) - 4.13(c), which show that there are marginal significant negative correlations between the DWT energy map and the \( \theta_{L5\%} \). Specifically, at (90 kPa, 925 RPM), see Figure 4.13(c), with a spark timing of 344 CAD, a cascade of regions with significant negative correlation between \( u^2(k, \theta, f) + v^2(k, \theta, f) \) and \( \theta_{L5\%} \) are seen. The cascade begins at \( (\Delta \theta_b=330-340, f = 56.6) \) through to \( (\Delta \theta_b=342-347, f = 112.5) \) and \( (\Delta \theta_b=347-355, f = 225) \). The \( (\Delta \theta_b=330-340, f = 56.6) \) region is associated with the pre-ignition flow, whereas the \( (\Delta \theta_b=342-347, f = 112.5) \) region is associated with the ignition event, and \( (\Delta \theta_b=342-347, f = 112.5) \) with the EFKD

\(^{\dagger}\)Note that the Taylor's hypothesis is valid for stationary flows with turbulence intensities less than 10%. No similar conversion from frequency to scale exists for non-stationary, high turbulence intensity flows.
Figure 4.13: Negative correlations between the DWT energy and $\theta_{LT5}$. Only the correlations significant at a 95% confidence level are shown.
period. An increase in the energy in any of these regions appears to increase the early mass burn rate.

As an aid to interpret how these $$(\Delta \theta_b, f)$$ regions at (90 kPa, 925 RPM) with significant correlations influence the early stages of the combustion process. consider:

(i) that the regions of significant correlations look similar to the energy cascades discussed in section 4.2.2: and (ii) that the energy at $$(\Delta \theta_b=330-340, f = 56.6)$$ cannot directly influence the combustion process because the ignition event hasn’t occurred yet. Based on this, the significant correlations at $$(\Delta \theta_b=330-340, f = 56.6)$$ are the result of the energy cascade process and do not directly influence the early mass burn rate.

Consider an individual cycle with a high energy region at $$(\Delta \theta_b=330-340, f = 56.6)$$. The results of section 4.2.2 suggest that this energy cascades down through higher frequencies as time progresses. that is. through $$(\Delta \theta_b=342-347, f = 112.5)$$ and then to $$(\Delta \theta_b=347-355, f = 225)$$. If. on the other hand. an individual cycle has low energy at $$(\Delta \theta_b=330-340, f = 56.6)$$ then the $$(\Delta \theta_b=342-347, f = 112.5)$$ and $$(\Delta \theta_b=347-355, f = 225)$$ regions will also have relatively low energy because there is no source of new energy. This implies that if one region correlates well with $$\theta_{L5\%}$$ then all the regions will correlate well. Since the region $$(\Delta \theta_b=330-340, f = 56.6)$$ occurs before the ignition event the energy in this region influences the mass burn rate as a turbulence energy source for the regions $$(\Delta \theta_b=342-347, f = 112.5)$$ and $$(\Delta \theta_b=347-355, f = 225)$$ which directly influence the early mass burn rate.

Next, to determine whether the increase in mass burn rate is due to mean flow or turbulence effects. the frequencies which significantly influence $$\theta_{L5\%}$$ are converted to length scales through $${\mathcal{O}}(2L/f)$$ and are $${\mathcal{O}}(1.5 \text{ mm})$$ and $${\mathcal{O}}(0.75 \text{ mm})$$. Since these scales are smaller than the flame kernel size throughout most of the EFKD period. the influence of these scales is to increase mass transfer of charge into the flame reaction zone through turbulence enhanced mixing. These results agree with Ting et al. (1995). where the early mass burn rate was strongly correlated with eddies 0.5–3 mm in size: Ting et al. (1995) did not, however, isolate the regions of $$(\Delta \theta_b,f)$$ where the different scales were most important.
A similar cascade process is seen for the (70 kPa. 925 RPM) operating point. see Figure 4.13(a). with a spark timing of \( \theta_{ST} = 340^\circ \). In this case, the \((\Delta \theta_b=330-342. f = 56.6)\) region is the energy source, from which the energy cascades down through \((\Delta \theta_b=342-355. f = 56.6)\) and \((\Delta \theta_b=342-355. f = 112.5)\). The cascade process in this case is across a wider CAD region when compared with operating point (90 kPa. 925 RPM). Larger scales seem to have more influence at 70 kPa: whether the significant correlations at the large scales directly influence the combustion process, or act simply through the energy cascade is unclear because, in this case, the large scale correlations occur after the ignition event.

At 1200 RPM the 80 and 90 kPa operating points, see Figures 4.13(h) and 4.13(i), show significant negative correlations, where similar correlation cascade regions are seen. The results from operating points (70 kPa. 1050 RPM). (70 kPa. 1200 RPM) and (90 kPa. 1050 RPM). see Figures 4.13(d). 4.13(g) and 4.13(f), respectively, also show regions of significant negative correlation, but without clear correlation cascade regions.

Thus, it is possible to conclude that increased local fine scale energy in the early stages of combustion increases the mass burn rate through enhanced turbulence mixing. The source of this increased local fine scale energy is the break-up of larger scales as TDC is approached. The cyclic variations in when and where these larger scale structures break down are a source of the cyclic variations in the \( \theta_L^{\text{eq}} \), which in turn influences the engine operation, as shown in section 4.3.2.

The remaining operating points show no significant regions of negative correlations: this is expected based on the results presented in section 4.4.1. The problem with these results might be that the measurement location was not close enough to the spark plug to pick up the influence of the flow field on the early mass burn rate. The results of Johansson (1993) suggest that the work is in the correct region, but another series of tests should be made to test this hypothesis. Note that even the significant correlations are marginal. Another problem might be engine instability, where it is more difficult to measure the influence of variations in the flow energy on the early mass burn rate if other parameters such as the excess air ratio and intake
air temperature are varying over the course of the measurements.

The use of the DWT: (i) to examine the spectral dynamics of engine flows and (ii) to isolate the influence of cyclic variations in the scale and phase of turbulence energy variations, is a novel application and shows a great deal of promise. The results give more insight into how the break down of turbulent structures as TDC is approached influence the combustion process in SI engines. Repeating these types of measurements in a variety of engine configurations should give a better insight into how the energy cascade processes vary from one engine to the next. The results should also provide an improved insight into how the different in-cylinder flows influence the early flame kernel development of natural gas fuelled engines. In this way, the engine configurations best suited for the natural gas fuelled spark ignition engines can be determined.
Chapter 5

Conclusions

The objective of this work was to determine the influence of in-cylinder flows on the natural gas combustion process. Results from a single cylinder V6 optical engine were reported. A physical interpretation: (i) of the flow field evolution, and (ii) of the interactions between the flow field and combustion process was presented. The following series of conclusions are supported by this work:

1. Examination of the flow field showed that:

   (a) The turbulence energy, for $f \geq 28.8 \text{ (engine cycle)}^{-1}$, scales with mean piston speed. The $u$ and $v$ component velocities are anisotropic in the large scales, $f \leq 28.8 \text{ (engine cycle)}^{-1}$, and isotropic in the mid-range scales $f = 56.6-250 \text{ (engine cycle)}^{-1}$. The motored large scale energy decreases during the $\Delta \theta_b = 335-385 \text{ CAD window}$, whereas the energy in the mid-range scales stays constant.

   (b) There is a significant increase in turbulence energy after the arrival of the flame front. Conditional averaging with respect to flame arrival time and flame kernel convection velocity is, however, needed to reduce biases due to quasi-periodic variations in the mean flow structures.

   (c) The DWT ensemble average contour plots show the in-cylinder energy cascade. Two energy cascades are seen prior to TDC for both the $u$ and $v$
component velocities at all the operating points. An energy deficit region is also seen immediately after TDC. The results suggest that vortex structures break up prior to TDC.

(d) The DWT individual cycle contour plots show that the energy transfer occurs: (i) from large to small scale, and (ii) between the u and v component flows. The energy distribution among the crank degree-frequency plane varies significantly from one cycle to the next. Advection of energetic local events occurs regularly and might be associated with cyclic variations in the mean.

2. The pressure-FOSP results indicate that:

(a) The FOSP measures a mass weighted velocity. The $\langle | \tilde{v} | \rangle$ depends on $S_p$ and $\rho_n^{0.65}$.

(b) The degree of swirl precession decreases with increasing intake flow momentum. This result is supported by the joint magnitude–direction PDF’s of the flame kernel convection velocity.

(c) The EFKD mass burn rate, as quantified through $S_g$, is strongly dependent on the convection velocity magnitude. A linear, first order equation describes the relationship between $S_g$ and $| \tilde{v} |$ on an individual cycle basis over all the operating points.

(d) The empirical turbulent flame model $S_T/S_L = 1 + 0.67 \ | \tilde{u} | / S_L$ was developed.

(e) The increase in $S_g$ with $\tilde{v}_r$ is overpredicted by the cubic-spline model.

(f) The cyclic variations in IMEP are found to be influenced by the following factors:

i. The final mass burn fraction corrected for late burning.

ii. The combustion phasing as characterised by $\theta_{L,50\%}$.

iii. The early mass burn rate.
iv. The variation in $S_p$ with the correlation of $|\bar{V}_c|$ removed.

v. The variation in $|\bar{V}_c|$ at $|\bar{V}_c| > 10$ m s$^{-1}$ has a significant, detrimental influence on the EFKD period and combustion phasing.

3. The velocity-pressure data analysis supports the following conclusions:

   (a) The cyclic mean flow energy at $f \leq 28.8$ (engine cycle)$^{-1}$ in the 330–355 CAD range, measured $<5$ mm away from the spark plug, does not influence the 0–5% mass burn duration.

   (b) Individual cycles with larger cyclic based turbulence energy at $f \geq 28.8$ (engine cycle)$^{-1}$ in the 330–355 CAD range, measured $<5$ mm away from the spark plug, have faster early mass burn rates.

   (c) Increased local fine scale energy in the early stages of combustion increases the mass burn rate through enhanced turbulence mixing. The source of this increased local fine scale energy is the break-up of larger scales as TDC is approached. The cyclic variations in when and where these larger scale structures break down are a source of the cyclic variations in the $\theta_{L5\%}$, which in turn influences the engine operation.

5.1 Recommendations

1. This work lays the foundation for future measurements in engines with different flows to determine how different flow fields influence the combustion process.

2. The velocity measurements should be repeated closer to the spark gap to determine if stronger correlations between the small scale energy and early mass burn rates can be found.

3. A more systematic approach looking into spark anemometry through simultaneous measurements of the secondary voltage and the in-cylinder flow field via LDV.
4. The FOSP models should be re-evaluated to reduce the bias introduced in \( S_7 \) by high convection velocities.

5. The stability of the SCV6 engines should be improved through stabilising the intake air temperature and excess air ratio. Throttle position control through the dynamometer to maintain a constant MAP is also envisioned.
References


Ancimer, R. (1995). Development of a spark ignition engine in-cylinder pressure measurement and analysis system [Department of Mechanical and Industrial Engineering]. (Engine R&D Laboratory - University of Toronto technical report)

Ancimer, R. (1997). Development of a spark ignition engine in-cylinder velocity measurement and analysis system [Engine Research and Development Laboratory - Department of Mechanical and Industrial Engineering]. (Manuscript in preparation)


Appendix A

Characterising turbulent flows

Turbulent flows are characterised by the turbulent kinetic energy and length scales. The turbulent kinetic energy, TKE, is calculated in stationary flows by first decomposing the flow into a stationary mean component and a fluctuating turbulent velocity (Tennekes & Lumley, 1972). Using tensor notation the formulas are given by Equations A.1–A.4.

\[
\begin{align*}
U_i &= \langle \hat{u}_i(t) \rangle \\
u_i(t) &= \hat{u}_i(t) - U_i \\
u'_i &= \sqrt{\langle (u_i(t))^2 \rangle} \\
q &= \frac{1}{3} u'_i u'_i
\end{align*}
\]  

(A.1) (A.2) (A.3) (A.4)

A double index denotes a summation, and \( \langle \cdot \rangle \) denotes a time average in the case of stationary, ergodic flows. \( \hat{u}_i(t) \), \( U_i \) and \( q \) are the instantaneous flow velocity, the mean velocity and the TKE, respectively. In stationary flows \( U_i \), \( u'_i \) and \( q \) are independent of time.

A.1 Correlation scales

Large scale processes in stationary turbulent flows are characterised by second order correlations: the integral length scale \( l_i \) is the region over which velocity correla-
tions exist in the turbulent flow, and is a measure of the energy containing scales at which turbulence production occurs (Tennekes & Lumley, 1972; Holmes, Lumley, & Berkooz, 1996). The integral length scale is proportional to the largest velocity scale, or large eddies, in the flow. A time scale, \( \tau_m \), associated with the integral scales is the characteristic eddy turnover time. It is defined as \( \tau_m = l_1/u' \) and is a measure of the average lifetime of a large scale eddy (Holmes et al., 1996). The large scale processes are important because they show the most structure, contain the most energy and may make a significant contribution to the Reynolds stresses which produce the turbulence (Holmes et al., 1996).

### A.2 Dissipative scales

The turbulence energy cascade is visualised as the integral scales extracting turbulent kinetic energy from the mean flow which is then passed down to smaller and smaller scales until viscous forces dissipate the TKE at the smallest scales (Tennekes & Lumley, 1972; Holmes et al., 1996). In flows with large \( \text{Re}_T \), where \( \text{Re}_T = u' l_1 / \nu \) is a non-dimensional parameter characterising the relative strength of the inertial and viscous stresses. Kolmogorov determined the smallest scale within a turbulent flow

\[
l_k = \left( \frac{\nu^3}{\epsilon} \right)^{\frac{1}{4}}
\]  

(A.5)

where \( \nu \) is the kinematic viscosity of the fluid and \( \epsilon \) is the dissipation rate of TKE (Tennekes & Lumley, 1972). The rate of dissipation, \( \epsilon \), is determined by the large scale processes. The greater the TKE, the greater the rate of dissipation and the smaller the Kolmogorov scales. The turbulent dissipation rate can be estimated as

\[
\epsilon = \frac{(u')^3}{\nu}
\]  

(A.6)

Another dissipative scale is the Taylor micro scale, defined by Equation A.7 (Ten-
The Taylor microscale $l_m$ is an estimate of the scale where the rate of dissipation at the turbulence intensity $u'$ is approximately balanced with the rate of turbulence produced at the large scales (Hinze. 1987).

**Characterising SI engine flows** The Reynolds decomposition of stationary and ergodic flows into a mean and fluctuating turbulence velocity is a two variable problem: the averaging time, $T$, must be much greater than the characteristic time of the non-stationarity of the process, $\tau_m$. This condition cannot be met in one engine cycle because of the valve and piston motions, that is, spark ignition engine flows are non-stationary. Stationary flow averaging techniques cannot be applied to SI engine flows; engine flows require a different averaging procedure to evaluate the fluctuating turbulence velocity. The details of the analysis problem are discussed in chapter 3.

Spark ignition engine flows may also be characterised by the turbulence velocity $u'_1$ and integral scales $l_l$. The turbulence velocity and length scales, however, vary with crank angle degree throughout the engine cycle (Arcoumanis & Whitelaw. 1987; Hill & Zhang. 1994), and may also vary from one cycle to the next (Hall. 1989; Whitelaw & Xu. 1995; Ozdor et al.. 1996; Brown. Stone. & Beckwith. 1996; Stone et al.. 1996). Past work has concentrated on the characteristics of the turbulent flow field during the combustion period, see for example Hall (1989). Kent et al. (1989). Baritaud (1989). Hadded and Denbratt (1991). Johansson and Olsson (1995), because the turbulence intensity and scales during this period have a direct impact on the combustion process.

$$l_m = \frac{u'_1}{\sqrt{\langle u'_{1,1} \rangle}}$$
Appendix B

Combustion fundamentals

A brief review of fundamental combustion processes is presented in this appendix.

B.1 Laminar flame structure and speed

A flame is a self-sustaining chemical reaction within a region of space called a flame front (Glassman, 1987). A schematic of a premixed flame is shown in Figure B.1. The flame consists of a preheat zone where the temperature of the unburned charge is increased from an initial value of $T_O$ to the ignition temperature $T_I$. Heating is done mainly through conduction. Where the charge temperature is high enough an exothermic reaction takes place. The reactants in the unburned charge are converted to burned products. The extent of the reaction zone ranges from where the charge is at the ignition temperature to where there are insignificant reactions occurring. The flame thickness $\delta_L$ is defined to extend over the preheat and reaction zones. In most cases the reaction zone region is much smaller in extent than the preheat zone. The burn rate is limited by the mass transfer of unburned charge into the reaction, or thin flame, zone.

Premixed charge combustion rates are characterised by the laminar flame burning speed $S_L$. The laminar burning speed is the speed of the flame front into the unburned gas in a quiescent mixture and is a thermochemical property of the mixture (Glassman, 1987). The magnitude of $S_L$ depends on the chemical reaction rate.
unburned charge is heated by conduction to the ignition temperature. The unburned charge is then converted to burned products in the rapid reaction zone. Drawn from (Glassman. 1987)

\[ \dot{\omega}, \] and the mass diffusivity, \( \mathcal{D} \), of the limiting reactant. The structure of the flame is characterised by the flame thickness, \( \delta_L \), and chemical reaction time scale, \( \tau_c \). The relationships are shown in Equations B.1–B.2.

\[
S_L \propto (\mathcal{D} \dot{\omega})^{1/2} \tag{B.1}
\]

\[
\tau_c = \frac{\delta_L}{S_L} = \frac{1}{\dot{\omega}} \tag{B.2}
\]

where \( \delta_L \) is more appropriately the characteristic thickness of the preheat zone (Glassman. 1987) and can be estimated by Equation B.3 (Heywood. 1988).

\[
\delta_L = \frac{4.6 \bar{k}}{\bar{c}_p \rho_u S_L} \tag{B.3}
\]

where \( \bar{k} \) and \( \bar{c}_p \) are the mean thermal conductivity and specific heat at constant pressure in the preheat zone.

The most important factors influencing \( S_L \) are the unburned gas temperature
and pressure, excess air ratio $\lambda$, amount of diluents, and fuel type (Glassman, 1987). Equations B.4–B.5 summarise these effects for methane-air mixtures (Heywood, 1988) in SI engines.

\begin{align*}
S_L & = S_{L,0} \left( \frac{\rho_a}{\rho_{ao}} \right)_s^{0.18} = \left( \frac{\rho_o}{p} \right)^{0.18} \quad (B.4) \\
S_L(x_b) & = S_L(x_b) \left( 1 - 2.06x_b^{0.77} \right) \quad (B.5)
\end{align*}

where $x_b$ in Equation B.5 is the residual gas fraction. Equation B.4 is valid for $\lambda = 0.83$–1.25 and $p = 1$–8 atmospheres: the values of $S_{L,0}$ are listed in Table B.1. Similar relations are available for other fuels (Heywood, 1988).

Table B.1: Values of $S_{L,0}$ for use in Equation B.4. Taken from Heywood (1988)

<table>
<thead>
<tr>
<th>$\lambda$</th>
<th>1.25</th>
<th>1.1</th>
<th>1.0</th>
<th>.9</th>
<th>.83</th>
</tr>
</thead>
<tbody>
<tr>
<td>$S_{L,0}$ (cm s$^{-1}$)</td>
<td>30</td>
<td>35</td>
<td>37</td>
<td>40</td>
<td>32</td>
</tr>
</tbody>
</table>

The laminar burning speed in SI engines may vary substantially because the temperature, pressure, and mixture composition of the in-cylinder charge vary (Heywood, 1988; Bradley et al., 1988). For example,

1. The laminar flame speed decreases with crank degree during the compression stroke because of the rise in in-cylinder temperature and pressure.

2. The laminar flame speed decreases as the throttle is closed because the amount of residual gas $x_b$ increases.

3. Variations in $S_L$ may exist from one cylinder to the next, or within one cylinder, due to imperfect mixing. The variations in $\lambda$ result in variations in $S_L$, see Table B.1.

4. The laminar burning speed of gasoline is larger than that of natural gas at equivalent quiescent conditions (Bradley et al., 1996). This is attributed to slower reaction rate chemistry for natural gas relative to gasoline. Spark ignition engine configurations need to be optimised for a particular type of fuel.
Understanding how $S_L$ varies and the consequences of these variations on the combustion process is important in improving the performance of SI engines, especially when optimising SI engine performance with respect to the fuel type.

### B.2 Turbulent flame structure and speed

A laminar flame exists when the flow conditions do not alter the chemical mechanisms or energy release rate of the combustion process. A turbulent flame exists when the fluctuations in the velocity, temperature, density and species concentration enhance the combustion rate (Andrews, Bradley, & Lwakabamba, 1975; Kuo, 1986). For premixed flames the primary influence of the turbulent flow field is through enhanced mixing. That is, the phenomenon is one of fluid dynamics and not of chemical kinetics, besides the issues of ignition and quenching (Kuo, 1986). The effect of the turbulent flow field is an increased mass burn rate characterised by the turbulent flame speed $S_T$. The influence of the turbulent flow field on the combustion process depends on the TKE and turbulence length scales, and the laminar flame burning speed, $S_L$, and thickness $\delta_L$ (Andrews et al., 1975; Williams, 1985).

The Damköhler plot, see Figure B.2, systematically characterises the turbulent flame structure using the dimensionless Damköhler number $Da$ and turbulent Reynolds number. $Re_T = (u' l') / \nu$. $Da$ is the ratio of the characteristic fluid mixing $\tau_m$ and chemical reaction $\tau_c$ times. It is estimated using Equations B.6–B.8.

\[
\tau_m = \frac{l}{u'} \tag{B.6}
\]
\[
\tau_c = \frac{d_L}{S_n} \tag{B.7}
\]
\[
Da = \frac{\tau_m}{\tau_c} \tag{B.8}
\]

and gives a relative measure of the influence of turbulence mixing on the combustion process (Glassman, 1987). $S_n$ is the stretched laminar flame burning speed: $S_n = S_L$ when the flame stretch rate is small. The influence of the flame stretch rate on $S_L$ is discussed in subsection B.3. For weak turbulence, $Da$ is large and the laminar
diffusion processes limit the reaction rate: the turbulence mixing has little influence on the combustion process. For strong turbulence the $Da$ is small and a distributed reaction zone results where reactions occur throughout the phase due to the enhanced turbulent mixing. These two regimes are labelled on the Damköhler plot. Figure B.2.

![Diagram showing Damköhler number vs. turbulent Reynolds number.](image)

Figure B.2: Damköhler number vs. turbulent Reynolds number. $u'$ is the turbulence velocity; $S_L$ is the laminar flame burning speed; $l_I$ and $l_k$ are the integral and Kolmogorov length scales, respectively; and $\delta_L$ is the laminar flame thickness. Figure drawn from (Heywood, 1988).

The structure of the turbulent flames between these two extremes depends on the magnitude of the TKE and the turbulent length scales $l_I$ and $l_k$ (Andrews et al., 1975; Kuo, 1986; Glassman, 1987). Three additional dimensionless parameters are used to characterise the turbulent flame structure: the size of the correlation and dissipative turbulent scales relative to the laminar flame front thickness. That is, $l_I/\delta_L$ and $l_k/\delta_L$ respectively; and the relative strength of the turbulence $u'/S_n$. Lines of constant $l_I/\delta_L$, $l_k/\delta_L$ and $u'/S_n$ are plotted in Figure B.2. These lines were obtained by assuming that the flow is homogeneous and isotropic where the relationship between the integral.
Kolmogorov and Taylor length scales is given by (Tennekes & Lumley, 1972)

\[
\frac{l_k}{l_I} = Re_T^{-\frac{2}{3}} \quad \text{(B.9)}
\]
\[
\frac{l_m}{l_I} = Re_T^{-\frac{1}{3}} \quad \text{(B.10)}
\]

**Simply connected reaction sheets** Reaction sheets, or wrinkled laminar flames, are simply connected laminar flame fronts which are wrinkled by the turbulence (Andrews et al., 1975). They exist when the turbulence scales are large relative to the flame front thickness and the relative turbulence intensity is low: sufficient conditions are \( \delta_L << l_I, \delta_L << l_k, \) and \( u' < S_n \). Strong dissipative vortex structures do not exist under these conditions. The reaction sheet regime is labelled on the Damköhler plot, Figure B.2. Since the \( Da \) number is large, the reaction rate chemistry is much faster than the turbulence mixing process and the burn rate is limited by the unburned gas mass transfer rate into the reaction zone (Andrews et al., 1975; Glassman, 1987). The turbulent flame speed \( S_T \) of reaction sheets is larger than the laminar flame burning speed \( S_n \) (Andrews et al., 1975). The burn rate increases because of the larger wrinkled flame front surface area. The larger surface area increases the rate limiting unburned charge mass transfer into the reaction zone. Spatial variations in the velocity gradients produce the flame front wrinkling (Checkel & Thomas, 1994): the flame area increases as the turbulent velocity fluctuations increase (Yoshida & Tsuji, 1982). The length scale of the wrinkles is insensitive to the turbulence length scales and unburned gas velocity (Yoshida & Tsuji, 1982). It is tacitly assumed that the local reaction rate is unaffected by the turbulent mixing processes. This is a valid assumption in the reaction sheet regime because the flame stretch induced by the large scale turbulence is small (Andrews et al., 1975): the factors which influence \( S_L \) are discussed in subsection B.3. If the local reaction rate is unaffected by the turbulence, then the charge burns locally at \( S_n \). The magnitude of the turbulent flame speed \( S_T \) is, therefore, dependent on the relative turbulence intensity and \( S_n \). One
proposed model is given by (Andrews et al., 1975).

\[ \frac{S_T}{S_n} \approx 1 + \frac{u'}{S_n} \]  

(B.11)

**Multiply-connected reaction sheets** The simply connected reaction sheet regime is characterised by low values of \( Re_{lm} = u'l_m/\nu \) where the dissipative regions are weak and the flame propagation is independent of fine scale mixing (Andrews et al., 1975). As \( Re_{lm} \) increases the dissipative regions become stronger and the small scale mixing processes become significant with respect to the combustion process. The approximate boundary for this condition is where the relative turbulence intensity\(^*\) \( u'/S_n = 1 \). Figure B.2 (Andrews et al., 1975; Williams, 1985; Ashurst, Checkel, & Ting, 1994). The wrinkled laminar flame front may become highly convoluted and multiply-connected: flamelets begin to propagate into the unburned charge ahead of the flame front. The mechanisms to account for multiply-connected reaction sheets are complex and are not fully understood (Andrews et al., 1975; Williams, 1985; Ashurst et al., 1994; Checkel & Thomas, 1994).

In the multiply-connected reaction sheet regime preferential combustion is thought to occur within the isotropic dissipative scale vortex structures (Andrews et al., 1975; Ashurst et al., 1994) because of the enhanced mixing within these structures. The local reaction may propagate in the dissipative regions at a speed much greater than \( S_n \), that is, flamelets propagate into the unburned charge. The flamelet propagation is dependent on the enhanced transport mechanisms within the dissipative regions, that is, \( S_T/S_n = f(\epsilon, \nu) \). The enhanced transport mechanisms are often characterised by the RMS of the turbulence strain rate, \( \sigma \), where \( \sigma \) is estimated from the turbulence

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\(^*\)In wrinkled laminar flames the large scale eddies burn at \( S_n \). When the lifetime of a large scale eddy, \( l_l/\nu' \), is approximately the same as the laminar burning time, \( l_l/S_n \), the eddy may decay before combustion is complete. Combustion may continue as a newly formed eddy or may cease (Andrews et al., 1975): under these conditions the turbulent flame front propagation via the small scale turbulence becomes significant.
dissipation rate, $\epsilon$ (Checkel & Thomas, 1994).

$$ G = \sqrt{\frac{\epsilon}{\nu}} $$  \hspace{1cm} (B.12)

$$ \frac{S_T}{S_n} = f(G) $$  \hspace{1cm} (B.13)

The flamelets move forward along the vortex structures at speeds $O(\bar{u}^\prime)$ disrupting the continuous flame front. Comparatively large volumes of unburned gas are left behind, on the $O(l_i)$ (Andrews et al., 1975; Ashurst et al., 1994). The pockets of unburned gas behind the flame front burn at $S_n$.

The increase in $S_T$ with respect to $u' \approx 1$ because the overall burn rate becomes limited by the advancement of the flame front between the vortical regions (Checkel & Thomas, 1994). Quenching of the flame front becomes an issue in this regime if the local strain rate is too high: quenching becomes likely when $l_k/\delta_L < 1$. The structure of turbulent flames when $l_l > \delta_L$ and $l_k < \delta_L$ is unclear. Quenching issues are discussed in subsection B.3.

**Influence of combustion on the turbulence** The energy released by combustion affects the turbulent flow by increasing the kinematic viscosity of the fluid. This decreases the turbulent Reynolds number via

$$ \frac{Re_b}{Re_u} = \frac{\nu_u}{\nu_b} \approx \left(\frac{T_u}{T_b}\right)^2 $$

An increase in temperature by a factor of 3, which is modest by combustion standards, leads to an order of magnitude reduction in $Re$. Combustion can, therefore, dampen the turbulent flow field behind the flame front significantly.

The propagating flame front may induce other effects such that the turbulence ahead of the flame front is increased. The rapid distortion theory of Chew and Butter (1992) predicts, based on the geometric strain and compression effects induced by the advancing flame front on the unburned gas vortices, an increased turbulence velocity in the unburned gas ahead of the flame front in a confined vessel. Increases in $u'$ of
up to 100%, depending on the conditions, are predicted.

**Summary** The different factors influencing turbulent flame speed are summarised by of the model proposed by Povinelli and Fuhs (1962). Equation B.14.

\[
\frac{S_T}{S_n} = \left(1 + \frac{u'}{S_n}\right) \int_0^{n_o} F(n)dn + \left(1 + \frac{\varepsilon}{\nu}\right) \int_{n_o}^{\infty} F(n)dn
\]  \hspace{1cm} (B.14)

The parameter \(n_o\) is a frequency separating the large and small scale turbulent fluctuations. \(\varepsilon = f(\epsilon)\) is the turbulent mass diffusivity and \(F(n)\) is the normalised spectral function of the TKE. The parameter \(n_o\) is dependent on the size of the turbulent eddies relative to the flame front thickness \(\delta_L\). The large scales wrinkle the flame front and the small scales influence the local transport properties. The distribution of the TKE among the wavenumbers influences the turbulent flame structure: i) a simply connected reaction sheet is expected if the TKE above \(n_o\) is small; ii) as the TKE above \(n_o\) increases, a multiply-connected reaction sheet is expected; and iii) if the TKE in the small scales is further increased a distributed reaction zone is expected as the turbulent mixing processes induce a distributed reaction zone.

### B.3 Flame stretching and the combustion process

As discussed in the previous section, the turbulent flow field increases the combustion rate relative to quiescent flows through enhanced mixing. Flow field induced flame stretching may also influence the combustion process: under certain circumstances flame stretching can lead to flame quenching. Flame stretching is an important issue with respect to stable SI engine operation, especially at light load conditions. An overview of how flame stretching influences the combustion process is presented next: how this influences the SI engine combustion is discussed in subsection B.4.

The flame stretch rate, \(\alpha\), is defined as the normalised time rate of change of an infinitesimal flame front surface area.

\[
\alpha = \left(\frac{1}{A} \frac{dA}{dt}\right)
\]  \hspace{1cm} (B.15)
The influence of the flame stretch rate on the $S_n$ is phenomologically quantified by an appropriate Markstein length, $L$, where the flame speed deficit is given by (Bradley et al., 1996).

\[ S_L - S_n = L \alpha \]  \hspace{1cm} (B.16)

$S_L$ and $S_n$ are the unstretched and stretched laminar burning velocities, respectively. Separate effects may be quantified by using a linear combination of each factor (Bradley et al., 1996), as shown by

\[ S_L - S_n = L_s \alpha_s + L_c \alpha_c \]  \hspace{1cm} (B.17)

where the effects of the aerodynamic strain rate $\alpha_s$ and flame front curvature $\alpha_c$ are assumed here to be the only significant factors influencing the the laminar burning speed deficit. The aerodynamic strain rate $\alpha_s$ accounts for both the global strain rate due to mean velocity gradients, and the turbulent strain rate.

A similar relationship can be shown to hold between the $S_L$ and $S_{nr}$.

\[ S_L - S_{nr} = L_{nr} \alpha_s \]  \hspace{1cm} (B.18)

where $S_{nr}$ is the speed associated with the rate of appearance of completely burned gas behind the flame front (Bradley et al., 1996). That is, Equation B.18 gives the influence of the flame stretch rate on the reaction zone whereas Equation B.17 gives the influence of the flame stretch on the rate of unburned gas entrained into the flame front. Under most conditions, the effect of flame front curvature on the reaction zone is insignificant relative to the effect of aerodynamic strain.

A non-zero Markstein length $L_i$ implies that: i) the flame speed may increase or decrease with the stretch rate, and ii) there is a flame-quenching stretch rate

\[^{1}\text{This is a first order approximation of the stretched flame front surface area. The error in this expression is } O(\varepsilon^2 u_i) \text{ where } \varepsilon=\delta_L \Lambda \text{ and } \Lambda \text{ is the wavelength associated with the perturbation (Bradley, Gaskell, & Gu, 1996).} \]
at which the burning speed becomes zero. These effects may manifest themselves through three modes, each associated with a different Markstein length: i) divergent energy losses due to flame front curvature $L_c$. ii) divergent energy losses due to flame front strain $L_s$. or iii) reaction zone stretching due to the small scale turbulence $L_{sr}$. The Markstein lengths are dependent on the fuel-air mixture composition and thermochemical properties and increase linearly with Lewis number $Le = \alpha' / D$ where $\alpha'$ is the thermal diffusivity of the mixture and $D$ is the mass diffusivity of the limiting reactant (Bradley et al., 1996; Daneshyar et al., 1983). For example, if a fuel-air mixture has $L_s > 0$ and there is insignificant flame front curvature and a positive strain rate, the flame speed decreases. In this case, there is divergent energy lost due to aerodynamic flame stretch which thins out the flame front and reduces the flame temperature with an associated speed deficit. This is characteristic of lean fuel-air mixtures with $Le > 1$ (Williams, 1985).

Flame quenching occurs when the laminar burning speed is reduced to zero. The conditions at which this may occur are given by:

$$\alpha_{iq} = \frac{S_L}{L_i} \quad (B.19)$$

where $i = c, s$ or $sr$, and it is assumed that the flame quenching stretching mode predominates the other modes of stretch at quenching. The susceptibility of a mixture to quenching increases with decreasing $S_L$ and increasing $L$. Quenching may occur:

1. Via $L_c$ just after ignition when the flame kernel radius is small. Under these conditions, the flame front curvature is large and curvature effects dominate the flame stretch rate. Flame front curvature may, therefore, lead to quenching of the flame kernel. To prevent this a large spark energy is needed to increase the burned gas temperature such that the resulting $S_L$ is large enough to overcome any curvature effects. Larger spark energy is needed for leaner flames because $Le$ is larger in lean mixtures, and $L_c$ increases linearly with Lewis number.

2. Via $L_s$ if there is excessive divergent energy lost due to aerodynamic strain $\alpha_s$ of the flame. The influence of $\alpha_s$ on the rate of unburned gas entrainment into the
flame front is characterised by $L_s$. For methane-air and gasoline-air mixtures $L_s$ is negative. This implies that a negative strain rate is required to extinguish the flame front: flame fronts are inherently unstable in negative strain rates. This effect has not been observed experimentally (Bradley et al., 1996).

3. Via $L_{sr}$ if there is excessive reaction zone stretching due to a small $Da$. Flame front quenching has been observed for positive strain rates (Bradley et al., 1996). This type of quenching is associated with reaction zone quenching where the entrained unburned gas fails to completely react. It is $S_{nr}$ which is quenched: once $Da$ is too high, the small scales eddies on $O(l_m)$ dissipate before they are completely burned, locally extinguishing the reaction zone.

If the Markstein lengths are unknown, the Karlovitz stretch factor $K_a$ may be used to characterise the flame front stretching. The Karlovitz stretch factor is the ratio of the chemical reaction time to that of the micro-scale mean eddy lifetime (Bradley et al., 1988; Klingmann & Johansson, 1998).

$$K_a = \frac{\tau_c}{\tau_m} = \left( \frac{\delta_L}{S_n} \right) \left( \frac{u'}{l_m} \right)$$

$$K_a = 0.157 \left( \frac{u'}{S_n} \right)^2 Re_T^{-\frac{1}{2}}$$

If the flame stretch rate due to the turbulence strain rate is estimated as $u'/l_m$, then $K_a$ is interpreted as the turbulence strain rate induced flame stretch normalised by $\delta_L/S_n$. In terms of $K_a$, quenching may occur when $K_a > 1$ (Bradley et al., 1988; Williams, 1985). The same interpretation holds as discussed for reaction zone quenching with respect to $L_{sr}$. The boundary can be shown to be where $l_k = \delta_L$. The quenching region on the Damköhler plot, see Figure B.2, is where $l_k < \delta_L$: flame front instability is expected in this region.

The Karlovitz stretch parameter can be estimated by Equation B.21. For low values of $K_a$, for example $< 0.10$, the flame front is a wrinkled laminar flame (Klingmann & Johansson, 1998). For mid range values of $K_a$, for example $\approx 0.20$, a convoluted, multiply-connected flame front begins to manifest itself due to the increased tur-
bulence strain rate (Klingmann & Johansson, 1998). The flame front tends to be convoluted and severely folded. At high values of $K_a$ there is complete quenching. This is associated with the turbulence strain rate extinguishing the reaction zone.

### B.4 Turbulent flames in SI engines

The region where gasoline powered SI engines operate is highlighted in Figure B.2. The engine operating range lies below the $u' = S_L$ line and for the most part above the $l_k = \delta_L$ line. The turbulent flame structure is multiply-connected reaction sheets (Keck, 1982; Cho & Santavicca, 1993; Whitelaw & Xu, 1995; Arcoumanis et al., 1998). Only at high speeds and low loads does the region lie below the $l_k = \delta_L$ line (Heywood, 1988). Whether the flame structure in this region is different has not yet been determined. Observations of engine flames to date lie above the $l_k = \delta_L$ line. Under the low load conditions, the laminar flame burning speed is significantly lower due to higher residual gas fraction: larger cyclic variations in the combustion process have been observed (Ting et al., 1995; Keck et al., 1987; Cho & Santavicca, 1993). Whether this is due to local extinction of the flame due to excessive in-cylinder aerodynamic strain remains to be determined. The aerodynamic strain rate in SI engines consists of stretching due to mean motions where the flame kernel is anchored to the spark plug, and the turbulent strain rate which is a local phenomena.

For NG fuelled engines, assuming the same turbulent intensities and length scales exist, the regime plotted in Figure B.2 lies towards smaller $D_\alpha$ because the laminar burning speed of NG is lower than that of gasoline: the laminar flame thicknesses are expected to be approximately equal. The lower $S_L$ of NG implies a larger part of the NG engine regime lies below the $l_k = \delta_L$ line. This suggests that NG engine operation might be less stable and more sensitive to the in-cylinder flow field characteristics under light load conditions (Bradley et al., 1996).

The $L_{sr}$ for NG is, however, lower than the $L_{sr}$ of gasoline when $\lambda \geq 1$. Under light load conditions when $\lambda \geq 1$, the limiting reactant is the fuel and $L_{sr}$ is dependent on the $Le$ of the fuel. The Lewis number for $\lambda = 1.25$ propane-air and gasoline-air
mixtures are 2 and 3.5 respectively (Tromans. 1983). The larger Lewis number for gasoline is due to the low mass diffusivity of the large molecular weight fuel: the Lewis number for NG is expected to be lower than that of propane. In fact, when $\lambda \geq 1$ the $L_{sr}$ of NG is negative for representative conditions at the time of ignition, whereas $L_{sr}$ of gasoline is positive (Bradley et al., 1996; Daneshyar et al., 1983; Arcoumanis et al., 1998). This implies that flame stretch decreases the laminar burning speed of gasoline, but increases that of NG (Daneshyar et al., 1983). The stability regions of gasoline and NG fuelled engines are, therefore, different (Arcoumanis et al., 1998). One of the issues under investigation in this project is the influence of different in-cylinder flows on the stable operation of SI engines fuelled by NG.
Appendix C

Simultaneous measurements

A brief outline of the technique used to make the simultaneous velocity-pressure-EFKD measurements is presented in this appendix. A control system had to be developed because the FOsp—pressure and velocity data are acquired on separate PC’s.

1. The TSI rotating machinery resolver, or RMR, takes in the TDC pulses from the BEI shaft encoder and converts the two TDC pulses associated with the four-stroke engine cycle into one pulse per engine cycle: a toggle feature is available to choose the single pulse as either the exhaust or compression TDC: the exhaust TDC pulse is chosen. This single exhaust TDC pulse is used to trigger the pressure-FOsp DAQ.

2. The RMR uses crank degree marker CDM pulses from the shaft encoder as an external clock source to tag the arrival of LDV data points. The shaft encoder CDM pulse train resolution is 0.2 CAD. The same CDM pulse train is used by the pressure-FOsp DAQ system as the external clock source.

3. The IFA750 digital burst correlator has an inhibit gate. Sending a high TTL signal into this gate stops LDV DAQ. DAQ resumes when a low TTL signal is sent to this gate. The pressure-FOsp DAQ Labview virtual instrument is used to configure a line from the digital port to send a high TTL signal to this gate to control the LDV system DAQ timing.
4. The IFA750 sends 5 words through the DMA cable to the AT-DIO-32F LDV DAQ board for every velocity data point in the configuration used to acquired the in-cylinder velocity data. Receive and acknowledge handshaking signals are also passed between the IFA750 and AT-DIO-32F board for each data word passed between the hardware: the acknowledge signal is routed to counter 1 of the AT-MIO-16E-1 board. The number of data points acquired by the LDV system is known at a specific time by reading from the counter and dividing by 5.

5. The methodology used to simultaneously measure the velocity-pressure-EFKD data is as follows:

(a) The LDV DAQ system is set to stand-by mode.

(b) The pressure-FOSP DAQ is started. This system first inhibits the LDV system from acquiring data by sending a high TTL signal to the burst gate inhibit.

(c) The LDV DAQ system is started. but it does not start acquiring data because the inhibit is on.

(d) The pressure-FOSP DAQ system reads from counter 1 and records the value. The first value is zero.

(e) The pressure-FOSP DAQ system removes the inhibit from the LDV system. The velocity data acquisition begins at this point. The pressure-FOSP DAQ system waits for the next exhaust TDC trigger pulse to start the pressure-FOSP DAQ. Data is acquired for one engine cycle.

(f) Once one engine cycle of pressure-FOSP data is acquired, the pressure-FOSP DAQ system again inhibits the LDV DAQ system.

(g) Counter 1 is read and the value is recorded by the pressure-FOSP DAQ system.

(h) Steps 5c–5g are repeated until the user specified number of cycles are acquired.
(i) The inhibit to the LDV DAQ system is turned off to allow the LDV system to finish acquiring the user specified number of velocity data points.

6. The pressure-FOSP cycles are then matched up with the velocity cycles in post-acquisition data processing: the matching is done based on the data counts read from counter 1 data. That is, the individual pressure-FOSP cycle is aligned with the velocity cycle which is enclosed by the counter 1 data values read immediately before and after the pressure-FOSP cycle.
Appendix D

Wavelet analysis

Much effort in the past has been given to separating IC engine instantaneous flow field, as measured by single point measurements techniques, into mean and turbulence components. The objective here is not, however, to separate the instantaneous velocity data into mean and turbulence components. Rather, we wish to examine the evolution of the flow field during the combustion process. Wavelet basis functions are ideally suited for this purpose because they act as a filter bank and localise the signal in both the time and frequency domains: this localising property allows the transient evolution of the flow field to be viewed through local coefficients (Burrus, Gopinath, & Guo. 1998). This is in contrast to the standard time, or crank degree, domain or standard frequency domain basis, where, a fine frequency domain resolution results in a coarse time domain resolution, or vice versa: this is unacceptable in non-stationary engine flows because one can no longer view transient events of the evolutionary process.
D.1 Discrete wavelet transform

A discrete signal $f(n)$ of length $N = 2^J$ and sampled at a rate $f_s$, can be represented by:

$$f(n) = \sum_k c_{j0}(k) \varphi_{j0,k}(n) + \sum_k \sum_j d_j(k) \psi_{j,k}(n).$$

where the coefficients $c_{j0}(k)$ and $d_j(k)$ are the discrete wavelet transform of $f(n)$. The functions $\varphi_{j0,k}(n)$ and $\psi_{j,k}(n)$ are the orthogonal wavelet scaling and basis functions, respectively. The discrete wavelet transform projects the signal $f(n)$ onto a 2D time-frequency plane, where the scale is represented by the parameter $j$ and the location in time is represented by the translation parameter $k$. The magnitude-squared of the DWT coefficients $|d_j(k)|^2$ represent the signal energy localised in the time-frequency plane.

The scale associated with the coefficients is $T/2^j$, with $T = N/f_s$ as the duration of the discrete sampling interval; $j_0$ defines the coarsest scale. For discrete signals of finite length $j_0 = 0$, such that the scaling coefficient $c_0(0)$ is proportional to the DC component of the signal: the finest scale is $T/2^{J-1}$, The frequency range associated with scale $j$ is $[f_s/2^{J-j}, f_s/2^{J-j-1}]$; these are a logarithmic set of bandwidths, where the ratio of bandwidth to the centre frequency is a constant. The translation parameter $k$ represents the location in time of the wavelet coefficient $d_j(k)$, where the translations are made in steps of $2^j/T$. There are $2^j$ translations at scale $j$, and hence the translations become larger as the scale increases.

There are a large number of wavelet scaling and basis functions which can satisfy the requirements needed to form an orthogonal basis (Daubechies. 1992): the choice of wavelet basis functions is signal dependent. Here we are interested in accelerations in the flow; three wavelets are examined in this work. the Daubechies 4, 20 and the Coiflet 12. Figure D.1 show a plot of the scaling and wavelet basis function recursion coefficients.
Figure D.1: Recursion coefficients of scaling and wavelet basis functions.
Multiresolution properties of wavelets} The basic recursive formulas for the scaling and wavelet basis functions are expressed in terms of the scaling recursion coefficients \( h(n) \) and wavelet recursion coefficients \( h_1(n) \) as

\[
\varphi(2^{j-1}l) = \sum_nh(n)\sqrt{2}\varphi(2^j - n) \tag{D.2}
\]

\[
\psi(2^{j-1}l) = \sum_nh_1(n)\sqrt{2}\varphi(2^j - n) \tag{D.3}
\]

These recursive relationships show the multiresolution features of the scaling and wavelet basis functions. That is, the space spanned by the scaling functions at a lower resolution \( \varphi(2^{j-1}l) \) can be expressed in terms of the scaling functions at a higher resolution \( \varphi(2^j l) \): the \( \sqrt{2} \) factor maintains the norm. The same holds true for the wavelet basis functions, where \( \psi(2^{j-1}l) \) can also be expressed in terms of the scaling function at higher resolution. The scaling \( \varphi(2^{j-1}l) \) and wavelet basis functions \( \psi(2^{j-1}l) \) must be orthogonal: this is ensured by the appropriate choice of the wavelet and scaling coefficients \( h(n) \) and \( h_1(n) \): one requirement is that the fourier transform of the wavelet and scaling recursion coefficients satisfy \( H(\omega) = H_1(\omega + \pi) \). The implication is that if \( \varphi(2^j l) \) spans a space \( \mathcal{V}_J \), \( \varphi(l) \) spans \( \mathcal{V}_{J-1} \) and \( \psi(l) \) spans the difference such that \( \mathcal{V}_J = \mathcal{V}_{J-1} \oplus \mathcal{W}_{J-1} \). It turns out that \( h(n) \) and \( h_1(n) \) act as low and high pass digital filters on the space \( \mathcal{V}_J \).

This multiresolution feature results in (Burrus et al., 1998).

\[
c_j(k) = \sum_mh(m - 2k)c_{j-1}(m) \tag{D.4}
\]

\[
d_j(k) = \sum_mh_1(m - 2k)c_{j-1}(m) \tag{D.5}
\]

which shows how the lower scale DWT expansion coefficients are derived from the higher scale expansion coefficients. Namely, that the recursion coefficients are convolved with the higher scale coefficients and then down sampled by a factor of 2. The question is then how to obtain estimates of the higher scale coefficients. It turns out that for high enough scale the scaling functions act as delta functions such that the
expansion coefficients $c_J(k)$ are samples of the signal itself. If the signal is sampled at a rate greater than the bandwidth of the signal, then the samples of the signal are good approximations to the expansion coefficients $c_J(k)$. Equations Equation D.4 and D.5 are then applied to calculate the DWT expansion coefficients at lower scales.

Filter banks and spectral leakage

As mentioned, the recursion coefficients $h(n)$ and $h_1(n)$ are low and high pass digital filters. These filters act on the signal at successively larger scales. In digital signal processing terminology they are quadrature mirror filters (Lyons, 1997): they act as a two-band filter bank. This filter bank property is illustrated in Figure D.2 for the Daubechies-4, 20 and Coiflet-12 wavelets, where the magnitude of the filters at each scale is projected onto the frequency domain: the scale $T/2^j$ is associated with filter bank $W_j$. Filtering at these scales occurs at each translation $k$.

The compactness of the filter banks in the frequency domain depends on the type of wavelet basis function. The Daub4 filter bank shows a much slower filter band roll off, which is demonstrated by substantial overlap of the filter banks at 3 dB reduction. The roll off rate of the filters is substantially greater for the Daub20 and Coif12 filter banks: the Daub20 has the least overlap between filter banks. The slow roll off in the filter banks results in spectral leakage from outside the desired filter bank region, and is an issue in SI engines because of the possibility of contamination of small scale information by large, energetic events.

Display of the discrete wavelet transform

The DWT of a signal is a 2D representation in the time-frequency plane. There are various ways in which to display the 2D expansion coefficients: the most basic method is a three dimensional contour plot, where the magnitude of the coefficients is plotted over the $(f_s/2^{j-1}, 2^j k/T)$ plane. A second method is to plot the projection of the signal at each scale, that is, the time signal is projected onto each $W_j$ space. An example of the contour and $W_j$ projections are shown in Figures D.3 and D.4: the velocity data was taken from the SCV6 3.1 L engine velocity at 70 kPa and 900 RPM.
Figure D.2: Filter bank representation of wavelet basis functions. The scale $T/2^j$ is associated with filter bank $\mathcal{W}_j$. 

(a) Daubauchies 4

(b) Daubauchies 20

(c) Coiflet 12
The contour plot is useful in that the evolution of the energy in the flow field over time and frequency can be seen. To interpret this plot, note i) that the light coloured regions are associated with high energy and the dark regions with low energy; and ii) the distinct time-frequency plane tiling associated with a two-band wavelet must be taken into account. At low frequencies there is a fine frequency resolution, but the time resolution of the energy is coarse: at high frequencies the frequency resolution is coarse and the time resolution is fine. For example, at the coarsest scale, that is, lowest frequency, there is only one translation over the CAD window. At the highest frequency, there are 64 translations. This is a 'natural' phenomena since the low frequency energy represents the average energy throughout the window, whereas the high frequency energy is highly localised in extent. When visualising the variations in energy in the time-frequency plane, this tiling must be considered in the interpretation.

One problem with the contour plots is, however, that high energy peaks can mask out the variations in the low energy regions unless an high resolution contour grid is used. This is not immediately obvious from the plot in Figure D.3, but this data was low pass filtered at 28.8 (engine cycle)$^{-1}$ to remove the large energy of the low frequency scales. This problem with the contour plots can be overcome through the use of the scale plots, that is, the $W_j$ projections shown in Figure D.4. The scale plots are slices, perpendicular to the $(f_s/2^{j-1}, 2^j k/T)$ plane, across the contour plot at constant scale. The scale plots are used to visualise the variation in signal energy at each scale with respect to crank degree. The usefulness comes through, for example, if the energy at the larger scales masks out the variations in the smaller scales on the contour plot, but the scale plots, since each plot can have different limits, give an idea to how the energy varies at each scale.

Another useful technique to prevent masking out variations in the small scale energy is to make a slice, parallel to the $(f_s/2^{j-1}, 2^j k/T)$ plane, at one energy level. This is useful to show which regions have energy above the cut-off energy level: the energy level can be chosen based on the energy in the smaller scales.
Figure D.3: Contour plot of DWT coefficients $|d_j(k)|^2$. Coiflet 12 wavelet basis function. u component ensemble average turbulence velocity data from scv6 3.1 L engine. The instantaneous velocity data was low pass filtered at 28.8 (engine cycle)$^{-1}$. 
Figure D.4: Scale plot of DWT coefficients $|d_j(k)|^2$. Coiflet 12 wavelet basis function. U component ensemble average turbulence velocity data from SCV6 3.1 L engine. The instantaneous velocity data was low pass filtered at 28.8 (engine cycle)$^{-1}$. 
D.2 Wavelet spectral leakage

Before we can examine the evolution of the in-cylinder flow field over time and scale, there is a spectral leakage issue which must first be addressed. The spectral leakage problem is the result of the slow roll off rate of the filter banks of the $W_i$ spaces, see Figure D.2. To demonstrate the problem, consider the $u$ and $v$ component scale plots shown in Figures D.5 and D.6: a Daubechies-4 wavelet basis function was used. The energy for the scale plots of Figure D.5 was calculated for all the flow energy, whereas the scale plots of Figure D.6 were calculated from the flow energy $f < 28.8$ (engine cycle)$^{-1}$: this cut-off frequency is equivalent to a 25 CAD scale.

One would expect the only significant difference between the two scale plots to be for scales $\geq 25$ CAD, since the energy above this scale has been removed for the data plotted in Figure D.6. However, a significant difference is noted in the mid-range scales, namely, 12.8, 6.4, and 3.2 CAD. The plot in Figure D.5 shows an increase in the 12.8 CAD scale energy after 370 CAD, whereas the scale plot of Figure D.6 shows a slow decrease in energy for $\theta > 350$ CAD. Similar differences are observed for the 6.4 and 3.2 CAD scale energy.

The problem is that the energy of the larger scales is an order of magnitude larger than the mid-range scales in this flow. This implies that although the filter bank roll off towards lower frequencies, see Figure D.2, associated with a mid-range scale might be small in magnitude, a significant bias can occur where energy from the larger scales leaks down into the mid-range scales.

We need to minimise this spectral leakage bias. This can be done by appropriate choice of wavelet basis function and low pass filter.

Minimising wavelet spectral leakage If we consider the spectral leakage in a qualitatively fashion, we would expect a broadening of the energy across the spectrum for a wavelet with worse spectral leakage. For a given spectrum $G_{uu}(f)$, if we compared the estimate of $\hat{G}_{uu}(f)$, this would translate into $\hat{G}_{uu}(f) > G_{uu}(f)$ for the high frequency components, say $f > f_m$, and $\hat{G}_{uu}(f) < G_{uu}(f)$ for the low frequency components.
The DWT expansion coefficients can be projected onto the frequency domain to obtain an estimate of $G_{uu}(f)$: the approximation is coarser than the estimate obtained using the FFT. Since we don't have $G_{uu}(f)$, the simplest way to determine which wavelet basis function, Daubechies-4,-20 or Coiflet-12, performs the best is to compare the estimates to one another. The estimate with the lowest high frequency energy, and largest low frequency energy is the best wavelet to use.

The projections of DWT onto the frequency domain for the Daubechies-4,-20 and Coiflet-12 wavelet basis functions were calculated, where all the energy of the flow field in the 335–385 CAD window was considered. The fraction of energy resident in the DC, low, mid-, and high frequencies is listed in Table D.1. The results indicate that more energy is resident in the DC component, that is, in the DWT expansion coefficient $c_0(0)$, for the Daubechies wavelets relative to the Coiflet-12 wavelet. The Coiflet-12 wavelet, however, has the most large scale energy resident and the least mid and small scale energy. These results indicate that the Coiflet-12 wavelet basis functions perform best when minimisation of spectral leakage into the mid and small scales is used as the criteria.

Based on the slower roll off of the Daubechies-4 filter banks, see Figure D.2, it is not surprising that the Daubechies-4 wavelet basis functions allow the most spectral leakage. Based on the fast roll off characteristics of the Daubechies-20 wavelet basis functions, would have expected this wavelet to perform the better than the Coiflet-12 wavelet. There are two characteristics of the Coiflet-12 wavelets which might explain the different behaviour. The first is that the Coiflet-12 scaling and wavelet basis functions are nearly symmetric, where the Daubechies-20 are clearly not, see Figure D.1. The second difference is that the Coiflet-12 wavelets have fewer recursion coefficients than the Daubechies-20 wavelets. This might cause the smearing of the energy across frequencies.

**Low pass filter** To minimise the spectral leakage, the Coiflet-12 wavelet basis functions should be used when visualising the evolution of the flow field through the DWT. To further reduce the bias, the energy of the large scales should be filtered out
Table D.1: Comparison of DWT expansion coefficients projected onto the frequency domain. The Daubechies-4-20 and Coiflet-12 wavelet basis functions are used. u component ensemble average turbulence velocity data from SCV6 3.1 L engine at 70 kPa and 900 RPM. All the flow energy in 335-385 CAD window is considered.

<table>
<thead>
<tr>
<th></th>
<th>Frequency range ((engine cycle)$^{-1}$)</th>
<th>Scale range (CAD)</th>
<th>Energy (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>DC</td>
<td>0-9.1</td>
<td>0</td>
<td>Daub 4 Daub 20</td>
</tr>
<tr>
<td>Low</td>
<td>9.4-75</td>
<td>25.6-51.2</td>
<td>29.5 29.1</td>
</tr>
<tr>
<td>Mid</td>
<td>75-300</td>
<td>3.2-12.8</td>
<td>56.2 56.8</td>
</tr>
<tr>
<td>High</td>
<td>&gt;300</td>
<td>&lt;3.2</td>
<td>12.0 11.8</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$f_{lp}$ (engine cycle)$^{-1}$</th>
<th>Fraction of energy (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0.03</td>
</tr>
<tr>
<td>25</td>
<td>0.80</td>
</tr>
<tr>
<td>28.8</td>
<td>3.5</td>
</tr>
<tr>
<td>30</td>
<td>6.1</td>
</tr>
</tbody>
</table>

Table D.2: Amount of energy contained in $W_2$ space.

before applying the DWT to the velocity data. The question arises as to what low pass frequency should be used. First off, we know that 3.2-12.8 CAD are the scales of interest: the evolution at larger scales cannot be visualised because the number of translations is $k \leq 2$, and the scales smaller than 3.2 CAD are of interest, but are biased, see Appendix I.

The low pass frequency is chosen such that if the filter bank acted on a spectra of white noise, then the energy within the $W_2$ space over $0 < f < f_{lp}$ is less than 1%: the wavelet space $W_2$ is associated here with a scale of 12.8 CAD. In this way, any energetic frequencies $f < f_{lp}$ are filtered out, but at the same time, the wavelet space $W_2$ loses less than 5% of its energy if the energy at the lower frequencies did not introduce a bias. The fraction of energy vs. values of $f_{lp}$ is listed in Table D.2. A low pass filter of $f_{lp} = 28.8$ (engine cycle)$^{-1}$ is chosen, which is the centre frequency associated with the next lower wavelet space $W_1$. 
Figure D.5: Scale plot of DWT coefficients $|d_j(k)|^2$. Daub4 wavelet basis function. $u$ component ensemble average turbulence velocity data from $scv6$ 3.1 L engine. Ensemble averaging was used to calculate the turbulence.
Figure D.6: Scale plot of DWT coefficients $|d_j(k)|^2$. Daub4 wavelet basis function. u component ensemble average turbulence velocity data from SCV6 3.1 L engine. All the flow energy $f < 28.8$ (engine cycle)$^{-1}$. or scales $< 25$ CAD. is considered.
Appendix E

Velocity analysis

Although in-cylinder velocity measurements have been made for over two decades, the issue of how to decompose the flow into mean and turbulence velocities remains unresolved. The problem stems from the fact that engine flows are non-stationary and are characterised by high turbulence intensities and quasi-periodic variations in the mean; the main difficulty in decomposing the flow lies in differentiating between mean and turbulent flow energy because of overlapping mean and turbulence scales.

The decomposition techniques generally used in engine flows include the Ensemble and Cyclic averaging techniques. Previous work by Sullivan et al. (1999), however, has shown that these methods do not decompose the flow into physically meaningful components, but an arbitrary mean and turbulence. Ensemble averaging uses a statistical Reynolds based filtering operator; Liou et al. (1984) were the first to show that Ensemble averaging over-estimates the turbulence energy by assigning quasi-periodic bulk flow energy as turbulence. Cyclic averaging, referred to here as Scale based averaging, uses a low pass filtering scheme to calculate the turbulence. A scale separating the bulk flow and fluctuating velocities defines the low pass filter. Liou et al. (1984). Lorenz and Prescher (1990) and Catania and Mittica (1990) have all introduced different methods to filter the measured velocity into mean and turbulent flow. All Scale based averaging techniques do not include any low frequency events as turbulence. The choice of filtering scale, therefore, biases the turbulence scale information. Neither the Ensemble nor Scale based techniques is entirely satisfactory.
in decomposing engine flows because of the biases they introduce.

Ideally, the technique used to decompose engine flows should be able to identify the quasi-periodic bulk flow variations. The investigation of coherent structures in stationary turbulence has focused on the identification of energetic, quasi-periodic events. The techniques used to identify coherent structures in stationary flows can also identify the quasi-periodic events in engine flows. Most of these methods of structure identification, however, require multi-point measurements. For example, Hayakawa and Hussain (1987, 1989) decompose the measured multi-point velocities in terms of vorticity or energy. It is impossible, however, to measure vorticity or extract spatial correlations from single point LDV measurements. For single point velocity measurements, Higuchi et al. (1994) have shown wavelet analysis to be a very effective method of extracting scale information and in identifying coherent, quasi-periodic events. Wavelet basis functions act as a filter bank and localise the signal in the time and frequency domain: they are effective in identifying quasi-periodic events as the localising property allows transient events to be modelled with a small number of coefficients. This is in contrast to Fourier basis functions which require a large (infinite) number of coefficients to model a transient event. Wavelet based decomposition methods have been successfully applied to single point data in a number of flows including the wake of two flat plates by Higuchi et al. (1994) and the three-dimensional wall jet by Sullivan and Pollard (1996).

Wiktorsson et al. (1996) and Sullivan et al. (1999) applied wavelet based filtering to SI engine flows. Both define coarse wavelet scales as the mean flow, and fine scales as the turbulence. Due to the filter bank property of the wavelets, this separation technique effectively divides the flow into high and low frequencies, and is the equivalent to the Scale based technique in that a filtering scale is introduced: the wavelet based decomposition is, however, more efficient numerically.

To avoid introducing a filtering scale a new turbulence decomposition technique, based on discrete-time wavelet transforms, is proposed using an Energy based filtering technique to separate high energy events within the turbulence. This procedure is similar to that proposed by Lancaster et al. (1976) where the turbulence velocity
calculated from the ensemble average was interpreted to consist of low and high
frequency components associated with cyclic variations in the mean and turbulence,
respectively. Here, we interpret the Ensemble based turbulence velocity to consist of
high and low energy events, where high energy local events are associated with the
quasi–periodic variations in the mean.

In the following subsections, the details of the Ensemble, Scale and Energy based
averaging techniques are presented. The different techniques are compared and criti-
tiqued based on the degree to which turbulence is biased by the energy from quasi-
periodic variations in the bulk flow: the coherence and phase–conditioned correlation
functions are used to determine the likelihood of the bias being present in the tur-
bulence.

### E.1 Ensemble based averaging

Ensemble based averaging defines the bulk, or mean, velocity as an exactly repeatable
phase average for all engine cycles. The mean velocity is calculated as the ensemble
average over $N$ cycles:

$$\bar{U}(\theta) = \frac{1}{N} \sum_{k=1}^{N} \bar{u}(\theta, k)$$

(E.1)

where $\bar{U}(\theta)$ is the mean velocity, $\bar{u}(\theta, k)$ is the instantaneous velocity in cycle $k$, and
$\theta$ is the crank degree, or phase. The fluctuating velocity is the difference between the
instantaneous and mean velocities

$$u(\theta, k) = \bar{u}(\theta, k) - \bar{U}(\theta).$$

(E.2)

and the statistical properties of the turbulent flow are calculated either over $N$ cycles.

$$[u'(\theta)]^2 = \frac{1}{N} \sum_{k=1}^{N} [u(\theta, k)]^2.$$  

(E.3)
or within an engine cycle $k$ over a crank angle range where the statistical properties of the flow are assumed to be stationary:

$$[u'(\theta_o, k)]^2 = \frac{1}{\Delta \theta} \sum_{\theta=\theta_o - \frac{\Delta \theta}{2}}^{\theta=\theta_o + \frac{\Delta \theta}{2}} [u(\theta, k)]^2.$$  \hspace{1cm} (E.4)

The crank angle range is centred on $\theta_o$ and is of extent $\Delta \theta$.

If the phase–conditioned turbulence velocities are calculated using Equation E.3, a bias is introduced by the energy of the quasi–periodic mean flow variations. Ar-coumanis and Whitelaw (1987) give: i) precessing swirl, ii) tumble and iii) flapping intake or squish jets, as examples of energetic events present in engine flows which exhibit cyclic variations in their phasing, amplitude and orientation. To demonstrate how the turbulence is biased, first consider the Ensemble based mean velocities at one point in the L-head research engine during the intake stroke as shown in Figure E.1(a). The non-stationary mean velocities associated with the intake jet are evident. Cycle to cycle variations in the phasing and orientation of this intake jet with respect to the measurement point introduce a bias into the turbulence. To show how these cyclic variations in the intake jet flow might bias the turbulence, the fluctuating velocity calculated from the Ensemble average for Cycle 15 of the data (Figure E.1(b)) shows two large scale, relatively high energy events at 30-40 CAD and 60-80 CAD. Superimposing the events at 30-40 and 60-80 CAD onto the mean flow indicates that they are in phase with the large accelerations in the mean: jitter in the phase or orientation of the intake jet is probably the cause of these large scale drifts.

To demonstrate this possibility, the bias introduced by phase jitter is estimated through a high energy event modelled as a sine wave with a 36 CAD period, as shown in Figure E.2(a), or $f = 20$ (engine cycle)$^{-1}$, that is, it is an event whose period can repeat itself 20 times in one 720 CAD engine cycle. Random jitter in the phase, that is, the starting crank angle, over 18 CAD was allowed, a parallel to variations in real engines, and the turbulence was modelled as white noise. One hundred cycles of data were generated and Ensemble averaging applied. The results are compared
with the input signal statistics in Figure E.2 and demonstrate, as expected, that the Ensemble average fails to capture the individual cycle mean. The Ensemble average is of significantly lower magnitude and longer duration. The resulting fluctuating velocity of modelled cycle number 10, plotted in Figure E.2(b), shows a large scale local event that is the result of the cyclic variations in the phase of the energetic event, and belongs to the mean flow. Ensemble averaging over-estimates the turbulence velocity where cyclic variations occur (Figure E.2(c)). The energy associated with these cyclic variations are within a frequency band width of 10–100 (engine cycle)$^{-1}$, where the Ensemble average turbulence PSD over-estimates the turbulence energy by a factor of approximately 50 (Figure E.2(d)). The turbulence is biased in a broad range of frequencies, which is due to the fact that a large number of Fourier frequencies are need to represent a transient, local event. This simple model demonstrates that the quasi-periodic variations in the mean can significantly bias the turbulence energy, and that the bias can extend over a broad range of frequencies.

Similar biasing of the turbulence occurs if the orientation of the intake jet with respect to the measurement volume changes from one cycle to the next. For example, consider if the phase of the intake jet remained constant, but the orientation of the intake jet changed from one cycle to the next. The proportion of the mean velocity of the intake jet projected onto the $u$ and $v$ component measurement directions would vary from one cycle to the next. These quasi-periodic variations in the $u$ and $v$ component mean velocities bias the turbulence.

There is an additional bias introduced because the intake jet is non-stationary, non-homogeneous and anisotropic. In this case, the statistics of the turbulence depend (i) on the phase of the measurement, and (ii) on the location and orientation of the measurement within the intake jet. The phase jitter and variations in the orientation of the jet with respect to the measurement location, therefore, cause quasi-periodic turbulence broadening biases. The Ensemble based turbulence statistics at a particular phase $\theta$ and location $\mathbf{r}$ are statistics averaged over time and space, where the extent of the average over time depends on the amount of phase jitter, and the extent of the average over space depends on the variations in the orientation of the jet.
with respect to the measurement point. To remove these quasi-periodic turbulence broadening biases, a conditioned ensemble average is needed, where conditioning with respect to intake jet phasing and orientation is needed. This can only be done with multi-point measurements.

To demonstrate that significant bias is introduced to the turbulence when Ensemble averaging is used to calculate the turbulence for IC engine flows, the cross correlation $C_{uv}(\theta)$ and coherence function statistics are examined. $C_{uv}(\theta)$ is defined through the statistics of $u(\theta, k)v(\theta, k)$ (Bendat & Piersol. 1986), as

$$
C_{uv}(\theta) = \frac{1}{N} \overline{u'(\theta) \cdot v'(\theta)} \sum_{k=1}^{N} [u(\theta, k)v(\theta, k)]
$$

(E.5)

Since $C_{uv}(\theta)$ quantifies the asymmetry in the joint $\langle uv \rangle$ PDF and can be reduced to statistically insignificant values by appropriate rotation of the measurement coordinate system (Tennekes & Lumley. 1972). The flow is anisotropic if a significant value of $C_{uv}(\theta)$ is found. Anisotropy is associated with the large scale processes. In engine flows, much of the large scale energy is associated with the mean flow. Significant $C_{uv}(\theta)$ therefore implies that significant large scale processes, the energy of which might belong to the mean, remain as turbulence. In addition to this, assuming there are cyclic variations in the flow, the probability that the turbulence broadening biases are present is high.

The coherence function $\gamma_{uv}$ is defined as

$$
\gamma_{uv}^2(f, \theta_0, \Delta \theta) = \frac{|G_{uv}(f, \theta_0, \Delta \theta)|^2}{G_{uu}(f, \theta_0, \Delta \theta)G_{vv}(f, \theta_0, \Delta \theta)} \leq 1
$$

(E.6)

$$
G_{uv}(f, \theta_0, \Delta \theta) = \frac{2}{\Delta \theta N} \sum_{k=1}^{N} [X^*(f, k, \theta_0, \Delta \theta)Y(f, k, \theta_0, \Delta \theta)]
$$

(E.7)

$$
X(f, k, \theta_0, \Delta \theta) = \int_{\theta=\theta_0-\frac{3\pi}{4}}^{\theta=\theta_0+\frac{3\pi}{4}} u(\theta, k) \exp(-j2\pi f \theta) d\theta
$$

(E.8)

$$
Y(f, k, \theta_0, \Delta \theta) = \int_{\theta=\theta_0-\frac{3\pi}{4}}^{\theta=\theta_0+\frac{3\pi}{4}} v(\theta, k) \exp(-j2\pi f \theta) d\theta
$$

(E.9)

where $f$ is frequency, expressed as (engine cycle)$^{-1}$. $k$ is cycle number. $\Delta \theta$ is the
crank degree range in which the coherence function is evaluated. and \( \cdot \) denotes a complex conjugate. The coherence function \( \gamma_{ur} \) is interpreted as follows: i) for \( \gamma_{ur}(f, \theta_0, -\Delta \theta) \ll 1 \), \( u \) and \( v \) are on average non-coherent, or uncorrelated, at that frequency within the crank angle range \( \theta = \theta_0 \pm \Delta \theta/2 \); ii) for \( \gamma_{ur}(f, \theta_0, -\Delta \theta) \approx 1 \), \( u \) and \( v \) are on average fully coherent, or correlated, at that frequency within the crank angle range \( \theta = \theta_0 \pm \Delta \theta/2 \). Relatively large values of \( \gamma_{ur} \) at characteristic frequencies in the turbulence suggest the presence of quasi-periodic coherent events at that frequency within the crank degree range considered.

\( C_{ur}(\theta) \) and coherence function plots, as shown in Figures E.3(a) and E.4(a), respectively, where all the flow energy for the intake stroke, 0-180°, is assigned to turbulence. That is, a constant mean velocity with respect to \( \theta \) is assumed. The \( C_{ur}(\theta) \) function shows a strong correlation at the beginning, 0-80 CAD, and the end of the intake stroke, 140-180 CAD, where a correlation coefficient of 0.15 is statistically significant at a 95% confidence level. The correlation peaks occur where strong accelerations and decelerations of the intake jet exist. The coherence function, see Figure E.4(a), shows that energetic coherent events occur at frequencies ranging between 0-120 (engine cycle)\(^{-1} \) and are associated with the intake jet.

The Ensemble average \( C_{ur}(\theta) \) and coherence function plots are shown in Figures E.3(b) and E.4(b), respectively. Significant \( C_{ur}(\theta) \) and coherence remains after Ensemble averaging. The significant coherence indicates that the energy of quasi-periodic variations in the mean remain as turbulence. Thus, the probability of the energy associated with the quasi-periodic events biasing the turbulence energy is high. The same trends are evident when assuming a constant mean throughout the intake stroke. Ensemble averaging does as well as a constant mean, thus implying a problem with the resulting physics and application of Ensemble average results to fluids models or equations.
E.2 Scale based averaging

Scale based averaging techniques estimate the mean by low pass filtering the instantaneous velocity either in the temporal or the frequency domain. The scale based averaging techniques proposed by Liou et al. (1984), Lorenz and Prescher (1990) and Catania and Mittica (1990) use a cut-off frequency where the energy below this frequency is assigned to the mean and the energy above is the turbulence. Fansler and French (1988) argue that the cut-off frequency should be chosen based on the physical scales of interest in the flow, thus the term Scale based averaging. Liou and Santavicca (1985) suggest an alternate method to choose the cut-off frequency based on the highest frequency with significant energy in the power spectral density function PSD of the Ensemble based average. With rapid variations in the mean flow, such as during the intake or exhaust stroke, the Ensemble mean velocity may contain energy at high frequencies. Choosing a cut-off frequency in this way may result in too severe a filtering. Regardless of how the cut-off frequency is chosen, Liou et al. (1984) and Catania and Mittica (1990) state the ensemble average of the individual cycle means, that is, the the low pass velocity, should coincide with the Ensemble based average velocity where \( \chi^2 = 1 \) is expected (Press et al., 1980).

To illustrate the effect of this filtering scheme the velocity data was filtered using a cut-off frequency of 120 (engine cycle)\(^{-1} \) for both the \( u \) and \( v \) velocity components. The cut-off frequency was chosen to filter out the low frequency coherence present in the unfiltered velocity data, see Figure E.4(a). There are no significant differences between the ensemble average of the low pass filtered velocity and the Ensemble based average velocities plotted in Figure E.1(a). A \( \chi^2 \) value of 0.8 is obtained for both the \( u \) and \( v \) velocity components.

The differences between the Ensemble and Scale based averaging techniques become evident when fluctuating velocities are compared. The local large scale, high energy events at 30–40° and 60–80° seen in Figure E.1(b) are not present in Figure E.1(c). The \( C_{ur}(\theta) \) and coherence function plots of the Scale based turbulence velocity are shown in Figures E.3(c) and E.4(c). The coherence function shows that
the low frequency, coherent motions are removed from the turbulence velocities. The energy associated with the low frequency quasi-periodic variations in the mean has been removed. The $C_{uc}(\theta)$ function, see Figure E.3(c), however, indicates that there is significant correlation, that is, $C_{ur} > 0.15$, remaining in crank angle ranges of 0–40° and 150–180°. The remaining high $C_{uc}(\theta)$ implies the flow is anisotropic in the high frequency scales, in this case greater than 120 (engine cycle)$^{-1}$. see Figure E.4(c). Since isotropy would be expected at the higher scales, this suggests these high frequencies are associated with large scale processes and might be associated with the quasi-periodic variations in the mean.

This highlights the main problem with Scale based averaging techniques, namely, that a constant low pass filter is applied to data with a non-stationary convection and turbulence velocity. As the convection and turbulence velocity vary with crank angle, the frequency associated with a length scale will also change. More than one length scale can be associated with a single frequency. Whether the filtering scale is defined in the frequency or crank angle domain is irrelevant. The physical interpretation of what scales are being filtered is difficult and it is uncertain whether the energy near the cut-off frequency is associated with large or small scale motions. The energy associated with a high frequency large scale quasi-periodic mean flow variation can still bias the turbulence. This problem may be partially alleviated by dividing the engine cycle into intervals where flow variations are not as severe. This is the reason why the intake stroke, 0–180°, is analysed separately from the compression/power and exhaust strokes of the engine cycle.

**E.3 Energy based averaging**

Next, a new Energy based approach is used to calculate the turbulence which identifies the main energy contributors in engine flows as the mean and the remainder as turbulence. The physical justification for this technique is based on the results of Arcoumanis and Whitelaw (1987) and Sullivan et al. (1999) who have shown that quasi-periodic variations in the mean velocity of engine flows are associated with
large, energy containing scales, and are not turbulence. The criteria is the removal of high energy local events from the turbulence. The filtering is done on an individual cycle basis using a discrete wavelet transform based energy filter. The intent is to look at the turbulent flow without introducing a filtering scale.

The Energy based averaging technique uses the ensemble average velocity as the first estimate of the individual cycle mean. The first estimate of the turbulence velocity in engine cycle \( k \) is the difference between the instantaneous and ensemble velocities.

\[
\bar{U}_{IC1}(\theta) = \frac{1}{N} \sum_{k=1}^{N} \tilde{u}(\theta, k) \\
u_{EST1}(\theta, k) = \tilde{u}(\theta, k) - \bar{U}(\theta)
\]  

(E.10)  

(E.11)

The discrete-time wavelet transform DWT of \( u_{EST1}(\theta, k) \) is calculated using a wavelet basis function. The Coiflet-12 wavelet is chosen as the Energy based filter wavelet basis function; the results from using different wavelet basis functions are compared in Appendix H.2. The wavelet coefficients of \( u_{EST1}(\theta, k) \) are then low pass filtered on an energy level basis. A cut-off energy level is chosen for each cycle, where the cut-off energy is chosen as a fraction of the highest energy wavelet coefficient in cycle \( k \). Any coefficients with energy above this cut-off level are assigned to the mean. The remaining coefficients are turbulence. This separation of \( u_{EST1}(\theta, k) \) into high and low energy components is given by \( u_{EST1}(\theta, k) = u_{HE}(\theta, k) + u_{LE}(\theta, k) \). A systematic method, described in the Appendix H.1, is used to choose the cut-off energy level. The inverse-DWT of \( u_{HE}(\theta, k) \) is added onto the Ensemble based average velocity giving the final estimate of the individual cycle mean as

\[
U_{IC2}(\theta, k) = \bar{U}_{IC1}(\theta) + u_{HE}(\theta, k)
\]

(E.12)

which leads to the definition of the fluctuating velocity as

\[
u(\theta, k) = \tilde{u}(\theta, k) - U_{IC2}(\theta, k).
\]

(E.13)
This procedure is similar to that proposed by Lancaster et al. (1976) where the turbulence velocity calculated by the Ensemble based average was interpreted to consist of low and high frequency components associated with the cyclic variations in the mean and turbulence, respectively.

E.4 Comparison of the Scale and Energy based techniques

The Energy based results are compared with Ensemble and Scale based results: the ensemble average of the individual cycle means are indistinguishable for the three techniques. The $\chi^2$ values are 0.8 and 0.25 for the Scale and Energy based results respectively. The coherence function plots, Figure E.4, indicate that the significant low frequency coherent events are removed by both the Scale and Energy based techniques. The $C_{ur}(\theta)$ functions, see Figure E.3, indicate that most of the $\langle uv \rangle$ correlation has been removed by the Energy based averaging technique. Significantly higher correlations still exist in the Scale based averaging results, most notably in the 10-40° and 150-180° crank angle ranges where large accelerations in the mean flow exist, see Figure E.1(a). The lower values of $C_{ur}(\theta)$ for the Energy based results implies that the turbulence energy is nearly anisotropic and that the probability of the turbulence being biased by quasi-periodic flow variations is significantly less in the Energy based results: the Energy based flow is nearly isotropic.

There are differences between the fluctuating velocities calculated on an individual cycle basis, as shown in Figure E.1. Both the Scale and Energy based filtering remove the large scale energetic structures evident in the 40-50 CAD range of the Ensemble based turbulence fluctuating velocity shown in Figure E.1(b). Low frequency, low energy local events are still present in the Energy based plot, but not in the Scale based plot. This is shown by the PSD's plotted in Figure E.5 where the Scale based technique has removed all the low frequency energy from the turbulence while the Energy based technique retains some low frequency energy as turbulence. Comparison of the Energy
based PSD plot. Figure E.5(b). to that of the Scale based PSD. Figure E.5(a). indicates that the Energy based filtering removes energy from across the spectrum. whereas the Scale based filtering removes only low frequency energy. The Energy based filtering scheme does not bias the PSD directly. whereas Scale based filtering does. The autocorrelation function and turbulence time scales are biased by the Scale based technique in a similar manner. This biasing of the scale information does not occur for the Energy based techniques.

Comparison is made between the turbulence velocities calculated from Equation E.3 and plotted in Figure E.6: the results from the Ensemble. Scale and Energy based averaging techniques are shown. As expected. the Ensemble turbulence velocities are greater than both the Scale and Energy turbulence velocities. The Scale based turbulence velocities are lower in magnitude. but still follow the same general pattern of the Ensemble based results: an initially high turbulence velocity begins to decay before rising and falling again. Both the Ensemble and Scale based results indicate approximately 10–15% deviation from isotropy in the 30-150 CAD range. The Energy based turbulence velocities are approximately stationary and isotropic throughout the intake stroke. notably different from the Ensemble and Scale based results. By the end of the intake stroke. 150-180 CAD. the turbulence velocities from the three techniques converge again to an equal isotropic level.

The Energy based technique associates the high energy events of the intake jet to the mean: the turbulence is nearly stationary and isotropic. The Ensemble and Scale based methods recognise the high energy events as turbulence. Since the Scale based technique has filtered out all the low frequency energy, this suggests that there is high frequency coherent energy in the intake stroke: the frequency of the event may be high. but the size of the structure may still be large because of the high convection velocities in the intake jet. The remaining high frequency energetic events might be associated with quasi-periodic variations in large scale processes. and suggests that the Scale based turbulence energy is biased by the energy of quasi-periodic events.
Summary  The Energy based filtering technique was developed to remove the cyclic variations in the mean from the turbulence. The Energy based filter defines high energy events as part of the coherent turbulence with the remainder shifted to non-coherent turbulence. The Energy based results give a different view of IC engine flows as compared with the Scale and Energy based averaging techniques. The Ensemble and Scale based techniques have a high probability of being biased by quasi-periodic variations in the flow, whereas the Energy based technique is not. The Energy based technique, however, removes all energy containing structures to the mean, such that the turbulence is nearly isotropic and stationary. Physically, given that the direction of the intake jet is not constant, the use of Energy based methods assists in removing turbulence biases due to the cyclic variations in the phase and orientation of the flapping jet seen at a fixed measurement point at the expense of removing all structure from the turbulence. The only other way to remove the turbulence broadening biases is through conditional ensemble averaging of the flow based on the phase and orientation of the jet, which can only be done with multi-point measurements.
Figure E.1: Ensemble mean velocities and the fluctuating u-component fluctuating velocity in cycle no. 15. Motored operation at \( \bar{\Omega} = (0\, -25\, 15) \), 900 RPM and \( N = 200 \) cycles.
Figure E.2: Ensemble average results. The input signal is one period of a sine wave with frequency 20 (engine cycle)$^{-1}$ and cyclic variations in the phase of the event. Statistics were calculated for $N = 100$ cycles.
(a) All energy assigned to turbulence.

(b) Ensemble average based filtering.

(c) Scale based filtering. Low pass filtered at 120 (engine cycle)$^{-1}$.

(d) Energy based optimal filtering. (u,v) filtered at (14.11)$\%$ using a Coiflet-12 wavelet.

Figure E.3: Normalised $\langle uv \rangle$ correlation of the turbulence as calculated by different flow decomposition techniques. A value of 0.15 is statistically significant at a $95\%$ confidence level. Motored operation at $F = (0.25, 15)$, 900 RPM and $N = 200$ cycles.
(a) All energy assigned to turbulence.  
(b) Ensemble average based filtering. 

(c) Scale based filtering. Low pass filtered at 120 (engine cycle)$^{-1}$.  
(d) Energy based optimal filtering.  
(u,v) filtered at (14.11)% using a Coiflet-12 wavelet. 

Figure E.4: Coherence functions of the turbulence as calculated by different flow decomposition techniques. Motored operation at $\bar{\tau} = (0, -25, 15)$. 900 RPM and N = 200 cycles.
(a) Scale based filtering. Low pass filter at 120 (engine cycle)$^{-1}$.

(b) Energy based filtering. (u,v) filtered at (14.11)\%.

Figure E.5: Power spectral density functions after filtering. Motored operation at $\bar{\tau} = (0, -25, 15)$. 900 RPM and $N = 200$ cycles.
(a) Ensemble average based filtering

(b) Scale based filtering. Low pass filter at 120 (engine cycle)$^{-1}$.

(c) Energy based optimal filtering. (u,v) filtered at (14.1)$\%$ using a Coiflet-12 wavelet basis function.

Figure E.6: Turbulence velocity after filtering. Motored operation at $J = (0, -25, 15)$, 900 rpm and $N = 200$ cycles.
Appendix F

Mass burn analysis sub-models

The mass burn analysis of the in-cylinder pressure needs thermodynamic sub-models to account for heat transfer, crevice effects, and residual gas effects.

F.1 Heat transfer correlations

The Woschni correlation (Woschni, 1967) is used to model the convective heat transfer effects.

\[ h_c = 131C_1(B)^{0.3}(p)^{0.8}(T)^{-0.53}(w')^{0.8} \]  

(F.1)

where \( p \) is the cylinder pressure (atmo) and \( T \) is the average cylinder temperature. The characteristic speed is.

\[ w = 2.28(\overline{S}_p + \overline{u}_{swirl}[m\ s^{-1}] + 0.0034C_2 \left( \frac{V_{disp}}{V_{IVC}} \right) \left( \frac{p_f - p_m}{p_{IVC}} \right) T_{IVC}[K] \]  

(F.2)

where \( \overline{S}_p \) is the mean piston speed (m s\(^{-1}\)). \( u_{swirl} \) is the swirl velocity (m s\(^{-1}\)). the subscript \( IV \) refers to conditions at \( IV \). \( p_f \) is the fired pressure, \( p_m \) is the motored pressure estimated via \( p^{1.3} = cst. \) and \( V_{disp} \) is the displaced volume. A modification to the Woschni correlation to account for swirling velocity flow has been incorporated by defining the characteristic speed to be the sum of the mean swirling velocity and the mean piston speed (Cheung & Heywood, 1993). Incorporation of the mean swirl velocity is appropriate because, physically, a higher degree of swirl is expected to
increase the heat transfer rate.

The parameters $C_1$ and $C_2$ in Equations F.2 and F.1 are engine specific and must be determined by calibrating the heat transfer model: both are $O(1)$. Physically, parameter $C_1$ scales the heat transfer coefficient, and $C_2$ scales the part of the characteristic velocity assumed to result from the expansion of gases on combustion (Gatowski et al., 1984). The technique used to calibrate the heat transfer model for a specific engine is to match the average final mass burn fraction, $x_{bf}$, with the combustion efficiency. Further details regarding the heat transfer model calibration are discussed by Ancimer (1995).

**F.2 Crevice effects**

There are several crevices in an operating engine, including: i) the space between the piston and the cylinder liner above the top piston ring; ii) the space around the spark plug; iii) the region between the cylinder block and head; and, iv) for the case where a pressure transducer is mounted, the region around the pressure transducer. Total crevice volume is typically $O(2\%)$ of the clearance volume. Although the crevice volume may be a small fraction of the total cylinder volume, because the crevice gases are cold relative to the in-cylinder gases (i.e., $T_w \approx 475$ K vs. $T_b \approx 2000$ K), a substantial fraction of the mass is contained in the crevices (Heywood, 1988; Thompson & Wallace, 1994).

The crevice volume effects are modelled in the one-zone system as a single aggregate volume at the in-cylinder pressure and the wall temperature. The total crevice volume mass is calculated using the ideal gas relation (Gatowski et al., 1984).

$$m_{crev} = \frac{\rho V_{crev} M_{avg}}{RT_w}$$  \hspace{1cm} (F.3)

where $V_{crev}$ must be estimated. The crevice model assumes that there are no flow restrictions. Mass flows into the crevice when the cylinder pressure is rising, and mass flows back into the cylinder when the pressure is decreasing. The crevice model does
not take into account any details of the mass flow, but is expected to account for the overall crevice effect.

F.3 Residual gas fraction calculation

The residual gas fraction has a strong influence on the combustion stability of spark ignition engines by acting as a diluent which extends the combustion duration (Fox, Cheung, & Heywood, 1993): A good estimate of the residual gas fraction is needed by the thermodynamic model to give an accurate mass burn analysis. A model for calculating the residual gas fraction for use in mass burn rate analysis has been developed by Fox et al. (Fox et al., 1993). The model relates the residual gas fraction to engine speed (N (rps)), inlet and exhaust pressures (p_i and p_e (bar)), an engine specific valve overlap factor (OF. (° m⁻¹)), compression ratio (r_e) and φ.

\[
x_{\text{res}} = 1.266 \frac{OF}{N} \left( \frac{p_i}{p_e} \right)^{-0.87} \sqrt{p_e - p_i} + 0.632 \frac{\phi}{r_e} \left( \frac{p_i}{p_e} \right)^{-0.74}
\]

The model is accurate at low to medium engine speeds where the cylinder pressure does not differ much from the exhaust port pressure when the intake valve opens (Fox et al., 1993). An empirical expression for estimating OF is given by

\[
OF = \frac{1.45}{B} (107 + 7.8 \Delta \theta + \Delta \theta^2) \frac{L_{v,\text{max}} D_v}{B^2}
\]

where \( \Delta \theta \) is the valve overlap in CAD (valve timing defined at 0.15 mm lift); \( B \) is the cylinder bore (mm), \( L_{v,\text{max}} \) is the maximum valve lift (mm) and \( D_v \) is the valve inner seat diameter (mm). The last two parameters are averages of the intake and exhaust valves. The OF for the single cylinder engines used in this study are listed in Table 3.1. Based on data presented by Fox et al. (Fox et al., 1993), the correlation predicts the residual gas mass fraction to within 5% for most cases. The model, however, tends to under-predict the residual gas mass fraction at low inlet pressures.
Appendix G

Fibre-optic spark plug data analysis

The objective of the FOSP data analysis is to extract parameters characteristic of the early flame kernel development on an individual cycle basis. There are three parameters of interest: i) the mean kernel expansion speed $S_g$, ii) the average kernel convection velocity $\bar{v}_c$ and iii) the flame kernel distortion, $F_d$. These parameters are calculated from the 2-D flame contour constructed using either the cubic spline model of Bianco et al. (1991) or the ellipse model of Kerstein and Witze (1990). The physical interpretation of each parameter is discussed below in terms of their respective definitions.

The flame arrival times, measured relative to spark timing, are determined by setting a voltage threshold: once the voltage from a fibre-optic probe exceeds this threshold value the flame is assumed to have arrived at that probe. The technique proposed by Geiser, Wytyrkus, and Spicher (1998) to set the threshold voltages for the eight fibres is used. One common threshold voltage is specified for all eight fibre-optic probes: this common threshold voltage is then adjusted for each fibre-optic probe by normalising the threshold voltage with respect to the average peak voltage of each fibre-optic probe. In this way, the threshold voltage is higher for a fibre-optic probe with a light responsivity greater than the average light responsivity of the eight fibre-optic probes. Using these threshold voltages, the eight flame arrival times are
obtained for each engine cycle. The cubic spline or ellipse model algorithms are then applied to the eight flame arrival times.

G.1 Cubic spline model

The cubic spline model constructs a 2-D flame contour of the 3-D flame from the flame arrival times: the contour is constructed at $FAT_m = 10$ CAD after the initial spark discharge event. The radial position of the flame contour at each probe at this time, $R(FAT_m, i)$, is calculated by assuming that the flame front velocity is constant in each direction $\nu(i)$.

\[
R(FAT_m, i) = \frac{FAT(i)}{FAT_m} R_s
\]

where $FAT(i)$ is the flame arrival time at fibre-optic probe $i$ and $R_s = 5.35$ mm is the radial position of the fibres relative to the centre electrode. The 2-D flame contour is constructed using cubic splines.

\[
R(\nu) = a(i)\nu^3 + b(i)\nu^2 + c(i)\nu + d(i)
\]

The constraints imposed on the cubic spline fit to calculate the 32 parameters are i) that the flame contour must pass through each point $\tilde{R}(i)$, and ii) that the first and second derivatives at each point must be continuous.

G.2 Characteristic parameters

The three characteristic parameters $\tilde{V}_c$, $S_y$, and $F_d$ are calculated from the 2-D flame contour on an individual cycle basis. The mean convection velocity describes the movement of the centre of mass of the flame kernel, and is calculated from

\[
\tilde{V}_c = \frac{\bar{OG}}{FAT_m} = (|V_c|, \nu_c)
\]
where \( \vec{OG} = (x_M, y_M) \) is the vector position of the centre of mass relative to the ignition point.

The mean expansion speed, \( S_g \), characterises the rate at which the flame kernel expands. The perimeter based \( S_g \) value is calculated as

\[
S_g = \frac{R_p}{FAT_m} \quad (G.4)
\]

\[
R_p = \frac{1}{2\pi} \int_0^{2\pi} \int_0^{FAT_m} \dot{R}(\psi) d\psi d\theta \quad (G.5)
\]

where \( R_p \) is the perimeter based characteristic length scale. The flame expansion speed is representative of the average rate of growth of the flame kernel over time and direction.

The centroid based \( S_g \) value, proposed by Lord et al. (1993), is calculated as

\[
S_c(\psi) = \frac{R_M(\psi)}{FAT_m} \quad (G.6)
\]

\[
S_g = \frac{1}{8} \sum_{i=1}^{8} S_c(\psi(i)) \quad (G.7)
\]

where \( R_M(\psi) \) is the distance from the flame contour centroid to the flame contour perimeter in direction \( \psi \), and \( S_c(\psi) \) is the flame front expansion rate in direction \( \psi \). The mean expansion rate is calculated over eight equi-spaced directions \( \psi(i) \). An alternate form to calculate the centroid based \( S_g \) is

\[
S_g = \frac{1}{2\pi} \int_0^{2\pi} S_c(\psi) d\psi \quad (G.8)
\]

The centroid based \( S_g \) definition as the average expansion rate relative to the flame contour centroid is distinctly different from the average expansion rate away from the ignition point. Equation G.7, although less computationally intensive than Equation G.8, is less desirable because the results may depend on the choice of the eight averaging directions used to calculate \( S_g \) if the flame contour is oddly shaped. Equation G.8 gives a consistent result because of integration over the entire flame contour.

The flame distortion parameter describes the distortion of the flame kernel relative
to a circle, and is calculated as

\[ F_d = \frac{R_a}{R_p} \quad (G.9) \]

\[ R_a^2 = \frac{1}{2\pi} \int_0^{2\pi} R^2(\theta, F AT_m) d\theta \quad (G.10) \]

where \( R_p \) is calculated from Equation G.5. \( R_a \) represents the characteristic length scale associated with the flame contour area, whereas \( R_p \) represents the length scale associated with the flame contour perimeter. A distorted or stretched flame kernel has a flame distortion parameter \( > 1 \).

The difference between the centroid and perimeter based \( S_g \) definitions in any direction \( \theta \) is given by

\[ R_M - R(\theta) = \sqrt{|OG|^2 - 2R_\theta (x_M \cos(\theta) + y_M \sin(\theta)) + R(\theta)^2} - R(\theta) \quad (G.11) \]

where \( R_M \) is the distance from the flame contour centroid to the perimeter and \( R(\theta) \) is the distance from the ignition point to the flame contour perimeter. If the entire perimeter of the flame contour is considered, then on average \( R_M > R(\theta) \) and the centroid based expansion rate is larger than the perimeter based expansion rate. This inequality exists because the centroid based \( S_g \) is implicitly weighted by the square of the distance to the flame contours, whereas the perimeter based \( S_g \) is weighted by the distance. The two are related through the flame distortion parameter.

### G.3 Ellipse model

The ellipse model, developed by Kerstein and Witze (1990), takes the streamwise development of the kernel to reflect the combined influence of combustion and convection, while the transverse development is deemed to reflect only the effect of combustion. These definitions are based on the physical requirement that increasing flame kernel area should be attributed to combustion and expansion, whereas convection is a flame kernel area preserving mechanism. Based on this reasoning, the ellipse model
assumes i) the flame kernel to be elliptical with constant eccentricity. ii) the major and minor axes increase linearly with time, and iii) the center of the ellipse moves away from the center electrode at a constant speed $S_c$. The growth rate of the minor axis is taken to be the growth rate due to combustion, $S_2$: the convection direction is taken to be the direction of the major axis; and the convection velocity is calculated as $V_c = S_z + S_c - S_2$, where $S_z$ is the growth rate of the major axis. A non-linear least squares parameter estimation algorithm is used to solve for the ellipse model parameters. It is assumed that $S_z > S_c$ so that finite arrival times are obtained for all eight fibres. The so-called unconstrained ellipse model formulation is used instead of the simplified circular or constrained formulations.
Appendix H

Energy based averaging

Cyclic variations in non-stationary IC engine flows are associated with the large, coherent, energy containing scales, and are not turbulence (Arcoumanis & Whitelaw, 1987; Rask, 1981). This is in contrast to stationary flows where coherent events are associated with the turbulence. An Energy based averaging technique is introduced which identifies the main energy contributors in engine flows as the mean: the remainder is turbulence. The criteria is the removal of high energy events from the turbulence. The filtering is done on an individual cycle basis using a discrete wavelet transform based energy filter.

The Energy based averaging technique uses the ensemble average velocity as the first estimate of the individual cycle mean. The first estimate of the turbulence velocity in cycle k is the difference between the instantaneous and ensemble velocities.

\[
\bar{U}_{IC1}(\theta) = \frac{1}{N} \sum_{k=1}^{N} \tilde{u}(\theta, k) \quad (H.1)
\]

\[
u_{EST1}(\theta, k) = \tilde{u}(\theta, k) - \bar{U}(\theta) \quad (H.2)
\]

The discrete-time wavelet transform, DWT, of \(u_{EST1}(\theta, k)\) is calculated using a wavelet basis function: the specific wavelet functions used in the analysis are discussed in subsection H.2. The wavelet coefficients of \(u_{EST1}(\theta, k)\) are then low pass filtered on an energy level basis. A cut-off energy level is chosen for each cycle, where the cut-off energy is chosen as a fraction of the highest energy wavelet coefficient in cycle k.
Any coefficients with energy above this cut-off level are assigned to the mean. The remaining coefficients are turbulence. This separation of \( u_{EST1}(\theta, k) \) into high and low energy components is given by Equation H.3

\[
u_{EST1}(\theta, k) = u_{HE}(\theta, k) + u_{LE}(\theta, k) \tag{H.3}
\]

A systematic method, described in the subsection H.1, is used to choose the cut-off energy level. The inverse-DWT of \( u_{HE}(\theta, k) \) is added onto the Ensemble based average velocity giving the final estimate of the individual cycle mean.

\[
\overline{U}_{IC2}(\theta, k) = \overline{U}_{IC1}(\theta) + u_{HE}(\theta, k) \tag{H.4}
\]

which leads to the definition of the fluctuating velocity as

\[
u(\theta, k) = \hat{u}(\theta, k) - \overline{U}_{IC2}(\theta, k) \tag{H.5}
\]

Statistical properties of the turbulence are calculated from \( u(\theta, k) \), for example, using Equation E.3 or E.4. Ensemble averaging of the individual cycles converges to the Ensemble based average by virtue of the technique employed. This satisfies the criteria specified for individual cycle mean estimate techniques.

This filtering scheme is designed to capture high energy local events. Wavelet basis functions are ideally suited to capture these events (Press et al., 1980; Daubechies, 1992): Fourier. or sinusoidal. basis functions are not. Wavelets are localised in both the crank angle and frequency domain. Sinusoids are localised in the frequency domain, but of infinite duration in the crank angle domain (Daubechies, 1992). The wavelet basis functions proved to be superior in selectively filtering out the energetic coherent events, and insensitive to the choice of low order wavelet basis function. Comparison of the results using different wavelet basis functions are discussed in subsection H.2.

This procedure is similar to that proposed by Lancaster (Lancaster et al., 1976) where the turbulence velocity calculated by the Ensemble based average was inter-
interpreted to consist of low and high frequency components associated with the cyclic variations in the mean and turbulence, respectively. This procedure is also similar to the triple decomposition (Hussain & Reynolds, 1970).

The LDV velocity data has been obtained from an optical L-head engine (Sullivan et al., 1999). The engine and LDV system characteristic parameters are listed in Tables 3.1 and 3.2 respectively, and a schematic of the optically accessible engine and the coordinate system definition is shown in figure 3.1. The u and v component velocity data were acquired in random mode and filtered using a 1 crank degree coincidence mode bin (Ancimer, 1997). Thus, aliasing of high frequency energy into lower frequencies does not occur.

H.1 Energy cut-off level

A cut-off energy level must be selected and is chosen to filter out any significant coherence without removing energy across a broad band of frequencies. A systematic method, based on $\chi^2$ values, is used to choose the energy cut-off level for the u and v velocity components. The results are independent of wavelet basis function choice: Coiflet-12 wavelet basis function filtering results are used to demonstrate. Figure H.1 contains a plot of the $\chi^2$ values resulting from filtering at different energy levels for the u and v component velocities. The $\chi^2$ values have a maxima which tapers off to 0 at filtering levels of 0 and 100%. At a 100% cut-off level the highest energy component is assigned to the mean. In this case the individual cycle mean does not vary from the Ensemble based mean velocity. At a 0% cut-off level all the energy is high energy and the individual cycle mean is the instantaneous velocity. The ensemble of the instantaneous velocity is also the Ensemble based mean velocity by definition. The maxima in $\chi^2$ is chosen as the optimal filtering level.

To demonstrate the validity of this choice, the normalised $\langle uv \rangle$ correlation and coherence function plots resulting from low, high and optimal cut-off energy levels are compared. The high filtering level was chosen as $(u,v) = (25%,25%)$ and the low filtering level as $(u,v) = (2.5%,2.5%)$. These levels were chosen as filter levels far
from the optimal \( (u, v) = (14\%, 11\%) \) to demonstrate the effect of varying the cut-off level. The normalised \( \langle uv \rangle \) correlation plots are shown in Figures H.2(b), H.2(c) and H.2(a), and the coherence functions are plotted in Figures H.3(b), H.3(c) and H.3(a).

Comparison of high to optimal threshold level \( \langle uv \rangle \) correlation plots. Figures H.2(c) and H.2(a) respectively, indicates that the general trends with respect to crank angle degree are the same, but the level of correlation for the optimal filtering case is significantly lower. The most noteworthy differences are i) the absence of a distinct trough at \( \theta = 40^\circ \), and ii) the near zero average correlation throughout \( \theta = 30-150^\circ \) for the optimal threshold case. Comparison of high to optimal threshold level coherence plots. Figures H.2(c) and H.2(a) respectively, indicates that: i) in the 0-80 (engine cycle)\(^{-1} \) range the coherence energy is 2-5 times lower in the optimal vs. high threshold case. ii) the optimal filter has selectively filtered out frequencies associated with 90 (engine cycle)\(^{-1} \) and 120 (engine cycle)\(^{-1} \); and iii) for frequencies > 200 (engine cycle)\(^{-1} \) the coherence functions are indistinguishable. It is concluded that the cut-off energy levels of \( (u, v) = (25\%, 25\%) \) is too high a threshold and not enough coherent energy is removed from the turbulence.

Figures H.2(b) and H.3(b) show that for \( (u,v) \) thresholds of (2.5\%, 2.5\%) there is little correlation or coherence left as turbulence. The coherence function. Figure H.3(b). indicates energy has been removed from all frequencies and the correlation function. Figure H.2(b). indicates that energy has been removed across all \( \theta \). This threshold is too low and too much energy is removed from the turbulence.

Figures H.2(a) and H.3(a) show that for \( (u,v) \) thresholds of (14\%, 11\%) optimal filtering has been achieved. The normalised \( \langle uv \rangle \) correlation plot. Figure H.2(a). shows that most of the correlation in the intake stroke has been removed from the turbulence. The remainder of the correlation is associated with the turbulence. The coherence function. Figure H.3(a). is flat, but energy has not been removed from all frequencies. The energy filter has selective filtered out scales at 90 and 120 (engine cycle)\(^{-1} \). This is similar to the results obtained through the use of continuous wavelet transforms, where the energetic scales associated with the mean were found to range from 80-120 indicating that the results are comparable. The maxima in the \( \chi^2 \) plots
is therefore interpreted as a transition point where, at cut-off levels below this value, energy throughout the cycle and at all frequencies is assigned to the mean and at cut-off levels above these values, not enough coherent energy is removed from the turbulence.

## H.2 Wavelet basis function

The Energy based filtering results from using the Daubechies-4 and -20. and Coiflet-12 wavelet basis functions (Press et al., 1980; Daubechies, 1992) are compared. These three wavelets correspond to second, tenth and sixth order wavelet basis functions respectively. The comparison of the results is used to determine: i) if the order, or scale, and ii) if the shape, of the wavelet influences the results. The comparison is made by first determining the optimal filtering level for each wavelet basis function by the method described in subsection H.1: the optimal filtering levels for the Daubechies-4 and -20. and Coiflet-12 wavelet are $(u, v) = (15\%, 12.5\%), (18\%, 20\%)$, and $(14\%, 11\%)$ respectively. The normalised \(\langle uv\rangle\) correlation, coherence functions and turbulence RMS velocities, \(u'\) and \(v'\), calculated via Equation E.3 using the three different wavelet basis functions are compared.

The normalised \(\langle uv\rangle\) correlation, Figures H.2(d), H.2(e) and H.2(a), and coherence function plots, Figures H.3(d), H.3(e) and H.3(a), show insignificant differences in trend or magnitude with respect to crank angle and frequency respectively. The only notable difference is that the Coiflet-12 wavelet appears to selectively filter out coherent energy in the range of 90 and 120 (engine cycle)$^{-1}$. Figure H.3(a).

The turbulence RMS velocities calculated using the three wavelet basis functions are plotted in Figures H.4(a)–H.4(c). Comparison of the low order wavelet function results, that is the Daubechies-4 and Coiflet-12 in Figures H.4(b) and H.4(a) respectively, indicates that differences in the trend and magnitude of the \(u'\) and \(v'\) are insignificant. Both plots show an isotropic turbulence in the \(x\) and \(y\) component directions, where an initially low turbulence velocity, at 0–40°, increases to a relatively constant level once the influence of the intake jet is felt at the measurement location.
Figure H.1: $\chi^2$ values as a function of energy filter level, where the Coiflet-12 wavelet basis function is used. Motored operation at $\dot{\varphi} = (0. -25, 15)$. 900 RPM and $N = 200$ cycles.

0–180°. Comparison of the high order wavelet basis function results. Daubechies-20 in Figure H.4(c) to the low order wavelet basis functions. Daubechies-4 and Coiflet-12 in Figures H.4(b) and H.4(b) respectively, indicates that the trend in turbulent RMS velocity with respect to crank angle is similar, but that $v' > u'$ for the Daubechies-20 wavelet. This difference may be due to the high order of the Daubechies-20 wavelet which may introduce an additional, undesirable filtering scale. It is concluded that a low order wavelet basis function should be used, and that there are no significant differences between the type of low-order wavelet function: the Coiflet-12 wavelet basis function is chosen because it highlights flow accelerations which are associated with coherent events (Holmes et al., 1996).
(a) Energy based optimal filtering. \((u,v)\) filtered at \((14.11)\%\) using a Coiflet-12 wavelet.

(b) Energy based filtering. \((u,v)\) filtered at \((2.5, 2.5)\%\) using a Coiflet-12 wavelet basis function.

(c) Energy based filtering. \((u,v)\) filtered at \((25.25)\%\) using a Coiflet-12 wavelet basis function.

(d) Energy based optimal filtering. \((u,v)\) filtered at \((15.12.5)\%\) using a Daubechies-4 wavelet basis function.

(e) Energy based optimal filtering. \((u,v)\) filtered at \((18.20)\%\) using a Daubechies-20 wavelet basis function.

Figure H.2: Normalised \(\langle uv \rangle\) correlation of the turbulence as calculated by different flow decomposition techniques. A value of 0.15 is statistically significant at a 95% confidence level. Motored operation at \(\bar{\omega} = (0. - 25.15)\). 900 RPM and \(N = 200\) cycles.
(a) Energy based optimal filtering. (u,v) filtered at (14.11)% using a Coiflet-12 wavelet.

(b) Energy based filtering. (u,v) filtered at (2.5.2.5)% using a Coiflet-12 wavelet basis function.

(c) Energy based filtering. (u,v) filtered at (25.25)% using a Coiflet-12 wavelet basis function.

(d) Energy based optimal filtering. (u,v) filtered at (18.20)% using a Daubechies-4 wavelet basis function.

(e) Energy based optimal filtering. (u,v) filtered at (18.20)% using a Daubechies-20 wavelet basis function.

Figure H.3: \( \langle uv \rangle \) coherence of the turbulence as calculated by different flow decomposition techniques. Motored operation at \( \tilde{F} = (0. -25.15) \). 900 RPM and \( N = 200 \) cycles.
(a) Energy based optimal filtering, (u,v) filtered at (14.11)% using a Coiflet-12 wavelet basis function.

(b) Energy based optimal filtering, (u,v) filtered at (15.12.5)% using a Daubechies-4 wavelet basis function.

(c) Energy based optimal filtering, (u,v) filtered at (18.20)% using a Daubechies-20 wavelet basis function.

Figure H.4: Turbulence velocity after filtering. Motored operation at \( \bar{\omega} = (0. -25. 15) \) RPM and \( N = 200 \) cycles.
Appendix I

Velocity data aliasing and even time sampling

A 2-component LDV system was used to measure the velocity data. An inherent problem with LDV measurements is that the data arrives randomly. That is, a valid velocity data point is obtained only if a seed particle is in the probe volume and the SNR of laser light scattered off the particle is high enough for the LDV hardware to make a measurement. A great deal of time was spent setting up the LDV and particle seeding system to maximise the data rates, such that any further increase in data rates cannot be realised with the current system: the average data rates were between 10-30 kHz per velocity component and the time resolution of the shaft encoder is 0.2 CAD. One question which arises is whether the LDV data rates are high enough to prevent aliasing of high frequency energy into low frequencies; a second related issue is, since we need to take the DWT and FFT of the data, how to obtain even time sampled velocity data from the LDV data while introducing minimal amounts of bias to the spectra: the compromise solution arrived at is outlined in this appendix.

I.1 Aliasing issues

To ensure that aliasing of high frequency energy into the low frequency components does not occur, either the velocity data must be sampled at a rate greater than
twice the maximum frequency at which there is significant energy in the flow. or
the data must be low pass filter prior to sampling the data. The work of Corcione
and Valentino (1994) shows that the bandwidth of the flow in IC engines is linearly
dependent on engine speed: a bandwidth of 20 kHz. with a maximum frequency
of 10 kHz at 1500 RPM is observed near TDC of the compression stroke in their
direct injection compression ignition engine. Similar bandwidths are observed in
other engines: see, for example, the work of Catania and Spessa (1996). To prevent
aliasing of the data, the minimum sampling rate \( f_s \) must be at least the bandwidth.
\[ f_s = \frac{\Delta f_{bw}}{N} = 1600 \text{ samples (engine cycle)}^{-1} \text{ or a 0.45 CAD sampling interval.} \]
The implication is that the 0.2 CAD encoder resolution used here is sufficiently fine to
prevent aliasing, assuming valid data exists at each point. Since valid data does not
exist at each point, the issue to address is how to minimise any biases while obtaining
even time sampled velocity data. There are a variety of options available, including
binning the data, interpolating the data, and down sampling the data.

\section*{I.2 Interpolation and down sampling}

The most straightforward technique which could be applied is interpolation and down
sampling. Fansler and French (1988) employed this technique through hardware,
where they simulate a continuous velocity signal by sample and holding the LDV data,
and then even time sample this simulated continuous velocity data at crank degree
intervals of \( \Delta \theta_b = (\text{Encoder resolution}) \times M \). This technique simply down samples
the data by a factor \( M \). Any additional data acquired within the crank window is
effectively thrown away: the binning technique, described next, is superior to this
technique in preventing aliasing to this method because it low pass filters the data
prior to down sampling.
I.3 Binning

Rask (1981) Liou and Santavicca (1985) Lorenz and Prescher (1990) and Foster and Witze (1988), obtained even time sampled velocity data by binning the LDV uneven time sampled data, where the bin width is defined by \( \Delta \theta_b = (\text{Encoder resolution}) \times M \). If \( M \neq 1 \), the effect of this technique is a two step filtering process: the data is first low pass filtered using a rectangular window function of width \( M \) and then down sampled by a factor \( M \). Ideally, binning should prevent aliasing from occuring and, at the same time, remove insignificant amounts of energy from the PSD below the cut-off frequency. The coarseness of \( \Delta \theta_b \) needs to be chosen wide enough to ensure there is at least one data point in each bin. The DWT and FFT of the individual cycle velocity data can then be evaluated. The question is how the choice of bin size biased the resulting PSD functions.

The effect of the low pass rectangular window function is a convolution in the time domain or a multiplication in the frequency domain: the influence of this process on the true PSD function \( G_{ii}(f) \) is through multiplication by a magnitude-squared Dirichlet kernel prior to sampling.

\[
\hat{G}_{ii,lp}(f) = |H(f)|^2 G_{ii}(f) \\
H(m) = \frac{1}{K} \frac{\sin(\pi m M/K)}{\sin(\pi m/K)} \\
f = \frac{mf_s}{K}
\]

where \( K \) is the length of the discrete data vector, \( M \) is the length of the low pass rectangular window, \( m \) is the index in the frequency domain, and \( f_s \) is the sampling frequency, assumed to be at least as large as the bandwidth of the flow \( f_{bw} \). The magnitude-squared Dirichlet kernel is plotted in Figure I.1(a) for \( M=5 \) and \( M=2 \), which are associated with \( \Delta \theta_b \) of 1 CAD and 0.40 CAD, respectively.

The effect of down sampling at \( f_i \) is to fold the energy above \( f_i/2 \) back into the
lower frequencies.

\[
\hat{G}_{ii}(f) = G_{ii,p}(f) + G_{ii,lp}(2f_l - f)
\]  

(1.4)

where this equation is valid as long as \( f_l > f_{bw}/2 \); if \( f_l < f_{bw}/2 \) then additional energy is added from the discrete spectra centred at \( 4f_l \). For \( M = 5 \) and \( M = 2 \), the folding frequencies are 360 and 900 (engine cycle)\(^{-1} \), respectively. The net result is a biased PSD function \( \hat{G}_{ii} \): the degree of biasing depends on the value of \( M \) and the form of \( G_{ii} \). The value of \( M \) is chosen such that at least one data point is obtained in each bin, and \( G_{ii} \) is fixed by the flow system under investigation.

To estimate the degree of biasing if binning is applied to the velocity data taken from the SCV6 3.1 L engine data, the LDV data at 900 RPM and 70 kPa MAP, in the window range from 335-385 CAD. is considered. An approximation to \( G_{ii} \) is obtained by interpolation and over sampling, such that data is obtained at 0.2 CAD intervals. The interpolation and over sampling technique is described in section I.4: the maximum allowable drop out range of data was 9 bins, or 2 CAD. Only the DC component of the velocity was removed before evaluating the PSD function. The resulting PSD is biased by the interpolation and over sampling technique, but should give a reasonably good approximation to the PSD \( G_{ii} \). The u and v-component PSD functions are plotted in Figure I.1(b): since both are similar only the u-component velocity PSD is used to evaluate the biases introduced by binning.

Bin widths of 1.0 and 0.40 CAD. associated with \( M = 5 \) and \( M = 2 \), respectively, are considered. The value of \( M = 5 \) is needed to obtain at least one data point in most of the bins throughout the 335-385 CAD window. The magnitude squared Dirichlet Kernel for \( M = 5 \) and \( M = 2 \) was applied to the u-component PSD: the results are plotted in Figure I.1(c): the results after down sampling are shown in Figure I.1(d). Fortuitously, the rapid decay seen at the higher frequencies for the PSD’s associated with the Dirichlet kernel filter is compensated for by the aliased energy. The aliased energy, however, is considered to be noise. To quantity the effects the offsetting biases are divided into two parts: how much energy is filtered out by the low pass filter and
how much high frequency energy is aliased into lower frequencies.

The effect of the low pass filter is examined over three frequency ranges: $f < 360$ and $f < 900$ (engine cycle)$^{-1}$, and the total PSD energy. The results, listed in Table I.1. as expected, indicate that the total PSD energy decreases significantly as the $\Delta \theta_b$ is increased from 0.2–1.0 CAD. Recalling that $M = 5$ is needed to obtain at least one data point in most of the bins, we see that the energy has decreased by almost 50% relative to the entire spectrum, and that the best case scenario is a underestimate by 14% in the range $f < 360$ (engine cycle)$^{-1}$. In comparison, the energy throughout the spectrum for a $\Delta \theta_b$ of 2 CAD results in a decrease in 25%, and a 2% decrease for the range $f < 360$ EC.

The total aliased energy is quantified as the energy of $G_{au,dp}$. Equation I.1. above the down sampling frequency $f_t$. The results, listed in Table I.1. indicate that significant aliasing occurs for both $M = 5$ and $M = 2$, where approximately the same amount of energy is aliased for both cases. The SNR for $M = 5$ would, however, be worse than that for $M = 2$ due to less total energy throughout the spectrum: for $M = 2$ most of the aliasing occurs at the higher frequency components where the low pass filter magnitude is greatest. The results show that the amount of aliased energy over the range $f < 360$ (engine cycle)$^{-1}$ for $M = 2$ is $< 0.50\%$, whereas all the frequencies are contaminated by aliased at $M = 5$.

The conclusion would have to be that the process of binning at $M = 5$ is not acceptable. Ideally, we would want to use $M = 1$ to get the best unbiased estimate of the total flow energy, or use $M = 2$ to get a good estimate of the spectra over the range of frequencies $f < 360$ (engine cycle)$^{-1}$. To do this, however, since there is not a valid data point in each bin, the data must be interpolated and over sampled. This introduces biases of its own: for example, the total PSD energy listed in Table I.1 shows that it underestimates the actual energy in the 335–385 CAD by 12%. The compromise solution between binning and using interpolation and over sampling is needed.
Table 1.1: Bias estimates of binning technique. u-component velocity data from SCV6 3.1 L engine at 900 RPM and 70 kPa. Actual energy is obtained by ensemble averaging the velocity data and is not biased by the interpolation and over sampling used to estimate the unaliased PSD $G_{uu}$ at $M=1$.

<table>
<thead>
<tr>
<th>$M$</th>
<th>$\Delta \theta_b$ (CAD)</th>
<th>Actual</th>
<th>PSD</th>
<th>$f &lt; 360$</th>
<th>$f &lt; 900$</th>
<th>Total</th>
<th>Aliased</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>1.0</td>
<td>–</td>
<td>1.15</td>
<td>1.06</td>
<td>–</td>
<td>0.10</td>
<td>0.10</td>
</tr>
<tr>
<td>2</td>
<td>0.4</td>
<td>–</td>
<td>1.69</td>
<td>1.18</td>
<td>1.60</td>
<td>0.09</td>
<td>0.0046</td>
</tr>
<tr>
<td>1</td>
<td>0.2</td>
<td>2.48</td>
<td>2.18</td>
<td>1.20</td>
<td>1.75</td>
<td>–</td>
<td>–</td>
</tr>
</tbody>
</table>

(a) Binning low pass filter – the magnitude squared Dirichlet kernel
(b) $G_{uu}$ and $G_{rr}$ from SCV6 3.1 L engine at 900 RPM and 70 kPa MAP

(c) $G_{uu,LP}$ – the u-component PSD filtered by the magnitude squared Dirichlet kernel
(d) $\hat{G}_{uu}$ – the u-component PSD filtered by the magnitude squared Dirichlet kernel and down sampled

Figure I.1: Effect of the binning the data on the PSD functions.
I.4 Interpolation and over sampling

Since the results from binning the data are unsatisfactory, a third option is used where a continuous signal is simulated by linear interpolation of the velocity data between the valid data points: this continuous signal is then over sampled at even crank degrees intervals: a maximum time lapse between valid data points of $\Delta \theta_b = (\text{Encoder resolution}) \times M$ is allowed. A variation on this technique was used by Wiktorsson et al. (1996), where they used cubic splines to interpolate between the data points. If the time resolution is fine relative to the mean flow accelerations, and the time lapse between data points is small, then linear interpolation should give a reasonably good approximation to the velocity where valid data does not exist.

An allowable data drop out ADO is defined to limit the time lapse allowed in an individual cycle: if there are more than ADO bins in a row without a valid velocity data point then the individual cycle is discarded. The minimum distance between data points which causes rejection of the cycle is the effective drop out width $\text{EDW} = (\text{ADO} + 2) \Delta \theta_b$. The ADO is interpreted as an effective sampling interval $\text{ESI} = (\text{ADO} + 1) \Delta \theta_b$.

Any scales finer than the ESI cannot be resolved everywhere.

The spectral bias introduced by interpolation and over sampling increases with increasing ADO level: the spectral bias should manifest as contamination of the PSD with noise throughout the spectrum and can be quantified as the difference between the energy calculated without the inclusion of interpolated data and the energy calculated including the interpolated data. The spectral bias can be calculated based on the ensemble PSD or on an individual cycle basis.

The spectral biases, calculated from the ensemble PSD, introduced to the u component velocity data from the SCV6 3.1 L engine at 900 RPM and 70 kPa, are listed in Table I.2. There is a trade off between the spectral bias level and the number of valid engine cycles from which the statistics are calculated. For example, if ADO is low, then the spectral bias should be low, but there will be fewer engine cycles and the statistical uncertainty in the data will be large; if, however, a large ADO level is used, the statistical uncertainty can be reduced significantly, but then larger spec-
tral ADO biases are introduced. The results indicate that if a straight interpolation and over sampling of the data is done, there is no trade off between the spectral biases and uncertainty which reduces the overall uncertainty to below 10%. This is an unacceptably high uncertainty, leading to 95% confidence intervals of $> \pm 20\%$.

The question is whether this interpolation and over sampling bias extends throughout the spectra, or whether it remains mainly concentrated in scales finer than the ESI. This calculation cannot be made directly because the actual energy cannot be broken down with respect to frequency without introducing the biases. The indirect approach is to low pass filter the velocity data, and then calculate the PSD energy and the average ensemble turbulence energy: ensemble averaging in the time domain excludes the interpolated velocity data from the calculations. There will be some contamination of the 'unbiased' energy due to ringing effects, but it should be very small as the ringing phenomena cannot be seen the data. The results for a series of low pass filter are listed in Table I.3, where the energy is also broken down into bands between the low pass filters. The results indicate that the majority of the PSD contamination is in the frequency range $>360$ (engine cycle)$^{-1}$. If we, therefore, restrict ourselves to examining the spectral data for frequencies $< 360$ (engine cycle)$^{-1}$, then the spectral biases will be significantly smaller. The spectral biases in this case are listed in Table I.4 are still fairly high. We wish to reduce the spectral bias to at most the statistical uncertainty, with the total uncertainty $<< 10\%$.

A compromise approach is used to reduce the spectral biasing even further: if the data is first binned at a $\Delta \theta_b$ of 0.4 CAD, and then the interpolation and over sampling technique is applied, the uncertainties decrease to more acceptable levels; the results are listed in Table I.2 and show that filtering the raw data in this way reduces the overall uncertainty to $\approx 6\%$, or a 95% confidence interval of $\pm 12\%$ if ADO levels of 3 or 4 are used.

These ADO levels imply that scales finer than 2 CAD cannot be resolved on average everywhere. Recall from section I.3 that the spectra in the range $f < 360$ (engine cycle)$^{-1}$ is a good representation of the actual spectra, with very little biasing introduced by the binning technique. Only frequencies in this range, or scales of size
<table>
<thead>
<tr>
<th>$M$</th>
<th>$\Delta \theta_b$ (CAD)</th>
<th>$\Delta \theta_i$ (CAD)</th>
<th>ADO</th>
<th>$N$</th>
<th>Energy (m² s⁻²)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.20</td>
<td>2.0</td>
<td>9</td>
<td>1119</td>
<td>2.48</td>
<td>2.18</td>
</tr>
<tr>
<td></td>
<td>1.6</td>
<td>7</td>
<td>815</td>
<td></td>
<td>2.48</td>
<td>2.18</td>
</tr>
<tr>
<td></td>
<td>1.2</td>
<td>5</td>
<td>372</td>
<td></td>
<td>2.53</td>
<td>2.24</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>3</td>
<td>28</td>
<td></td>
<td>2.78</td>
<td>2.56</td>
</tr>
<tr>
<td>2</td>
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<td>2.0</td>
<td>4</td>
<td>1212</td>
<td>2.18</td>
<td>2.07</td>
</tr>
<tr>
<td></td>
<td>1.8</td>
<td>3</td>
<td>929</td>
<td></td>
<td>2.19</td>
<td>2.10</td>
</tr>
<tr>
<td></td>
<td>1.2</td>
<td>2</td>
<td>505</td>
<td></td>
<td>2.18</td>
<td>2.10</td>
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<td>0.8</td>
<td>1</td>
<td>69</td>
<td></td>
<td>2.31</td>
<td>2.27</td>
</tr>
</tbody>
</table>

Table I.2: Ensemble spectral energy bias estimates of the interpolation and over sampling technique. u component velocity data from scv6 3.1 L engine at 900 RPM and 70 kPa. The reported error is one standard deviation.

2 CAD and larger, are considered to be accurately represented by the data.

If we now estimate the spectral errors introduced by the interpolation and over sampling for $f < 360$ (engine cycle)⁻¹, the results are listed in Table I.4, and indicate that the spectra are biased overall by $< 5\%$, where the spectral bias has essentially disappeared: only statistical uncertainty remains. This is the best we can hope to do with the data.

The same analysis procedure was repeated for all 9 operating points and the estimates of the spectral error are reported in Table I.5. Similar results to those of operating point 7A1 are obtained for all the operating points. The spectral bias errors are all below the statistical uncertainty.

The error introduced on an individual cycle basis was also evaluated. The average energy in the 335–385 CAD window was calculated for each cycle for the data with and without interpolation; the first moments of the data were also calculated. Note that outliers beyond ±3.0σ were removed so as to not bias the results. The results, listed in Table I.6, indicate that on average the spectral bias is of the same magnitude as the error introduced on the ensemble average throughout the spectrum: for comparison, see Table I.2. It is also expected that the error on an individual cycle basis for $f < 360$ (engine cycle)⁻¹ to be much smaller.

The large standard deviations in the magnitude of the individual cycle energy
Table I.3: Ensemble spectral energy bias estimates broken down with respect to frequency. $f_{\text{band}}$ is the bias energy in the frequency range above the cut-off frequency, but below the next highest cut-off frequency. u component velocity data from SCV6 3.1 L engine at 900 RPM and 70 kPa.

<table>
<thead>
<tr>
<th>$f_{co}$ (engine cycle)$^{-1}$</th>
<th>Energy (m$^2$ s$^{-2}$)</th>
<th>$&lt; f_{co}$</th>
<th>$f_{band}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Actual</td>
<td>Biased</td>
<td>Actual</td>
</tr>
<tr>
<td>1440</td>
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<td>0.15</td>
<td>0.19</td>
</tr>
<tr>
<td>720</td>
<td>0.82</td>
<td>0.56</td>
<td>0.63</td>
</tr>
<tr>
<td>360</td>
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<td>0.53</td>
</tr>
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<td>0.18</td>
</tr>
<tr>
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<td>2.00</td>
<td>0.20</td>
</tr>
<tr>
<td>14.4</td>
<td>2.48</td>
<td>2.18</td>
<td>0.18</td>
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Table I.4: Ensemble spectral energy bias estimates of the interpolation and over sampling technique for $f < 360$ (engine cycle)$^{-1}$. u component velocity data from SCV6 3.1 L engine at 900 RPM and 70 kPa. The reported error is one standard deviation.

<table>
<thead>
<tr>
<th>$M$</th>
<th>$\Delta \theta_b$ (CAD)</th>
<th>ESI (CAD)</th>
<th>ADO</th>
<th>N</th>
<th>Energy (m$^2$ s$^{-2}$)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
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<td></td>
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<td>Statistical</td>
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<td>4.2</td>
</tr>
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<td></td>
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<td>7</td>
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<td>815</td>
<td>1.13</td>
<td>5.0</td>
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<td></td>
<td>1.2</td>
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<td>1</td>
<td>69</td>
<td>1.58</td>
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</table>
Table I.5: Ensemble spectral energy bias estimates for $f < 360$ (engine cycle)$^{-1}$. u component velocity data from SCV6 3.1 L engine: ADO of 4 and $\Delta \theta_b$ of 0.4 CAD.

indicates that the energy varies significantly from one cycle to the next: the correlation coefficient between the unbiased and biased energy are $> 0.99$ for all the operating points. This indicates that even though there is significant bias introduced into the magnitude of of the energy on an individual cycle basis, the variations in the magnitude from one cycle to the next are well represented by the biased results. This implies that correlating the energy on an individual cycle basis with mass burn parameters will be equivalent to the unbiased estimates of the energy when using the data binned at 0.4 CAD with $\text{ADO} = 4$.  

<table>
<thead>
<tr>
<th>Operating point</th>
<th>$N$</th>
<th>Energy (m$^2$ s$^{-2}$)</th>
<th>Error standard deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Actual</td>
<td>Biased</td>
</tr>
<tr>
<td>7A1</td>
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<td>1.33</td>
<td>1.33</td>
</tr>
<tr>
<td>7B1</td>
<td>1142</td>
<td>1.52</td>
<td>1.52</td>
</tr>
<tr>
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<td>523</td>
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<td>2.17</td>
</tr>
<tr>
<td>8A1</td>
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<td>1.26</td>
<td>1.27</td>
</tr>
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<td>1.48</td>
</tr>
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</tr>
<tr>
<td>9C1</td>
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</table>
Table I.6: Individual cycle spectral energy bias estimates. u component velocity data from scv6 3.1 L engine: ADO of 4 and $\Delta\theta_b$ of 0.4 CAD. $C_{ij}$ is the normalised correlation coefficient.
Appendix J

Single cylinder engine operation

J.1 SCV6 3.1 L engine operating difficulties

The following list of problems were encountered while getting the optical engine operational.

Oil leaks Upon assembly of the optical engine, oil was found to leak from the crankcase into the Bowditch piston extension block. This is a problem because the engine is designed to operate oil free to minimise fouling of the quartz window. The measures taken to solve this problem were three-fold: i) the oil control and compression rings were installed on the stock pistons; ii) the meat of the stock piston crowns were milled out to reduce the pressure pulsations in the crank case; and iii) a vacuum was applied to the crankcase so that the pressure fluctuations in the crankcase never rise above atmospheric.

Variable spark timing A highly variable spark timing, with a typical 2 crank degree standard deviation, was observed under fired operation of the optical engine. A loose component in the distributor was isolated as the source of the problem. The measures taken to solve the problem were: i) all parts of the distributor and magnetic pick-up were tightened up; and ii) the stock oil pump was installed. The installation of the oil pump solved the problem because the distributor sits on top of the oil
pump. The oil pump applies a steady pressure on the distributor thereby preventing it from jumping about.

**Loose piston crown machine screws** After running the optical engine for relatively long periods of time without disassembly, it was found that the machine screws holding down the piston crown loosened. On two occasions the screws fell out into the combustion chamber and severely damaged the piston crown, and caused minor damage to the cylinder head walls. The screws loosened because only 20 in lbs of torque could be applied by the allen wrench. The piston crown was redesigned to allow for larger machine screws. The screws are now tightened down in 10 in lb increments up to 40 in lbs.

**Engine instability** An instability problem during fired engine operation was encountered. The average mass air flow into the engine never settled down to a constant value. The problem was most notable at high speed and load conditions. The problem was thought to be caused by expansion of the Bowditch piston due to heating. Temperature measurements were made along the Bowditch piston extension and the change in compression ratio was calculated based on the amount of expansion. At 1200 RPM and excess air ratio of 1, the compression ratio was found to increase from 7.4 under cold conditions to 7.9 after running at WOT for 10 minutes.

**Piston crown overheating** The piston crown overheated and was damaged after running under high load conditions for a more than a few minutes. These problems occurred for MAP> 85 kPa at 1500 RPM and MAP> 70 kPa at 1800 RPM. A high temperature high thermal conductivity paste is now used between the piston crown and piston to try to alleviate the overheating problem.

To reduce the heat damage to the piston crown, the engine was operated under a 'burst fired' option: under burst fired mode the engine was never fired for more than a few minutes at a time. The engine instability problem, however, then reared its' ugly head. That is, because the engine was never allowed to fully warm up under burst fired mode, the compression ratio changed over time and the engine operation was
unstable. The compromise solution was to make measurements at lower speeds and load conditions such that burst fired mode was not needed to prevent overheating of the piston crown.

J.2 Alignment and cleaning procedures

Alignment and cleaning procedures were developed for the optical engine to optimise the LDV data acquisition. These procedures are outlined briefly below.

**Alignment procedure**  The probe volume is aimed at a target zero point when the cylinder head is off. The zero reference point is in centre of cylinder bore in the plane of the piston crown top face at TDC. The probe volume is then moved to a reference target external to the engine. The distance move to the external reference point is recorded as $x_{ref}$. The engine is then reassembled. To get back to the zero point, the probe is first focused on the external target and then moved $-x_{ref}$.

The repeatability of this procedure results in a displacement error of less than 0.3 mm in any one dimension, or less than 0.5 mm overall. This is confirmed by the velocity measurements, where consistent velocity data is obtained at the same data point. This is true regardless of the number of times the engine is reassembled in between data acquisitions.

**Cleaning procedure**  Immediately after the optical engine is reassembled, the best LDV data rates are obtained. For example at $f = (0.0, -20)$ the motored data rates are 25+ kHz per channel, and fired the data rates are 10+ kHz per channel. The data rates slowly drop off as window and mirror get dirty. Once overall motored data rates fall below 10 kHz, or fired data rates fall below 4 kHz, the engine seed flow is turned off and LDV data acquisition stops.

The cleaning procedure begins by running the engine for 5 minutes to flush out any loose seed. The engine is then shut down and the spark plug is removed. The piston is moved to TDC and a flashlight is used to light the combustion chamber. pressurised nitrogen is blown through a hypodermic tube to clean off the surface of
the quartz window beneath the spark plug. Once the window is cleaned, the mirror is removed and cleaned. After reinstalling the mirror, the path of the laser beams is tested to make sure the beams reflected back to the probe are symmetric about the incoming beams. If they are not, the mirror angle is adjusted. The probe volume is aimed at the external target and moved to $-x_{ref}$ to get back to the zero point.

This cleaning procedure allows a 70% data rate recovery. This procedure can only be repeated 2–3 times before the data rates never rise above 10 kHz in the motored case, or 4 kHz in the fired case.

The thorough cleaning procedure requires that the engine be disassembled to get to the quartz window. When cleaning the quartz window it is imperative that absolutely no streaks be left. If there are any streaks at all, the seed will immediately build up in the area and cannot be blown off. That is, data can only be acquired in the initial run for a short period of time. The disassembly and reassembly of the engine takes a total of about 5 hours.